PREFACE

The first international conference on fluid power in Tampere was held in 1987. That was the start of the series of Scandinavian fluid power conferences. In 1993 the conference was named as Scandinavian International Conference on Fluid Power and it was decided to held the conference every second year alternately in Tampere and Linköping. So we have already almost 30 years’ tradition.

The 14th Scandinavian International Conference on Fluid Power conference is organized by Department of Intelligent Hydraulics and Automation (IHA) at Tampere University of Technology (TUT) together with network of Fluid Power Centres in Europe (FPCE).

At this time the conference includes various themes like hybrids, drives, digital hydraulics and pneumatics. Special attention in the program is given for energy efficiency, renewable energy production and energy recovery. They are reflecting well the situation, where environmental issues and energy saving are increasingly important issues.

We received about 125 interesting and high-level abstract proposals. In addition to three invited speakers, about 70 papers were selected for the final programme. This year for the first time in the SICFP conference also peer-review of papers was available for those who asked it and 30 papers passed the evaluation. We appreciate the work what the reviewers have been done. We believe that the conference will give the participants fine opportunities to listen interesting presentations, to exchange opinions and strengthen of old contacts and to establish new ones.

This time the conference proceeding will be published as a printed abstract book and as a USB memory stick and the papers will be publicly available later 2015. We hope that this proceedings will serve you well during the conference but also far in the future as a source of reference.

We would like to express our sincere appreciation to everybody who has contributed to the success of the conference.

Tampere, 8th May, 2015

Kalevi Huhtala
Professor, Conference Chairman
Arto Laamanen
Dr. Tech., Conference Manager
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SICFP15 Exhibitors

Rexroth Bosch Group
HYDAC
John Deere
Ponsse
Sandvik
Siemens
# Programme

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<td>13:00 - 14:30</td>
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<td>B1: Hydraulic Systems</td>
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<td><strong>14:30 - 15:00</strong></td>
<td><strong>Coffee / Exhibition Area</strong></td>
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<tr>
<td>15:00 - 16:30</td>
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<td>B2: Components</td>
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<td><strong>19:00 -</strong></td>
<td><strong>Reception by the City of Tampere, Tampere City Hall</strong></td>
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<td>10:30 - 12:00</td>
<td>A3: Controls</td>
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<td><strong>12:00 - 13:15</strong></td>
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<td>A4: Pumps</td>
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<td>15:15 - 16:45</td>
<td>A5: Robotics</td>
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<tr>
<td><strong>19:00 -</strong></td>
<td><strong>Gala Dinner, Solo Sokos Hotel Torni Tampere</strong></td>
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10:00 - 10:30 Coffee / Exhibition Area
10:30 - 12:00 A6: Hybrids
              B6: Pneumatics
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              B7: Fluid Storages

19:00 - 22:00 Cruise & Dinner (Not included in the basic conference fee)

Peer-reviewed papers are marked with a logo
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<tr>
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| 10:00 - 12:00 | Opening and Invited Speakers  
Session Chair: Kalevi Huhtala, Tampere University of Technology  
- Janne Uotila, Sandvik  
- Ville Kyrki, Aalto University  
- Bernhard Menz & Karl-Heinz Vogl, Bosch Rexroth |
| 12:00      | Lunch                                                 |
| 13:00 - 14:30 | A1: Drives  
Session Chair: Peter Achten, INNAS  
A Hydraulic Hybrid Wheel Loader with a Novel Power Split Hydraulic Transmission  
Feng Wang, Kim Stelson  
Comparison Studies of Different Nonlinear State and Disturbance Estimators for a Hydrostatic Transmission  
Hao Sun, Harald Aschemann, Robert Prabel  
Design Study and Parameter Optimization for a Light-Weight Series Hydraulic Hybrid Vehicle  
Katharina Baer, Liselott Ericson, Petter Krus  
First Experimental Results of a Hydraulic Hybrid Concept System for a Cut-To-Length Forest Harvester  
Kalle Einola, Aleksi Kivi  
Gain Scheduling Full State Feedback with D-Implementation for Velocity Tracking of Hydrostatic Drive Transmission  
Joni Backas, Reza Ghabcheloo, Kalevi Huhtala |
| 14:30      | Coffee                                                |
| 15:00 - 16:30 | A2: Digital Hydraulics  
Session Chair: Matti Linjama, Tampere University of Technology  
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Mitsuaki Hayashi, Yuuichi Miura  
A Linear Valve Actuated Switched Inertance Hydraulic System  
Nathan Peter Sell, David Nigel Johnston, Andrew R. Plummer, Sylwester Kudzma, Min Pan  
On Efficiency of Switched Inertance Control for Hydraulic Systems  
Anton Sinyakov, Pavel Greshnyakov, Asgat Gimadiev, Victor Sverbilov, Andrew Plummer, Nigel Johnston  
Fault Tolerance of Digital Hydraulics in High Dynamic Hydraulic System  
Lauri Siivonen, Mikko Huova, Matti Linjama, Heino Försterling, Edgar Stamm, Till Deubel  
Energy Saving Using a Multi-Chamber Accumulator: Experimental Results and Proof of Concept  
Christian Stauch, Joachim Rudolph |
**Wednesday, May 20, 2015, Session B, Sopraano**

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<td><strong>Session Chair:</strong> Jouni Mattila, Tampere University of Technology</td>
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<td><strong>High Response Overload Protection Valve (HROPV) for Heavy Hydraulics</strong></td>
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<td><strong>Session Chair:</strong> Jouni Mattila, Tampere University of Technology</td>
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<td><strong>Power Plant Fuel Valve Characteristics Considering Hydrodynamic Force</strong></td>
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<td><strong>Frequency Response Correction of Launch Vehicle Fuel Line</strong></td>
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<td><strong>Water Hydraulic Check Valve Researches</strong></td>
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<td>Zaipeng Man, Fan Ding, Minsheng Deng, Shuo Liu</td>
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<td>Juha Lahtinen, Werner Händle</td>
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<td>Asgat Gimadiev, Dmitry Bratchinin, Dmitry Stadnik</td>
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<td>15:00 - 16:30</td>
<td>B2: Components</td>
<td><strong>Investigation of the main impacts on electrostatic charging in filters</strong></td>
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<td><strong>Focus on technical cleanliness in hydraulic manifold system manufacturing</strong></td>
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<td><strong>Design of a High Speed Single Piston Pump for Piston Pair and Slipper Pair Oil Film Investigation</strong></td>
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<td><strong>Constant Improvement in Biohydraulics – A Challenge, a Dream or an Impossibility?</strong></td>
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<td><strong>Application of Nano-Structured Coatings to the Heat Transfer Surface of Heat Exchangers</strong></td>
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<td>Philipp Cedric Weishaar, Hubertus Murrenhoff</td>
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<td>Santtu Pyymäki, Lenna Pitkälä, Jari Rinkinen</td>
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<td>Liming Lao, Hua Zhou, Anhuan Xie, Ruilong Du</td>
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<td>Merja Länsä</td>
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<td>Luca Pastorello, Antonino Bonanno</td>
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<td>Time</td>
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| 9:00 - 10:00 | **Energy Efficient systems**  
*Session Chair: Hubertus Murrenhoff, RWTH Aachen University*  
**Control of a Semi-Binary Hydraulic Four-Chamber Cylinder**  
Edwin Heemskerk, Ralf Bonefeld, Henno Buschmann  
**General Rules for the Design of Efficient Hydrostatic Machines**  
Peter Achten |
| 10:00       | Coffee                      |
| 10:30 - 12:00| **A3: Controls**  
*Session Chair: Andrew Plummer, University of Bath*  
**Load Independent Velocity Control on Boom Motion Using Pressure Control Valve**  
Jesper Kirk Sørensen, Michael Rygaard Hansen, Morten Kjeld Ebbesen  
**P-Type Iterative Learning Control for two Coupled Hydraulic Cylinders**  
Robert Prabel, Harald Aschemann  
**Design of Disturbance Observer of Electro-Hydraulic Loading System for Helicopter Manipulating Booster**  
Yunhua Li, Zhiqing Sheng, Shaoping Wang  
**Adaptive Damp Control of Drilling String for Offshore Platform Passive Compensator Under Different Sea Conditions**  
Zhengzhe Cui, Yinglong Chen, Hua Zhou, Huayong Yang  
**Hardware-In-The-Loop Electronic Control System for a Universal Test Rig For Hydraulic Servo Cylinders Motion Synchronization and Servo Valves Testing**  
Taher Mohamed Salah EL Din Fahmy, Saad Abd elfattah Kassem |
| 12:00       | Lunch                       |
| 13:15 - 14:45| **A4: Pumps**  
*Session Chair: Victor Juliano De Negri, Federal University Of Santa Catarina*  
**Design of a Vane Pump Power Split Transmission for a Highway Vehicle**  
Biswa Ranjan Mohanty, Feng Wang, Kim A. Stelson  
**Modeling and Simulation of Thermal Hydraulic Coupling in Electro-Hydrostatic Modules Involving Fixed-Displacement Vane Pumps**  
Emanuele Gnesi, Jean-Charles Maré, Jean Luc Bordet  
**A Novel Concept for a Variable Delivery External Gear Machine**  
Ram Sudarsan Devendran, Andrea Vacca  
**Leakage Past Active Contacts in Involute and Cycloidal Gear Hydrostatic Units**  
Rathindranath Maiti, Manab Kumar Das, Vineet Sahoo, Krishna Chaitanya Avula, Anukaran Arzare, Vishva Prakash Tolambia, Abhijit Nag  
**Simulation Analysis of Ring Gear’s Micro Motion in Internal Gear Machines**  
ZHOU Hua, DU Ruilong, YANG Huayong, LV Chenghui |
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**Session Chair:** Marcus Geimer, Karlsruhe Institute of Technology

**Design Overview of the Hydraulic Quadruped Robots HyQ2Max and HyQ2Centaur**
Claudio Semini, Jake Goldsmith, Bilal Ur Rehman, Marco Frigerio, Victor Barasuol, Michele Focchi, Darwin G. Caldwell

**DEVELOPMENT OF A LIGHTWEIGHT ON-BOARD HYDRAULIC SYSTEM FOR A QUADRUPED ROBOT.**
Hamza Khan, Satoshi Kitano, Yifu Gao, Darwin G. Caldwell, Claudio Semini

**Vehicle Mass Estimation for Hydraulic Drive System using Longitudinal Motion Model**
Miika Ahopelto, Tomi Krogerus, Kalevi Huhtala

**NOVEL HAPTIC CONTROLLER FOR NON-ROAD MOBILE MACHINE TELEOPERATION**
Jani Erik Heikkinen, Heikki Handroos, Takao Nishiumi

**LOW-COST 3D LIDAR FOR THE MAPPING OF AUTONOMOUS MOBILE WORK MACHINE**
Antti Kolu, Kimmo Rajapalvi, Mika Hyvönen, Petteri Multanen, Kalevi Huhtala
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| 15.15 – 16.45 | B5: Energy Efficiency  
*Session Chair: Huayong Yang, Zhejiang University*  
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Damiano Padovani, Monika Ivantysynova  
Investigation and Improvement of the Energy Efficiency of Hydraulic Deep Drawing Presses  
Harald Lohse, Jürgen Weber, Sebastian Neumann, Werner Händle, Dirk Klug  
Analysis of a hydrostatic transmission system for horizontal axis wind turbines  
Eduardo Augusto Flesch, Henrique Raduenz, Victor Juliano De Negri  
New ISO25119 Compliant 6-Indipendent Wheels Electro-Hydraulic Steering System for Agricultural Machine  
Massimiliano Ruggeri, Andrea Cervesato  
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*Session Chair: Monika Ivantysynova, Purdue University*  
**Modular Software Design of Safety Related Systems for Mobile Machinery – Reliability, Testability and Simulation**  
*Cornelia Weltzien, Erik Lautner*  
**Digital Hydraulics on Rails – Pilot Project of Improving Reliability on Railway Rolling Stock by Utilizing Digital Valve System**  
*Helmut Fischer, Arto Laamanen, Anssi Iso-Heiko, Oliver Schäfer, Matti Karvonen, Otso Karhu, Kalevi Huhtala, Veli-Pekka Pulkkinen, Ali Huttunen* |
| 10:00      | Coffee                                       |
| 10:30 - 12:00 | **A6: Hybrids**  
*Session Chair: Kim Stelson, University of Minnesota*  
**Hybrid Load Sensing – Displacement Controlled Architecture for Excavators**  
*Ken Sugimura, Hubertus Murrenhoff*  
**Hybrid Pump Drive**  
*Seppo Tikkanen, Henrik Tommila*  
**Comparative study of fuel reduction methods for hybrid excavators**  
*Qian Zhu, Qingfeng Wang*  
**Hydraulic hybrid actuator - theoretical aspects and solution alternatives**  
*Matti Linjama, Mikko Huova, Matti Pietola, Jyri Juhala, Kalevi Huhtala*  
**Improving Energy Efficiency of a Reach Stacker Using a Potential Energy Recovery System**  
*Thomas Schaep, Wilfrid Marquis-Favre, Eric Bideaux, Eric Noppe, Pierre Rodot, Jean-Christophe Bernigaud, Vincent Langlois* |
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*Session Chair: Seppo Tikkanen, Tampere University of Technology*  
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*Yaoxing Shang, Xiaoshu Zhang, Zongxia Jiao, Shaoping Wang, Liang Yan*  
**Review of hydraulic technologies in wind turbines**  
*Le Tu, Wei Li, Yonggang Lin, Hongwei Liu*  
**Research on the Stiffness of the Hydraulic Transformer Controlled System**  
*Chongfeng Di, Wei Wu, Jibin Hu, Shihua Yuan*  
**Dynamics of Volume Controlled Mechanical Ventilation System**  
*Yan Shi, Jinglong NIU, Maolin CAI, Weiqing XU* |
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| 10:30 - 12:00 | **B6: Pneumatics**  
*Session Chair: Songjing Li, Harbin institute of technology*  
**Development of A Parallel Valve Control for A Hot Gas Bulge Test**  
Johannes Storz  
**Development of Experimental Equipment for the Analysis of Flowmeter Characteristics in Conditions of Gas Pulsating Flow**  
Asgat Gimadiev, Ilyas Kashapov, Marat Gimadiev  
**Energy Efficiency Comparisons of Pneumatic Systems: Effects In After Treatment: Theory and Verification with Time Series Measurements**  
Jyrki Parkkinen  
**Design And Fabrication of An Electromagnetic Microvalve for Pneumatic Control of Microfluidic Systems**  
Xuling Liu, Songjing Li  
**Simplified Fluid Transmission Line Model for Pneumatic Control Applications**  
David Rager, Rüdiger Neumann, Hubertus Murrenhoff |
| 12:00         | Lunch                                                                |
| 13:15 - 14:45 | **B7: Fluid Storages**  
*Session Chair: Matti Pietola, Aalto University*  
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Elliott Gruber, Kenneth Cunefare  
**An Approach to Optimize the Design of Hydraulic Reservoirs**  
Alexander Wohlers  
**Second Order Dynamic Accumulators, the Features, the Applications and the Feasibility**  
Mohamed Ahmed Elgamil, Ahmed Rabie Abdelbaki, Chahinaz Abdelrahman Saleh |
A HYDRAULIC HYBRID WHEEL LOADER WITH A NOVEL POWER SPLIT HYDRAULIC TRANSMISSION

Feng Wang and Kim A. Stelson
Center for Compact and Efficient Fluid Power
Department of Mechanical Engineering, University of Minnesota
Minneapolis, MN 55455 USA
E-mail: wang2148@umn.edu, kstelson@umn.edu

ABSTRACT

A novel power split hydraulic transmission has been applied to the drivetrain of a hydraulic hybrid wheel loader in this paper. Unlike a typical power split hydraulic transmission consisting of planetary gear set and hydrostatic units, the new transmission consists of a vane pump based hydrostatic transmission (vHST) and a variable displacement motor. The vHST functions like a conventional hydrostatic transmission (HST) but has a different form. It uses a double-acting vane pump with a floating ring. By coupling the floating ring to an output shaft, the vane pump becomes a hydraulic transmission. The vHST combines the pumping and motoring functions in one unit, making it much simpler than a conventional HST. By delivering the vHST output flow to a variable displacement motor coupled to the vHST output shaft, a hydraulic power split transmission is created. In this paper, a hydraulic hybrid wheel loader with this new power split hydraulic transmission was studied. A dynamic simulation model was built to help the system parameter design and to verify the system performance. Simulation results have demonstrated the feasibility of applying this new transmission to a hydraulic hybrid wheel loader.

KEYWORDS: Hydraulic hybrid wheel loader, power split hydraulic transmission, vane pump

1. INTRODUCTION

Hydraulic hybrid drivetrains are an effective approach to improving the fuel economy of modern heavy-duty vehicles. The high power density of the hydraulic powertrain allows for lower vehicle weight, more regenerative braking and faster acceleration. Comparing hydraulic and electric hybrids, a hydraulic accumulator has a longer lifetime than a battery, reducing the maintenance cost considerably [1, 2]. Hydraulic hybrids have shown great advantages in on-road heavy duty vehicles, such as Eaton’s Hydraulic Launch Assist (HLA) system [3], EPA’s world first series hydraulic hybrid delivery truck [4], Lightning Hybrids’ parallel hydraulic hybrid system [5], Altair’s series hydraulic hybrid city bus [6], and Parker Hannifin’s hydraulic power split system [7].

Few efforts were made on the hybridization of the off-road vehicles, a notable exception being the Caterpillar 336E H hydraulic hybrid excavator [8]. The challenge comes from the variety of applications, ranging from agricultural and construction machines to forestry equipment, since the hybrid solutions are largely cycle-dependent. Unlike agricultural and forestry machines moving on muddy ground, the ground condition of the wheel loader allows for regenerative braking. Medium or large size wheel loaders are usually designed for a
single application such as loading material. This means the duty cycle is highly repetitive. In a typical loading cycle, there are many accelerations and decelerations, showing large hybridization potential [9].

There are some studies conducted on the drivetrain hybridization of wheel loaders. Achten proposed a new hydraulic hybrid system consisting of hydraulic transformers for both the wheel drive and the working functions in a large wheel loader. Simulation results show a fuel consumption reduction of about 50% in a short loading cycle [10]. A power split electric hybrid compact wheel loader with planetary gear set was studied at the University of Pisa. Simulation results show it reduces the fuel consumption about 12% compared with a hydrostatic drivetrain [11]. Volvo released its parallel hybrid wheel loader – Volvo L220F Hybrid, claiming to offer 10% lower fuel consumption than the non-hybrid version and also higher productivity [12]. Hitachi built an electric hybrid wheel loader, with the test results showing the fuel consumption reduction by 30-40% compared to a hydrostatic drivetrain in the short loading cycle [13]. Deutz, Atlas-Weyhausen and Heinzmann joined together to develop a hybrid wheel loader prototype - Atlas AR 65-Hybrid, which has a fuel economy improvement ranging from 15% to 20% over conventional machines [14].

A hydraulic continuously variable transmission (CVT), a hydrostatic transmission, is a good candidate for the drivetrain hybridization since it can decouple the engine speed from vehicle speed. Combined with an energy storage device such as a hydraulic accumulator, full engine management becomes possible. However, the inefficiencies of hydraulic pumps or motors have limited the application of hydrostatic transmissions in hybrid vehicles. One way to make an HST more competitive is to combine it with a gearbox to create a power split transmission, which combines the high efficiency of a gearbox and the continuously variable ratio of an HST.

A typical power split transmission employs a planetary gear set as the power-split device. Although this transmission is more efficient than an HST, it is still bulky due to the heavy gears. The initial cost is also relatively high due to the costly variable displacement pump or motor [15,16]. Therefore in this paper a more compact and cost-effective power split hydraulic transmission is proposed and applied to a hybrid wheel loader. The drivetrain parameter design and its controller design are introduced. A dynamic simulation model is built to help the system parameter design and to verify the system performance.

2. HYDRAULIC HYBRID WHEEL LOADER

2.1. New power split hydraulic transmission

The key component of the new power split hydraulic transmission is the vane pump based hydrostatic transmission (vHST). The structure of the vHST and its hydraulic symbol are shown in Figure 1. It is based on a double-acting vane pump since it has a balanced design and therefore a longer lifetime and quieter operation than a gear or piston pump [17]. The hydraulic symbol is newly proposed to reflect the fact that it has a floating ring. The rotor with the vanes is coupled to the input shaft and the floating ring is coupled to the output shaft.

The vHST combines both the pumping and motoring function in one single unit, making it function like a conventional HST. The pumping unit, consisting of the input shaft, rotor/vane assembly and the floating ring, while the motoring unit consists of the floating ring and the output shaft coupled to it. The fluid pressure inside the vHST acts on the pumping and motoring unit simultaneously, thus transfers the power. The delivery flow at the output port is proportional to the relative rotary speed between the input and output shafts [18,19].

Compared to a conventional HST, the vHST reduces the parasitic losses in the pump and motor and eliminates the line losses in between, making the unit simpler and more efficient. Another reason why the vHST is more efficient than a conventional HST is that the viscous drag between vane tips and floating ring helps to drive the output shaft, in contrast to conventional HST where the case is stationary causing more of the viscous drag energy to be converted into heat.
By coupling the floating ring to the output shaft, the vHST splits the mechanical power on the input shaft into mechanical power on the output shaft and hydraulic power at the output port. With a relief valve, the vHST always discharges oil so that the output shaft never turns faster than the input shaft. In an extreme condition when the output port is plugged, the input and output shafts have the same rotary speed. This turns the vHST to a direct drive, efficiently transferring the torque between the input and output shafts.

Instead of delivering the flow to a relief valve, a new power split hydraulic transmission is created by delivering the flow to a variable displacement motor coupled to the vHST output shaft, as shown in Figure 2.

In the new transmission, the hydraulic power is turned into mechanical power through a variable displacement motor and added to the vHST output shaft. By changing the motor displacement, the power transferred through mechanical path and hydraulic path can be adjusted [20].

2.2. Hydraulic hybrid wheel loader
The drivetrain schematic of the hydraulic hybrid wheel loader with new power split hydraulic transmission is shown in Figure 3. Since this study only focuses on the drivetrain design, other subsystems such as working hydraulic system, power steering and fan drive system are not included in the figure.

![Figure 3. Drivetrain schematic of hydraulic hybrid wheel loader with new power split hydraulic transmission](image)

The input shaft of the power split hydraulic transmission is coupled to the engine shaft and the output shaft is coupled to the final drive. A hydraulic accumulator is connected to the high pressure line. The power split hydraulic transmission shown in Figure 3 is slightly different from the one shown in Figure 2. The variable motor in Figure 3 can run either in motoring or pumping mode while the variable motor in Figure 2 only runs in motoring mode. This is because the vHST cannot absorb flow. With an accumulator, regenerative braking through the variable motor becomes possible.

A small variable displacement pump is connected to the input shaft of the vHST, between the engine and the vHST. The output of this control pump is connected to the high pressure line. To run the engine more efficiently, an energy management strategy generates optimal engine operation points (engine torque and speed). The engine has its controller to control the engine speed by adjusting the throttle. However the engine controller itself cannot control the engine torque. To track the desired engine torque, the engine load needs to be adjusted. A variable displacement pump is needed to adjust the engine load since the displacement of the vHST is fixed. Together with the engine speed control, the optimal engine torque and speed generated by the energy management system can be achieved.

3. POWER SPLIT HYBRID DRIVETRAIN

3.1. Drivetrain model

As shown in Figure 3, the engine power is transferred to the wheel through both the mechanical and hydraulic path. The modes of the variable motor (motoring or pumping) and the accumulator (charging or discharging) depend on the system operation. The following sign conventions are defined:

1. The engine speed and final drive speed are always positive (a reverse gear is used when the vehicle moves backward);
2. For the control pump, vHST and variable motor, the torque is positive when motoring and negative when pumping; the displacement and flow are positive when motoring and negative when pumping;
3. The accumulator flow is positive when discharging and negative when charging.

The engine shaft dynamics is:

\[(J_e + J_c + J_{vp})\omega_e = T_e - (-T_c) - (-T_{vp})\]  \(\text{(1)}\)

where \(J_e, J_c\) and \(J_{vp}\) are the engine, control pump and vHST pump moments of inertia, \(\omega_e\) is the engine speed, \(T_e, T_c\) and \(T_{vp}\) are the engine, control pump and vHST pump torque. There are negative signs on the control pump and vHST pumping torque, which are due to the pumping sign convention.
The final drive shaft dynamics is:

\[(J_{vm} + J_m + J_f)\omega_f = T_{vm} + T_m - T_f\]  \hspace{1cm} (2)

where \(J_{vm}, J_m\) and \(J_f\) are the vHST motoring, variable motor and final drive moments of inertia, \(\omega_f\) is the final drive speed, \(T_{vm}, T_m\) and \(T_f\) are the vHST motoring, variable motor and final drive load torque.

The final drive moment of inertia is:

\[J_f = \frac{mR_{wh}^2}{k_f^2}\]  \hspace{1cm} (3)

where \(m\) is the vehicle mass, \(R_{wh}\) is the radius of the wheel and \(k_f\) is the final drive ratio.

The final drive load torque is:

\[T_f = \frac{1}{k_f}(\mu m g + \frac{1}{2} \rho v^2 C_D A)R_{wh}\]  \hspace{1cm} (4)

where \(\mu\) is the rolling friction coefficient of the wheel, \(g\) is the gravity, \(\rho\) is the air density, \(v\) is the vehicle speed, \(C_D\) is the vehicle drag coefficient, \(A\) is the front area of the vehicle.

The final drive speed is:

\[\omega_f = k_f \frac{v}{R_{wh}}\]  \hspace{1cm} (5)

The torque and flow of the control pump, \(T_c\) and \(Q_c\), are:

\[T_c = \frac{p x_c D_c}{\eta_{cm}}\]  \hspace{1cm} (6)

\[Q_c = x_c D_c \omega_e \eta_{cv}\]  \hspace{1cm} (7)

where \(x_c\) is the control pump displacement fraction \((-1 \leq x_c \leq 0\), only in pumping mode), \(D_c\) is the full displacement of the control pump, \(\eta_{cm}\) and \(\eta_{cv}\) are the mechanical and volumetric efficiencies of the control pump, and \(p\) is the accumulator pressure.

The input and output shaft torque of the vHST, \(T_{vp}\) and \(T_{vm}\), are:

\[T_{vp} = k_{vp} \frac{p D_{vHST}}{\eta_{vpm}}\]  \hspace{1cm} (8)

\[T_{vm} = k_{vm} p D_{vHST} \eta_{vvm}\]  \hspace{1cm} (9)

where \(D_{vHST}\) is the displacement of the vHST, \(\eta_{vpm}\) and \(\eta_{vvm}\) are the mechanical efficiencies of the vHST pumping and motoring unit, \(k_{vp}\) is the vHST pumping sign \((k_{vp} = -1)\), \(k_{vm}\) is the vHST motoring sign \((k_{vm} = +1)\).

The output flow of the vHST, \(Q_{vp}\), is:

\[Q_{vp} = k_{vp} D_{vHST}(\omega_e - \omega_f)\eta_{vpv}\]  \hspace{1cm} (10)

where \(\eta_{vpv}\) is the volumetric efficiency of the vHST.

The torque and flow of the variable motor, \(T_m\) and \(Q_m\), are:

\[T_m = px_m D_m \eta_{mm}^{km}\]  \hspace{1cm} (11)

\[Q_m = x_m D_m \omega_f \eta_{mm}\]  \hspace{1cm} (12)
where \( x_m \) is the variable motor displacement fraction \((-1 \leq x_m < 0\) in pumping mode, \(0 \leq x_m \leq 1\) in motoring mode), \( D_m \) is the full displacement of variable motor, \( \eta_{mm} \) and \( \eta_{mv} \) are the mechanical and volumetric efficiencies of variable motor, \( k_m \) is the variable motor index \((k_m = -1\) for pumping and +1 for motoring).

The accumulator flow, \( Q \), is:

\[
Q = Q_c + Q_{vp} + Q_m
\]  

(13)

The state of charge, \( SOC \), is defined as:

\[
SOC = \frac{p - p_{min}}{p_{max} - p_{min}}
\]  

(14)

where \( p_{max} \) and \( p_{min} \) are the maximum and minimum accumulator pressures.

The accumulator pressure is:

\[
p = (p_{max} - p_{min})SOC + p_{min}
\]  

(15)

Thus,

\[
\dot{p} = (p_{max} - p_{min})\dot{SOC}
\]  

(16)

Assuming the gas change in the accumulator is isothermal, the gas volume in the accumulator, \( V_g \), is:

\[
V_g = \frac{p_0 V_0}{p}
\]  

(17)

where \( p_0 \) is the accumulator precharge pressure and \( V_0 \) is the accumulator volume.

The accumulator flow is:

\[
Q = \dot{V}_g = -\frac{p_0 V_0}{p^2} \dot{p}
\]  

(18)

The accumulator flow is positive when discharging and negative when charging.

### 3.2. Controller design

There are two variable displacement units (control pump and variable motor) in the hybrid powertrain as shown in Figure 3. As discussed previously, the control pump is used to bring the engine to the optimal point (engine torque and speed) together with the engine speed controller. The variable motor is used to control the wheel loader speed. Two controllers are developed in the system: the engine torque/speed controller and the vehicle speed controller.

To run engine more efficiently, the engine torque/speed controller is developed to track the optimal engine point (engine torque and speed) by adjusting the engine load. This optimal engine point is the most efficient point on the engine map for the desired power. Together with the engine throttle control, the desired engine operating point generated by the high-level energy management strategy can be tracked. To reject disturbances from the engine and line pressure, the engine speed error is fed to a proportional (P) controller to generate the control pump displacement fraction command. The strategy could be either a simple rule-based one or a complex one based on dynamic programming.

The vehicle speed controller controls the wheel loader speed. In a non-hybrid wheel loader, the accelerator and brake pedal control the vehicle speed. In a hydraulic hybrid wheel loader, adjusting the variable motor torque controls the vehicle speed. The variable motor has an over-center function so it can run in motoring or pumping mode, adding or absorbing torque to or from the final drive. A proportional plus integral (PI) controller uses vehicle speed error to generate the motor displacement fraction command thus simulating driver behavior.
3.3. Energy management strategy

A good energy management strategy is essential to improving the fuel efficiency of hybrid vehicles. The main objective of the energy management system is to distribute the engine and energy storage power to the wheel in an effective and efficient way. Although there are many advanced energy strategies, a simple rule-based strategy is used in this study since the focus of this paper is the hybrid powertrain design. More advanced energy strategies for this powertrain will be considered in the future.

One rule used in this study is to make the input shaft speed of the vHST always higher than the output shaft speed since the vHST cannot absorb flow. The condition is met by making the engine speed proportional to the vehicle speed. An appropriate final drive ratio is chosen so that the input shaft speed of the vHST is always higher than the output shaft speed at all vehicle speeds.

Another rule is to choose the engine torque based on the current SOC. To maintain the controllability of the engine speed, the engine torque needs to be higher than the vHST pumping torque which is proportional to the line pressure and the SOC. This “excess” engine torque also needs to take the initial and final SOC into consideration. In this study, the excess engine torque is chosen to make the final SOC the same as the initial SOC during one duty cycle. More details of the rule-based strategy are described in section 4.2.

4. SIMULATION STUDY

4.1. Simulation parameters

The simulation parameters of the model are listed in Table 1. The case studied is a compact wheel loader with the maximum engine power of 58 kW and operating weight of 5680 kg. The wheel loader is similar to the Volvo L30G (maximum engine power of 55 kW and operating weight of 5500 kg) or the Caterpillar 906H2 (maximum engine power of 55 kW and operating weight of 5630 kg). The cycle used in the simulation is the short loading cycle described in our previous work [21,22].

Table 1. Simulation parameters of the power split hydraulic hybrid wheel loader

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle</td>
<td></td>
<td></td>
</tr>
<tr>
<td>operating weight</td>
<td>5680</td>
<td>kg</td>
</tr>
<tr>
<td>differential ratio</td>
<td>3.2</td>
<td>-</td>
</tr>
<tr>
<td>planetary ratio</td>
<td>4.0</td>
<td>-</td>
</tr>
<tr>
<td>wheel radius</td>
<td>0.452</td>
<td>m</td>
</tr>
<tr>
<td>tire friction coefficient</td>
<td>0.02</td>
<td>-</td>
</tr>
<tr>
<td>drag coefficient</td>
<td>0.3</td>
<td>-</td>
</tr>
<tr>
<td>front area</td>
<td>5.574</td>
<td>m²</td>
</tr>
<tr>
<td>air density</td>
<td>1.2</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Engine</td>
<td></td>
<td></td>
</tr>
<tr>
<td>max engine power</td>
<td>58</td>
<td>kW     @2800rpm</td>
</tr>
<tr>
<td>max engine torque @1900rpm</td>
<td>217</td>
<td>Nm</td>
</tr>
<tr>
<td>engine inertia</td>
<td>0.18</td>
<td>kg⋅m²</td>
</tr>
<tr>
<td>Control pump</td>
<td></td>
<td></td>
</tr>
<tr>
<td>displacement</td>
<td>10</td>
<td>cm³/rev</td>
</tr>
<tr>
<td>mech./vol. eff.</td>
<td>0.90</td>
<td>-</td>
</tr>
<tr>
<td>vHST</td>
<td></td>
<td></td>
</tr>
<tr>
<td>displacement</td>
<td>20</td>
<td>cm³/rev</td>
</tr>
<tr>
<td>pumping mech. eff.</td>
<td>0.90</td>
<td>-</td>
</tr>
<tr>
<td>motoring mech. eff.</td>
<td>0.95</td>
<td>-</td>
</tr>
<tr>
<td>vol. eff.</td>
<td>0.90</td>
<td>-</td>
</tr>
<tr>
<td>Variable motor</td>
<td></td>
<td></td>
</tr>
<tr>
<td>displacement</td>
<td>107</td>
<td>cm³/rev</td>
</tr>
</tbody>
</table>
Constant efficiencies are used for pumps and motors in the simulation since the focus of this study is to prove the concept instead of making detailed fuel economy evaluations. These efficiencies agree with conventional engineering practice. Since the viscous drag between the vane tips and the ring helps to drive the output shaft of the vHST, a higher than usual motoring mechanical efficiency is used in the vHST.

4.2. Simulation results

The vehicle speed control is shown in Figure 4. The vehicle speed is tracked by controlling the variable motor displacement based on vehicle speed error. The first plot shows good tracking performance (the two curves overlap). The vehicle speed is defined in a short loading cycle consisting of several repeated accelerations and decelerations. The second plot shows how the vHST and the variable motor help to drive the final drive. It shows that the majority of the final drive torque is provided (acceleration) or absorbed (deceleration) by the variable motor. The vHST only provides a small motoring torque through its output shaft. The third plot shows the SOC change and the line pressure (accumulator pressure) change. The SOC is defined as a function of the accumulator pressure and is proportional to the accumulator pressure. In the simulation, the accumulator pressure is 28 MPa @ SOC = 1 and 12 MPa @ SOC = 0. The SOC and the accumulator pressure show the same change in the plot. The last plot shows the variable motor displacement fraction change. Results show that the accumulator discharges during the acceleration and charges during the deceleration and zero vehicle speed (when the vehicle is stopped for loading material or dumping).

![Figure 4. Vehicle speed control (line pressure: accumulator pressure)](image-url)

The engine speed control is shown in Figure 5. The vehicle speed is shown in the first plot for reference. The second plot shows the engine speed tracking performance. The engine speed is controlled by controlling the control pump displacement. A simple rule-based strategy is used in this study where the input shaft speed of the vHST is always higher than the output shaft speed throughout the cycle since the vHST cannot absorb flow.

It is shown in Figure 5 that the engine speed is well tracked by using the control pump. The third plot shows the engine, control pump and the vHST pump torque. Both the control pump and the vHST pump torque are negative due to the sign convention. The last plot shows the displacement fraction of the control pump (negative due to the sign convention).
The power flow in the drivetrain is shown in Figure 6. The first plot shows the vehicle speed for reference. The second plot shows the engine and the final drive speed, where the final drive speed is proportional to the vehicle speed. The engine speed is constant at 900 rpm when the vehicle speed is below some point and then increases with the vehicle speed. The engine speed is always higher than the final drive speed to assure normal operation of the vHST. The third plot shows the flow distribution in the drivetrain. When the vehicle speed is below some point, the vHST charges the accumulator a lot. During large vehicle acceleration the majority of the variable motor flow is provided by the accumulator. The variable motor flow is fully absorbed by the accumulator during deceleration. The last plot shows the engine, accumulator and the final drive power. It is shown that the final drive power during large acceleration is mainly from the accumulator. The engine drives the vHST to charge the accumulator when there is low power demand from the final drive.

The power distribution of the vHST is shown in Figure 7. The second plot shows the comparison between engine speed (equal to vHST pump speed) and final drive speed (equal to vHST motor speed). The third plot shows the vHST pump and vHST motor power. The difference between the two is the power transferred through the hydraulic path of the vHST. A power split factor is defined for the vHST, indicating the fraction of the power transferred through the mechanical path. The power split factor throughout the cycle is shown in the last plot. The power split factor is higher at the higher vehicle speeds. A different engine control strategy may result in a different power split factor.
5. CONCLUSIONS

A novel power split hydraulic transmission has been applied to the drivetrain of a hydraulic hybrid wheel loader in this paper. The key component of the new transmission is a vane pump based hydrostatic transmission (vHST). The vHST is based on a double acting vane pump with a floating ring. By coupling the floating ring to output shaft, it becomes a transmission. The output flow of the vHST is proportional to the relative rotary speed between the input and output shafts.

The vHST combines the pumping and motoring functions in one unit. Compared to a conventional HST, the vHST reduces the parasitic losses in the pump and motor and eliminates the line losses in between, making the unit simpler and more efficient. By delivering the flow of the vHST to a variable displacement motor coupled to the vHST output shaft, a power split hydraulic transmission is created.

The hydraulic hybrid wheel loader with the new power split hydraulic transmission has been proposed in this paper. Study shows that a small variable displacement control pump is necessary to run the engine at the desired point together with the engine throttle control. The vehicle speed is tracked by controlling the variable motor displacement. With a hydraulic accumulator, regenerative braking is possible. A dynamic simulation model was built to help the system parameter design and to verify the system performance. A simple rule-based energy management strategy was developed. Simulation results have proved the concept and demonstrated the feasibility of applying this new transmission to a hydraulic hybrid wheel loader.

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REFERENCES


COMPARISON STUDIES OF DIFFERENT NONLINEAR STATE AND DISTURBANCE ESTIMATORS FOR A HYDROSTATIC TRANSMISSION

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ABSTRACT

In this paper, different state and disturbance estimation techniques are investigated for the tracking control of a hydrostatic transmission, which is typically used in the drive trains of working machines. A nonlinear control-oriented model of a hydrostatic transmission is derived, in which the system disturbances – the leakage volume flow and a resulting disturbance torque – are modelled as lumped parameters. The implemented optimal control structure requires a feedback of all the system states and disturbances. Therefore, the controller is extended with state and disturbance estimators providing estimates of the unknown disturbances as well as the unmeasurable system states – the normalised tilt angles of the pump $\tilde{\alpha}_P$ and the motor $\tilde{\alpha}_M$. In this contribution, a nonlinear reduced-order observer, an Extended Kalman Filter as well as an Sigma-Point Kalman Filter are designed and implemented. A validation as well as a comparison among these estimation techniques are performed by both simulations and experiments.

KEYWORDS: hydrostatic transmission, state and disturbance observer, optimal control

1. INTRODUCTION

A hydrostatic transmission as depicted in Fig. 1 uses hydraulic oil to transmit power from the power source to the drive mechanism. A basic hydrostatic transmission consists of a hydraulic pump and a hydraulic motor, of which at least one must have a variable displacement, operating together in a closed circuit. The mechanical torque provided from the engine is transformed by the hydraulic pump into a pressurised fluid flow that is transformed back to mechanical torque by the hydraulic motor. By varying the tilt angle and, thereby, changing the displacement of either pump or motor, any desired transmission ratio can be obtained within the boundaries determined by the mechanical design, cf. [1].

Hydrostatic transmissions are widely used as a characteristic component of drive chains in practically all types of working machines like harvesters, wheel loaders, excavators, telehandlers and agricultural tractors. They are typically operated in combination with diesel engines for mobile applications and offer a variety of advantages in comparison with pure mechanical transmissions. Besides the capability of a continuously variable transmission with high power density and the generation of large traction forces at low speeds, hydrostatic transmissions allow for reversing the direction of rotation without changing the gear. Moreover, it is possible to perform wearless breaking manoeuvres. Due to the significant advantages in comparison with mechanical gearboxes, hydrostatic transmissions have also attracted the attention of engineers from the renewable
energy sector, cf. [2] and [3]. Hydrostatic transmissions, however, are subject to several nonlinearities and characterised by uncertain system parameters as well as unknown disturbances. Even so, gain-scheduled PID-controllers are still widely used in current industrial practice for the feedback control of such systems, see [4]. To improve both the energy efficiency and the control performance in practical applications, however, nonlinear control approaches and efficient drive structures like power-split drive trains must be considered. Regarding nonlinear control, plenty of work has been devoted in the last two decades, see [6-15]. A flatness-based controller is proposed in [5], in which only the unknown disturbance torque was considered as a lumped parameter. A disturbance compensation is realised by employing a nonlinear reduced-order disturbance observer. The simulation results and the experimental evaluation show a good tracking accuracy as well as active damping of pressure oscillations. The simplifying assumptions concerning the actuator dynamics and a constant leakage coefficient, however, restrict the applicability of this approach. In subsequent work [8-10], several advanced nonlinear approaches like adaptive inverse dynamics, robust inverse dynamics and sliding mode control have been investigated for tracking control of the hydrostatic transmission system, in which the actuator time constants and leakage volume flow are considered as uncertain parameters and disturbance input, respectively. These centralised control approaches have been evaluated by simulations only. A decentralised flatness-based controller is presented in [9]. This innovative approach leads to a high tracking accuracy for both controlled outputs, and a singularity due to the vanishing pressure difference, cf. [5], can be avoided. Moreover, the implementation of the decentralised control structure is even simpler than the centralised versions. Thereafter, several innovative nonlinear control approaches were proposed and validated in experiments, cf. [15, 16, 17].

Using the decentralised optimal control structure proposed in [15], a comparison of different state and disturbance estimators for tracking control of a hydrostatic transmission is carried out based on both simulation and experimental results. This paper is organised as follows: In Sect. 2, the model of the mechatronic drive system is briefly described. The decentralised optimal control structure is introduced in Sect. 3. In Sect. 4, the alternative state and disturbance estimators are discussed. In Sect. 5, both simulation results considering measurement noise as well as quantization errors, and experimental results are presented. Finally, conclusions and a short outlook on future work are given in Sect. 6.

2. CONTROL-ORIENTED MODELLING OF A HYDROSTATIC TRANSMISSION

A hydraulic scheme of the considered test rig is depicted in Fig. 2 together with the necessary components. This overall system can be split into a hydraulic subsystem and a mechanical subsystem, which are coupled by the torque generated by the hydraulic motor.
2.1. Hydraulic subsystem

With the assumptions of a small swashplate angle $|\alpha_P| \leq 18^\circ$ of the hydraulic pump and a small bent axis angle $|\alpha_M| \leq 20^\circ$ of the hydraulic motor, cf. [6], the pump and motor volume flow can be described as follows

$$q_P = \bar{V}_P \bar{\alpha}_P \omega_P, \quad q_M = \bar{V}_M \bar{\alpha}_M \omega_M. \quad (1)$$

Here, $\bar{V}_P$ and $\bar{V}_M$ are constant parameters resulting from the geometric structure of the hydraulic pump and motor. The variable $\bar{\alpha}_P \in (-1, 1)$ is the normalised swashplate angle of the pump, whereas $\bar{\alpha}_M \in (\varepsilon_M, 1), \varepsilon_M > 0$ denotes the normalised bent axis angle of the motor. The angular velocity of the pump and motor is represented by $\omega_P$ and $\omega_M$, respectively.

Neglecting pressure losses in the hydraulic hoses and introducing a reasonable symmetry assumption, an order reduction can be achieved regarding the pressure dynamics. This results in a first-order differential equation for the difference pressure $\Delta p$

$$\Delta p = \frac{2}{C_H} \left( \bar{V}_P \bar{\alpha}_P \omega_P - \bar{V}_M \bar{\alpha}_M \omega_M - \frac{q_U}{2} \right). \quad (2)$$

Here, the hydraulic capacitance is given by $C_H = \frac{V}{\beta}$, where $\beta$ denotes the effective bulk modulus of the fluid, and $V$ is the total volume (hydraulic hose and chamber) of one pressure side. The corresponding leakage volume flow $q_U$ results from internal and external leakage volume flows in the system.

The dynamics of the displacement units for both pump and motor are modelled by first-order lag systems, respectively. With $i \in \{P, M\}$, the differential equation for the corresponding normalised tilt angle becomes

$$T_{ui} \dot{\bar{\alpha}}_i + \bar{\alpha}_i = k_i u_i. \quad (3)$$

The actuator time constants are denoted by $T_{ui}, i \in \{P, M\}$, and the input voltages $u_i$ of the corresponding proportional valves for the displacement units serve as physical control inputs. Furthermore, $k_i$ represent the proportional gains of the first-order lag systems. Saturation functions account for the limited outputs of the actuators

$$\text{sat}^a_b(\bar{\alpha}) = \begin{cases} a & \bar{\alpha} \geq a \\ \bar{\alpha} & \text{for } b < \bar{\alpha} < a \\ b & \bar{\alpha} \leq b \end{cases}, \quad (4)$$

where $a = \bar{\alpha}_{\text{max}}$ and $b = \bar{\alpha}_{\text{min}}$ represent the upper and lower output limits determined by the mechanical design.

In the simulation model, Eq. (3) is implemented with limited integrators for $\bar{\alpha}_P$ and $\bar{\alpha}_M$, respectively.
2.2. Mechanical subsystem

The dynamics of the output side of the hydrostatic transmission, see Fig. 2, is governed by the following equation of motion

\[ J_V \dot{\omega}_M + d_V \omega_M = \dot{V}_M \Delta p \bar{\alpha}_M - \tau_U, \]  

with the abbreviation \( J_V = J_M + J_E \). Here, \( J_M \) represents the mass moment of inertia of the hydraulic motor and \( J_E \) the one of the electric motor on the load side. The parameter \( d_V \) is the damping coefficient at the drive shaft. The driving torque of the hydraulic motor is denoted by \( \tau_M \), an unknown disturbance torque by \( \tau_U \).

2.3. Models for the decentralized control design

The overall system model comprises four first-order differential equations. For the envisaged decentralized control design, they are partitioned as follows: the first design model is linear and given by the differential equation for the normalized motor angle \( \bar{\alpha}_M \)

\[ \dot{\bar{\alpha}}_M = -\frac{1}{T_aM} \bar{\alpha}_M + \frac{k_M}{T_aM} u_M, \]  

where \( u_M \) serves as control input. Introducing the normalized pump tilt angle \( \bar{\alpha}_P \), the pressure difference \( \Delta p \), and the motor angular velocity \( \omega_M \) as state variables, the state vector of the second design model results in

\[ x = [\bar{\alpha}_P, \Delta p, \omega_M]^T. \]  

The corresponding nonlinear state-space representation \( x = f(x, u_P, \tau_d) \) becomes

\[ \begin{bmatrix} \dot{\bar{\alpha}}_P \\ \dot{\Delta p} \\ \dot{\omega}_M \end{bmatrix} = \begin{bmatrix} -\frac{1}{T_pH} \bar{\alpha}_P + \frac{k_p}{T_pH} u_P \\ \frac{2 \gamma \omega_M \omega_C}{c_H} \operatorname{sat}_1^1 (\bar{\alpha}_P) - \frac{2 \gamma \omega_M \omega_C}{c_H} \operatorname{sat}_1^1 (\bar{\alpha}_M) - \frac{\gamma \omega_C}{c_H} \\ -\frac{d_V}{F} \omega_M + \frac{\dot{V}_M}{F} \Delta p \operatorname{sat}_1^1 (\bar{\alpha}_M) - \frac{\gamma \omega_C}{F} \end{bmatrix}. \]  

The control input is given by \( u_P \), the disturbance vector by \( \tau_d = [q_U, \tau_c]^T \).

3. DECENTRALISED OPTIMAL CONTROL STRUCTURE

An optimal control approach is implemented for the tracking control of the hydrostatic transmission. The control scheme is illustrated in Fig. 3, where \( \bar{d} = [\dot{\bar{\alpha}}_P, \dot{\bar{\alpha}}_M, \dot{q}_U, \dot{\tau}_C]^T \) denotes the vector of the unmeasurable states and disturbances provided by the alternative state and disturbance estimators. An inspection of the overall system

![Figure 3. Scheme of the decentralised optimal control in combination with a state and disturbance estimator.](image-url)
\(\tilde{\alpha}_M\) and the motor angular velocity \(\omega_M\). Here, a LQR-based proportional-integral (PI) state feedback control is employed for tracking control of \(\tilde{\alpha}_M\), see Fig. 4, where the control parameters \(k_p, k_I\) and \(k\) are determined by the LQR design. The feedforward control part \(u_F\) involves a linear combination of the desired value and its time derivatives, which is chosen in such a way that the numerator and denominator polynomials of the closed-loop transfer function become identical.

The optimal control of the motor angular velocity \(\omega_M\) consists of a flatness-based feedforward control action with the desired value of the motor angular velocity \(\omega_d\) as well as its derivatives, and a feedback control action based on solutions of the state-dependent Riccati equation (SDRE). The corresponding state-dependent feedback gains are depicted in Fig. 5. A detailed derivation, simulation results and an experimental validation can be found in [15], which highlights the excellent tracking performance for both controlled variables.

\[\dot{\alpha}_{Md} \quad \text{Feedforw. control} \quad k_p \quad \text{Motor displacement unit} \quad \tau_{d} \quad u_F \quad y\]

\[\begin{align*}
\frac{1}{\tau} & \quad k_I & \quad k
\end{align*}\]

\[\begin{align*}
\tilde{\alpha}_{Md} & \quad k_{\omega,1} & \quad 0.6 & 0.7 & 0.8 & 2 \times 10^{-2}
\end{align*}\]

\[\begin{align*}
k_{\omega,2} & \quad 3.2 \times 10^{-5}
\end{align*}\]

\[\begin{align*}
k_{\omega,3} & \quad 2.4 \times 10^{-9}
\end{align*}\]

\[\begin{align*}
k_{\omega,1} \quad k_{\omega,2} \quad k_{\omega,3}
\end{align*}\]

4. STATE AND DISTURBANCE ESTIMATION

To estimate both unmeasurable system states - the normalised swashplate angle \(\tilde{\alpha}_P\) and the normalised bent axis angle \(\tilde{\alpha}_M\) - and unknown system disturbances - the leakage volume flow \(q_U\) and a disturbance torque \(\tau_U\) acting on the hydraulic motor, alternative nonlinear state and disturbance estimators are introduced in this section: a nonlinear reduced-order observer (NROO), cf. [18], an Extended Kalman Filter (EKF), cf. [19], and a Sigma-Point Kalman Filter (SPKF), cf. [20]. In combination with the optimal control, the proposed approaches are evaluated by simulation and experimental results. According to the test rig instrumentation, the pressure difference \(\Delta p\) and the motor angular velocity \(\omega_M\) can be measured by installed pressure sensors and encoders, respectively. Furthermore, in the following observer design, integrator models for the unknown disturbances are employed with the reasonable assumption that the disturbances are only slowly varying according to

\[
\tau_d = [q_U, \tau_U]^T = 0.
\]
4.1. Nonlinear reduced-order observer

For the design of a nonlinear reduced-order observer, the state equations (2) and (5) are extended with two integrators as disturbance models

\[
\dot{y}_m = f(y_m, \tau_d, x_u, u), \\
\dot{x}_u = f_u(x_u, u), \\
\tau_d = 0,
\]

where \( y_m = [\Delta p, \omega_M]^T \) contains the measurable pressure difference and the motor angular velocity. Moreover, \( x_u = [\hat{\alpha}_P, \hat{\alpha}_M]^T \) is the vector of the normalised actuator angles, \( \tau_d = [q_U, \tau_U]^T \) represents the vector of system disturbances and \( u = [u_P, u_M]^T \) denotes the vector of the control inputs. The estimated states \( \hat{x}_u \) and the disturbances \( \hat{\tau}_d \) follow from the output equation

\[
\begin{bmatrix}
\dot{\hat{x}}_u \\
\dot{\hat{\tau}}_d
\end{bmatrix} =
\begin{bmatrix}
h_{11} & 0 & h_{31} & 0 \\
0 & h_{22} & 0 & h_{42}
\end{bmatrix} 
\times
\begin{bmatrix}
y_m \\
\hat{x}_u \\
\hat{\tau}_d
\end{bmatrix}
\end{bmatrix} + 
\begin{bmatrix}
\hat{z}_1 \\
\hat{z}_2
\end{bmatrix}.
\]

where \( H \) represents the observer gain matrix. The state equations for the observer state vector \( z \) are chosen as

\[
z = \Phi(y_m, \hat{x}_u, u).
\]

The vector of nonlinear functions \( \Phi \) is determined in such a way that the steady-state observer error \( \hat{x}_t = x_t - \hat{x}_t \) converges to zero. Thus, \( \Phi \) results from the demand for a vanishing steady-state estimation error according to

\[
\dot{x}_t = 0 = x_t - H \cdot \dot{y}_m - \Phi(y_m, x_t - 0, u).
\]

Considering \( \tau_d = 0 \) and using \( x_u = f_u(x_u, u) \), (14) yields the function

\[
\Phi(y_m, x_t, u) = -H \cdot \begin{bmatrix}
\frac{2 \mu_p}{\zeta_H} - \frac{2 \mu_d}{\zeta_H} - \frac{q_U}{\zeta_H} \\
-\frac{d \mu_d}{dx_t} + \frac{2 \mu_d}{x_t} - \frac{q_M}{x_t}
\end{bmatrix} + \begin{bmatrix}
-\frac{q_p + k_p u_p}{\zeta_H} \\
-\frac{q_d + k_d u_d}{\zeta_H}
\end{bmatrix}.
\]

The linearised error dynamics \( \dot{x}_t \) has to be asymptotically stable. Therefore, all eigenvalues of the Jacobian are placed in the left complex half-plane according to

\[
\det\left(sI - \frac{\partial \Phi(y_m, x_t, u)}{\partial x_t} \right)_{x_t = \hat{x}_t} = \prod_{i=1}^4 (s + s_{Bi}).
\]

With positive values \( s_{Bi} > 0, i \in \{1, 2, 3, 4\} \), the observer gains follow directly from (16).

4.2. Extended Kalman Filter

An EKF represents a nonlinear version of the well-known Kalman Filter (KF). Here, a discrete-time version is envisaged for a combined estimation of the state variables as well as disturbances of the hydrostatic transmission. For this purpose, the state equations (6) and (7) are re-arranged and extended by two integrator disturbance models according to (8). Then, a discrete-time state-space representation can be calculated by
explicit Euler discretization as follows

\[
\begin{bmatrix}
\Delta p_{k+1} \\
\omega_{M,k+1} \\
\alpha_{p,k+1} \\
\alpha_{M,k+1} \\
q_{U,k+1} \\
\tau_{U,k+1}
\end{bmatrix}
= \begin{bmatrix}
\Delta p_k \\
\omega_{M,k} \\
\alpha_{p,k} \\
\alpha_{M,k} \\
q_{U,k} \\
\tau_{U,k}
\end{bmatrix} + Ts + \begin{bmatrix}
\frac{2\psi_\omega}{\omega_{M,k}} \alpha_{p,k} - \frac{2\psi_\omega}{\omega_{M,k}} & \frac{q_{U,k}}{\omega_{M,k}} - \frac{q_{U,k}}{\omega_{M,k}} - \frac{1}{\omega_{M,k}} \alpha_{p,k} + \frac{1}{\omega_{M,k}} p_{M,k} \\
\frac{2\psi_\omega}{\omega_{M,k}} \alpha_{M,k} & \frac{q_{U,k}}{\omega_{M,k}} & \frac{1}{\omega_{M,k}} \alpha_{M,k} + \frac{1}{\omega_{M,k}} p_{M,k} \\
0 & 0 & 0
\end{bmatrix} + \mathbf{w},
\]

Here, \(T_s\) stands for the sampling time. The vectors \(\mathbf{x}_{c,k}\) and \(\mathbf{u}_k\) represent the state and input vector at the discrete point of time \(t_k\), respectively, and the corresponding measured output vector is denoted by \(\mathbf{y}_{m,k}\). Furthermore, the process noise and the measurement noise are denoted by \(\mathbf{w}\) and \(\mathbf{v}\), respectively. Both are assumed to be zero-mean Gaussian white noise processes with zero cross-correlation. The vanishing cross-correlation leads to diagonal covariance matrices \(\mathbf{Q}\) and \(\mathbf{R}\), characterizing the process noise \(\mathbf{w}\) and the measurement noise \(\mathbf{v}\).

With the error covariance matrix \(\mathbf{P}_k\), the algorithm for the discrete-time EKF can be summarized at each time instant \(t_k\) as follows:

- State prediction
  \[
  \hat{\mathbf{x}}_{c,k+1} = \varphi_e(\hat{\mathbf{x}}_{c,k}, \mathbf{u}_k) \quad (18)
  \]

- Prediction of the error covariance matrix \(\hat{\mathbf{P}}_{k+1}\)
  \[
  \hat{\mathbf{P}}_{k+1} = \phi_c \hat{\mathbf{P}}_k \phi_c^T + \mathbf{Q}, \quad \text{with} \quad \phi_c = \frac{\partial \varphi_e(\mathbf{x}_{c,k}, \mathbf{u}_k)}{\partial \mathbf{x}_{c,k}} \bigg|_{\mathbf{x}_{c,k} = \hat{\mathbf{x}}_{c,k}} \quad (19)
  \]

- Update of the gain matrix \(\hat{L}_{k+1}\)
  \[
  \hat{L}_{k+1} = \hat{\mathbf{P}}_{k+1} \mathbf{C}^T \left( \mathbf{C} \hat{\mathbf{P}}_{k+1} \mathbf{C}^T + \mathbf{R} \right)^{-1} \quad (20)
  \]

- Update of the state vector \(\hat{\mathbf{x}}_{c,k+1}\)
  \[
  \hat{\mathbf{x}}_{c,k+1} = \hat{\mathbf{x}}_{c,k+1} + \hat{L}_{k+1} \left( \mathbf{y}_{m,k+1} - \mathbf{C} \hat{\mathbf{x}}_{c,k+1} \right) \quad (21)
  \]

- Update of the error covariance matrix for the next sampling interval
  \[
  \hat{\mathbf{P}}_{k+1} = \left( \mathbf{I} - \hat{L}_{k+1} \mathbf{C} \right) \hat{\mathbf{P}}_{k+1} \quad (22)
  \]

4.3. Sigma-Point Kalman Filter

The SPKF belongs to a class of filters that is also denoted Unscented Kalman Filters. It represents a nonlinear version of a KF that overcomes a week spot of the EKF. In contrast to an EKF, which uses a Taylor series expansion at the covariance propagation, the SPKF involves a direct nonlinear transformation to approximate the probability density distribution. The state distribution is represented by a minimal set of sampling points,
which are denoted as sigma points, cf. [20].

Considering the discrete-time representation of the hydrostatic transmission (17), a set of sigma points has to be determined. In the given implementation, $2n + 1 = 13$ sigma points $\chi_{i,k|k-1}$ have to be chosen for the extended system with a dimension of $\dim(n) = 6$. Then, the mean and the covariance of the random variables are given by

$$\hat{x}_{k|k-1} = \sum_i w_{m,i} \chi_{i,k|k-1},$$

$$P_{x,k|k-1} = \sum_i w_{c,i} ((\chi_{i,k|k-1} - \hat{x}_{k|k-1})(\chi_{i,k|k-1} - \hat{x}_{k|k-1})^T).$$

Here, $w_{m,i}$ and $w_{c,i}$ denote the corresponding weights for both state and covariance, respectively. The initial condition is described by

$$\hat{x}_{x,0} = E[\chi_{x,0}],$$

$$P_{x,x,0} = E[(\chi_{x,0} - \hat{x}_{x,0})(\chi_{x,0} - \hat{x}_{x,0})^T].$$

The sigma points for the actual state estimation are calculated according to

$$\chi_{k|1} = \left[ \hat{x}_{x,k|k-1}, \hat{x}_{x,k|k-1} + \gamma \sqrt{P_{x,x}}, \hat{x}_{x,k|k-1} - \gamma \sqrt{P_{x,x}} \right],$$

with $\gamma$ as a scaling parameter depending on the number of states. Therefore, the SPKF algorithm can be stated as:

- State prediction based on the transformation of sigma points

$$\chi_{k|k-1} = \varphi_c(\chi_{k-1|k-1}, u_{k-1}),$$

$$\hat{x}_{x,k} = \sum_{i=0}^{2n} w_{m,i} \chi_{i,k|k-1}$$

- Prediction of the error covariance matrix

$$P_{x,x,k} = \sum_{i=0}^{2n} w_{c,i} ((\chi_{i,k|k-1} - \hat{x}_{x,k})(\chi_{i,k|k-1} - \hat{x}_{x,k})^T) + Q_k$$

- Prediction of the measurements

$$Y_{k|k-1} = C_m X_{k|k-1},$$

$$\hat{y}_{k} = \sum_{i=0}^{2n} w_{m,i} Y_{i,k|k-1}$$

- Update of the gain matrix $L_k$

$$L_k = P_{x,y,k} (P_{y,y})^{-1},$$

with the cross covariance as well as the output covariance

$$P_{x,y,k} = \sum_{i=0}^{2n} w_{c,i} ((\chi_{i,k|k-1} - \hat{x}_{x,k})(Y_{i,k|k-1} - \hat{y}_{k})^T$$

$$P_{y,y,k} = \sum_{i=0}^{2n} w_{c,i} ((Y_{i,k|k-1} - \hat{y}_{k})(Y_{i,k|k-1} - \hat{y}_{k})^T + R_k$$
• Update of the state vector \( \hat{x}_k \)

\[
\hat{x}_{e,k} = \hat{x}_{e,k}^+ + L_k (y_k - \hat{y}_k^+)
\]  

(32)

• Update of the error covariance matrix for the next sampling interval

\[
P_{x,e,k} = P_{x,e,k}^- - L_k P_{y,k} L_k^T
\]

(33)

In comparison with an EKF, there is no need of calculating any Jacobians for the implementation of the SPKF. This is an advantage when Jacobians are difficult to obtain or do not exist. The closed-loop stability of the optimal control structure in combination with different state and disturbance estimators has been investigated thoroughly in a simulation analysis.

5. SIMULATION AND EXPERIMENTAL RESULTS

In this section, the proposed state and disturbance estimation techniques combined with the decentralised optimal control strategy are investigated by both simulation and experimental studies. For this purpose, the synchronised desired trajectories of the controlled variables depicted in Fig. 6 are designed offline. By exploiting the flatness property of the system at trajectory design, any saturation of the actuators can be avoided.

![Figure 6. Desired trajectories for the controlled variables.](image)

5.1. Simulation results

In order to render the simulation model more realistic and to come close to the real situation at the test rig, measurement noise of the pressure sensors as well as quantization errors of the encoders are taken into account. In the simulation studies, an unknown disturbance torque is employed according to

\[
\tau_U = 0.1 J_I \omega_{Md} + 7 \tanh \left( \frac{\omega_{Md}}{0.1} \right),
\]

(34)

whereas the for the leakage volume flow \( q_U \) a volume flow

\[
q_U = k_t \Delta p
\]

(35)

proportional to the pressure difference \( \Delta p \) is considered. Here, a constant leakage coefficient \( k_t > 0 \) is used. Fig. 7 depicts a comparison based on simulation results, where the disturbances obtained from the three alternative estimators are shown together with the simulated one. It illustrates that – in the given case – the NROO leads to the best estimation of the leakage volume flow \( q_U \), whereas both EKF and UKF provide better results.
Concerning a reconstruction of the disturbance torque $\tau_U$.

A comparison of the unmeasurable states – the normalised tilt angles $\tilde{\alpha}_P$ of the pump and $\tilde{\alpha}_M$ of the motor – is shown in Fig. 8. It can be concluded that all proposed estimators offer excellent estimation results. As a consequence, the states and disturbances provided by the estimators can be directly employed within the control structure. The tracking performance of the optimal control approach in combination with different estimators is illustrated in Fig. 9, where the tracking errors of both controlled variables is shown. The best tracking of the motor angular velocity $\omega_M$ is achieved with the NROO, whereas the combination with an SPKF leads to a superior tracking of the normalised bent axis angle $\tilde{\alpha}_M$ of the motor. The corresponding root-mean-square (RMS) errors can be found in Table 1 for a numerical comparison of the estimators.
Table 1. Root-mean-square errors for the controlled variables and the disturbance estimates.

<table>
<thead>
<tr>
<th></th>
<th>( \omega_M )</th>
<th>( \alpha_M / (1e-3) )</th>
<th>( \dot{\omega}_U / (1e-6) )</th>
<th>( \delta_U )</th>
</tr>
</thead>
<tbody>
<tr>
<td>NROO</td>
<td>0.0135 rad/s</td>
<td>0.2554</td>
<td>0.1783 m/s</td>
<td>0.0050 Nm</td>
</tr>
<tr>
<td>EKF</td>
<td>0.0195 rad/s</td>
<td>0.2943</td>
<td>0.2773 m/s</td>
<td>0.0069 Nm</td>
</tr>
<tr>
<td>SPKF</td>
<td>0.0191 rad/s</td>
<td>0.1319</td>
<td>0.3384 m/s</td>
<td>0.0063 Nm</td>
</tr>
</tbody>
</table>

5.2. Experimental results

For the implementation and validation of the estimators, all the design parameters have been chosen properly: the gain matrix \( H \) of the NROO, and the covariance matrices \( Q, R \) as well as the initial covariance matrix \( P_0 \) for both the EKF and the SPKF. They are selected in such a way that a suitable trade-off is achieved between a fast error convergence and a sufficiently smooth estimation of the states and disturbances. The experiments are carried out with a constant load torque of 30 Nm generated by the electric motor.

The comparison of the estimation results is depicted in Fig. 10 for the estimated disturbances and in Fig. 11 for the unmeasurable system states. The tracking errors of \( \omega_M \) and \( \alpha_M \) can be found in Fig. 12. The optimal control strategy allows with each of the three alternative estimators for an excellent tracking performance of the hydrostatic transmission. Among these, the best results are achieved in combination with the NROO. The other two combinations with either EKF or SPKF lead to a similar performance with slightly larger tracking errors. This may be caused by a different estimation performance concerning the system disturbances, see Fig. 10. Another obvious disadvantage of both the EKF and the SPKF is in the given case the relatively larger noise level, see Fig. 12. This is, however, not critical as confirmed by the small tracking errors, i.e., small magnitudes of the noise. Table 2 shows the RMS values of the tracking errors.
Figure 12. Tracking errors of the controlled variables using optimal control: comparison of alternative state and disturbance estimators.

Table 2. Experimental results for the root-mean-square errors of the controlled variables.

<table>
<thead>
<tr>
<th></th>
<th>NBOO</th>
<th>EKF</th>
<th>SPKF</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \omega_0 )</td>
<td>0.0797 rad/s</td>
<td>0.1637 rad/s</td>
<td>0.1418 rad/s</td>
</tr>
<tr>
<td>( \tilde{\alpha}_M / (1e^{-3}) )</td>
<td>0.0211</td>
<td>0.1938</td>
<td>0.3299</td>
</tr>
</tbody>
</table>

6. CONCLUSION AND OUTLOOK

In this paper, a comparison is presented for alternative state and disturbance estimators, which are combined with an existing optimal tracking control of a hydrostatic transmission. A nonlinear system model of a hydrostatic transmission, which is characterised by unknown disturbances, is derived. To estimate the unmeasurable states and the unknown disturbances, three alternative approaches have been investigated: a nonlinear reduced-order observer (NROO), an Extended Kalman Filter (EKF), and a Sigma-Point Kalman Filter (SPKF). The optimal tracking control structure in combination with the different estimators has been thoroughly investigated by simulations, and performed well in an experimental validation too. The comparison demonstrates that all the proposed state and disturbance estimators provide an excellent estimation quality regarding both system states and unknown disturbances. Thereby, a high accuracy is achieved by the optimal tracking control in combination with each of the estimators. In the given case, the NROO leads to a superior performance in comparison with the alternative methods. Moreover, the NROO is characterized by a simple structure and requires only a small computational effort. Both the EKF and SPKF guarantee an excellent performance as well. Due to a more complicated structure, however, the corresponding implementation effort as well as the higher computational demand may restrict the application of these methods. In the near future, a robust observer structure – a sliding mode observer – will be investigated and implemented on the test bench.

References


DESIGN STUDY AND PARAMETER OPTIMIZATION FOR A LIGHT-WEIGHT SERIES HYDRAULIC HYBRID VEHICLE

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ABSTRACT

Hydraulic hybrid drives are one potential way of improving the fuel efficiency of vehicles, including the possibility of recuperating braking energy in a hydraulic accumulator. The high power density of fluid power is mainly advantageous for heavy vehicles, or duty cycles characterized with frequent braking and acceleration. For smaller vehicles, hydraulic hybrid drives are thus most interesting under urban and suburban driving conditions. Amongst the existing architectures, the series hydraulic hybrid offers the advantage of operating the internal combustion engine independently of the vehicle speed, at the cost of a less efficient transmission path than a purely mechanical one. Previously, a series hydraulics hybrid light-duty vehicle was modelled in the transmission-line modelling (TLM)-based simulation software Hopsan from the division of Fluid and Mechatronic Systems (Flumes) at Linköping University. This paper studies through simulation-based optimization how the fuel-optimal vehicle design is affected by various mixes of urban and suburban driving requirements. Both the system’s hardware and the parameters of a basic control strategy are considered. The results show quite similar designs for most performance requirements combinations, and can be the base for further studies addressing additional requirements, conditions and objectives.

KEYWORDS: Series Hydraulic Hybrid Vehicle, Fuel Efficiency, Simulation-based Optimization

1. INTRODUCTION

Energy efficiency is a key requirement for vehicle transmissions. One area of interest is to allow for energy recuperation, which can be achieved through hybridization. For passenger vehicles, hybrid electric solutions are already available on the market, while hybrid hydraulic drivetrains are subject to on-going research and concept development. In comparison, electric solutions surpass hydraulic ones in terms of range (energy density) but offer a lower power density [1]. The higher power density makes hydraulic hybrid drives interesting for applications where heavy masses need to be moved, and where short and frequent acceleration is required.

The basic hybrid architectures each offer advantages and disadvantages (see, e.g., [2]). Series hybrids forego a mechanical power transmission path, thus leading to a lower overall efficiency due to the necessary power conversion. However, they allow for operating the internal combustion engine independently of the vehicle speed, as opposed to parallel or power-split hybrids or mechanical transmissions. Next to the architecture, the control strategy of hybrid drives is of utmost importance, defining under which conditions and in what form additional power previously stored from energy recuperation and load-levelling is to be utilized.
Research into hydraulic hybrid vehicles often addresses fuel economy improvements which can be realized by applying advanced, optimized control strategies (see for example [3]) to an existing system. In this paper, the design of a series hydraulic hybrid vehicle is studied simultaneously with a basic control strategy, similar to the works in [4] and [5]. Thus, possible interdependencies and compensation potentials can be observed. Component and control parameters are subjected to simulation-based optimization utilizing the Complex-RF method, with a focus on fuel consumption during urban driving.

2. SERIES HYDRAULIC HYBRID VEHICLE APPLICATION

2.1. Series Hydraulic Hybrid Vehicle Model

2.1.1. Components of the Series Hydraulic Hybrid Model

The series hydraulic hybrid vehicle (SHHV, Figure 1) is modelled in Hopsan, a simulation tool developed at the division of Fluid and Mechatronic Systems (Flumes) at Linköping University [6]. The model contains a one-dimensional vehicle component with final drive ratio, but no further gearing. The hydraulic machines’ losses are obtained from Rydberg's efficiency model [7] in combination with measured values, deriving the efficiencies from the machine's speed, displacement setting angle and the pressure difference over it. The accumulator losses are as well modelled to match previous measurements [8]. The fuel consumption of the diesel engine driving the pump is determined from a BSFC-map (brake-specific fuel consumption map).

![Figure 1. Basic layout of the Series Hydraulic Hybrid Vehicle (SHHV) model](image)

2.1.2. Control Strategy

The pump/motor's displacement setting is determined via a proportional-integral feedback of the vehicle velocity. Due to the series hybrid architecture, the hydraulic pump charging the system and the diesel engine driving it can be controlled independently of the vehicle reference speed. For this study, pump and diesel engine are operated based on the current state-of-charge of the accumulator in the hydraulic circuit, similar to [9]: if the state-of-charge falls below the predefined level, the system is charged until an upper boundary state-of-charge is reached. The state-of-charge values can be expressed through system pressure levels [10]. The lower control pressure limit \( p_{\text{low}} \) is then the minimum system pressure, while the margin between the upper control pressure limit \( p_{\text{high}} \) and the maximum system pressure allows for energy recuperation. During the charging period, the pump is operating as much as possible at full displacement, while the
engine’s reference speed is approximately following the path of the lowest specific fuel consumption as suggested by the BSFC-map. When the system pressure is below the lower state-of-charge boundary pressure, e.g., at system start-up, the engine is driven at maximum speed to enable fast charging.

2.2. Vehicle Application

For this study, a vehicle of the light-duty category has been selected, and it is assumed to be half-loaded (with a mass of approximately 2700kg). The focus is on urban and suburban driving, represented through the Urban Dynamometer Driving Schedule (UDDS) and, to emphasize shorter, more aggressive accelerations, the New York City Cycle (NYCC) ([11], Figure 2). Highway driving and high vehicle speeds are omitted. To ensure comparability, the accumulator is considered to be completely discharged at the beginning of driving, and when examining the system under combined requirements, the driving schedules are not run consecutively to avoid results distorted by residual accumulator charge.

![Urban Dynamometer Driving Schedule](image1.png) ![New York City Cycle](image2.png)

*Figure 2. Drive cycles used as driving requirements for hydraulic hybrid vehicle design*

3. OPTIMIZATION

The light-weight vehicle designs are supposed to be capable of following both drive cycles closely. In order to determine whether a proposed vehicle design follows the reference velocity profiles accurately, the dimension-less Average Relative Velocity Deviation (ARVD) was defined as

\[
ARVD = \frac{1}{x_{\text{max}}} \int_{t} |v_{\text{ref}}(t) - v_{\text{veh}}(t)| dt
\]

with \( v_{\text{ref}}(t) \) as the current reference velocity, \( v_{\text{veh}}(t) \) as the current vehicle velocity, and \( x_{\text{max}} \) as the total distance covered by the drive cycle.

3.1. Optimization Problem

The parameter set for the vehicle optimization includes the sizes of the hydraulic machines, the hydraulic accumulator and the diesel engine, as well as the pressure levels representing the control strategy and the maximum system pressure. The continuous ranges for all of the parameters are given in Table 1. The BSFC-map of the diesel engine is scaled linearly to the engine size.
When optimizing the hybrid hydraulic vehicle, a number of aspects are of interest. First and foremost the goal is to find a fuel-efficient setup, i.e., to minimize the fuel consumption. Due to the different characteristics of the two drive cycles employed, the fuel consumption (FC) for each drive cycle is normalized. However, the vehicle still needs to fulfill performance requirements, which are in this study set by the driving conditions represented through the standard drive cycles’ velocity profiles. For the optimization, the accuracy limits (ARVDNYCC, ARVDUDDS) are both set to 1.0 %, while further performance improvement does not reflect positively on the objective function. The variation of the main components’ sizes influences the vehicle mass, which is implemented in the model as mass functions: for the components, linear approximations were derived from mass specifications for commercially available equipment. Finally, additional conditions concerning the state-of-charge pressure levels and the system’s maximum pressure need to be considered.

The optimization problem presents itself as follows:

$$\text{min } F(x) = (FC_{NYCC}(x), FC_{UDDS}(x))^T$$

$$x = (x_1, x_2, ..., x_7)$$

$$x_i^l \leq x_i \leq x_i^u \quad \forall \; i \in \{1,2, ..., 7\}$$

$$ARVD_j(x) \leq 0.01 \quad \forall \; j \in \{NYCC, UDDS\}$$

$$x_7 < x_6$$

$$x_6 \leq x_5$$

For the optimization algorithm, the problem is reformulated to combine the normalized fuel consumption over both drive cycles using the weighted sum approach (weights $\lambda_{NYCC}$ and $\lambda_{UDDS}$), and to also include the four constraints $C_k$ (two for the accuracy, two for the pressure parameters), penalized with a factor $w_k$:

$$\text{min } F(x) = \sum_j \lambda_j \cdot \frac{FC_j(x)}{FC_{j,0}} + \sum_k C_k(x) \cdot w_k$$

$$C_k(x) = [0, 1]$$

When the design is optimized for only one of the standard drive cycles, the respective other fuel consumption weight $\lambda_j$ and accuracy penalty factor $w_k$ are set to zero in the objective function.

### 3.2. Optimization Algorithm

In this paper a modified version of the original Complex method by Box [12] is used. This is a direct search (non-gradient) method that among other things has been used for simulation-based system optimization of hydraulic systems, see for example [13]. The method is initialized with randomly created design points, and with each of the parameters sets a simulation run is performed. In every following iteration, the worst point

### Table 1. Optimization parameter ranges

<table>
<thead>
<tr>
<th>Parameter $x_i$</th>
<th>Dimension</th>
<th>Range $[x_i^l, x_i^u]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x_1$: Pump displacement $D_p$</td>
<td>$10^{-6}$ m$^3$</td>
<td>[0, 250]</td>
</tr>
<tr>
<td>$x_2$: Pump/motor displacement $D_{pm}$</td>
<td>$10^{-6}$ m$^3$</td>
<td>[75, 250]</td>
</tr>
<tr>
<td>$x_3$: Upper control pressure limit $p_{high}$</td>
<td>$10^5$ kg/(m$^2$s$^2$)</td>
<td>[15, 45]</td>
</tr>
<tr>
<td>$x_4$: Lower control pressure limit $p_{low}$</td>
<td>$10^5$ kg/(m$^2$s$^2$)</td>
<td>[12.5, 44]</td>
</tr>
<tr>
<td>$x_5$: Maximum system pressure $p_{max}$</td>
<td>$10^5$ kg/(m$^2$s$^2$)</td>
<td>[25, 45]</td>
</tr>
<tr>
<td>$x_6$: Maximum diesel torque $T_{max}$</td>
<td>kg.m$^2$/s$^2$</td>
<td>[150, 420]</td>
</tr>
<tr>
<td>$x_7$: Accumulator volume $V_{bacc}$</td>
<td>$10^{-3}$ m$^3$</td>
<td>[10, 100]</td>
</tr>
</tbody>
</table>
according to the objective function is replaced by a reflection of itself through the centroid of the remaining points. The model is then simulated and the objective function re-evaluated. The Complex-RF algorithm contains two major modifications to the Complex method: a random element is introduced for each optimization parameter which decreases in absolute value the more a parameter converges. This prevents premature convergence of the optimization into unsatisfactory solutions. A forgetting factor for evaluated parameter sets ensures that mainly recently evaluated points are considered. The method and its properties are described in further detail by Krus and Ölvander [14], where a meta-optimization is conducted for the main parameters of the Complex-RF method as well.

4. RESULTS

The optimization problem as formulated potentially has several solutions towards which the optimization can converge. This means that different parameter sets can result in similar values for the objective function and thus for the fuel consumption. In the following sections, the three best results for each optimization case are presented to illustrate the differences in the designs obtained.

4.1. Optimization Results for Individual Drive Cycle Requirements

Before addressing performance requirements combined from both driving schedules, the design is optimized for each cycle individually, but evaluated for both drive cycles (Figure 3). It can be seen that the fuel consumption varies greatly for the non-optimized drive cycle. In all cases for NYCC-optimized designs and even one for UDDS the accuracy condition is violated for the respective other velocity profile, which however is not penalized in this case. When focusing on the parameters (Figure 4), the designs found have similar pump sizes, diesel engine sizes and maximum pressure levels in common. For the inner-city drive cycle a small accumulator is sufficient, whereas with higher velocities of the UDDS a bigger component is beneficial. Most variation occurs for the size of the pump/motor and the control pressure levels. In the UDDS it is noteworthy that one parameter set has considerably smaller components (pump/motor and accumulator) and lower pressure levels than the others, while achieving similar fuel consumption. This constellation, however, is the only one that cannot fulfill the driving requirements of the NYCC, indicating a weaker performance in terms of acceleration.

![Figure 3. Fuel consumption results over both drive cycles for designs obtained from single drive cycle optimization](image-url)
4.2. Optimization Results for Combined Drive Cycle Requirements

When combining both drive cycles as performance requirements for the design optimization, both accuracy requirements from equation (2) need to be fulfilled. The fuel consumptions over the drive cycles, normalized with the Utopian values obtained through individual optimization in 4.1, are then combined in different ratios $\lambda_{NYCC} : \lambda_{UDDS}$. Thus, the extreme weighting cases represent the previously presented single drive cycle fuel consumption optimizations, but with an additional condition. With increasing emphasis on the fuel efficiency during NYCC, only little further improvement can be achieved (Figure 5) at the cost of increased fuel consumption in suburban driving. In contrast, only when the design is optimized solely focusing on the UDDS fuel consumption considerably more fuel is needed for the inner-city driving. However, when comparing the best and worst fuel consumption obtained through optimization for the combined requirements, it can be noted that the difference for the NYCC driving is higher than for the UDDS (approximately 50% vs. 24%).
For the different design parameters (Figure 6) similarly to before the smallest variation can be observed for the installed pump displacement and the maximum system pressure. A high maximum system pressure is sought as it allows for dynamic responses and wide pressure ranges while this parameter is not specifically penalized otherwise. Again, when only considering the fuel consumption in the UDDS, a noticeably bigger accumulator is suggested which aids in energy recuperation. The control pressure levels can be higher, and a smaller engine can be used. On the other hand, a big accumulator does affect the performance and fuel efficiency during NYCC driving. In cases with predominantly inner-city driving a smaller accumulator is sufficient as more additional power is not required; the pressure levels are typically lower to allow for some energy recuperation. The comparably big parameter range for the maximum system pressure in the case \( \lambda_{NYCC} : \lambda_{UDDS} = 0.3 : 0.7 \) is somewhat unexpected as all other designs converge to the upper parameter limit. This affects the upper control pressure limit, as the low values in both parameter ranges belong to the same design, thus ensuring a nonetheless adequate potential for energy recuperation.

4.3. Detailed Optimization Results for Equally Weighted, Combined Drive Cycle Requirements

Of the different cases of weighting fuel consumption over the two standard drive cycles, the equally weighted case is possibly the most interesting one. All three design points achieve similar values for the objective function (Table 2). While Figure 6 gives an idea about how far the parameters are spread, it does not contain information on the individual parameter combinations. In comparison to the other cases, the wider ranges for the pump size and the lower control pressure level stand out. In a more detailed view (Figure 7) it can be seen that in Design 2 a small pump/motor is compensated for by a higher low pressure level, i.e. minimum pressure in the system, guaranteeing a sufficiently dynamic response. For a smaller accumulator it is necessary to keep a big enough margin for energy recuperation between the upper control pressure and the maximum system pressure. As a result, the difference between the lower and upper control pressure level is smaller than for the other designs, which might account for why this design leads to the highest fuel consumption of the three presented here. Comparing Design 1 and 3, again the relationship between pump/motor displacement and lower control pressure can be observed. With the exception of the accumulator, the components of Design 3 are smaller and thus lighter than in Design 1.
Table 2. Optimized objective function $F(x)$ values for equally weighted case

<table>
<thead>
<tr>
<th>Parameter Set x</th>
<th>$F(x)$</th>
<th>$\frac{FC_{NYCC}(x)}{FC_{NYCC,0}}$</th>
<th>$\frac{FC_{UDDS}(x)}{FC_{UDDS,0}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design 1</td>
<td>1.13204</td>
<td>1.16396</td>
<td>1.10026</td>
</tr>
<tr>
<td>Design 2</td>
<td>1.15986</td>
<td>1.19211</td>
<td>1.12777</td>
</tr>
<tr>
<td>Design 3</td>
<td>1.12625</td>
<td>1.17024</td>
<td>1.08241</td>
</tr>
</tbody>
</table>

Figure 7. Parameter values (normalized) for designs obtained from combined drive cycle optimization with equal weighting (0.5 : 0.5)

4.4. Optimization Performance Index for the Series Hydraulic Hybrid Optimization Problem

For evaluating the performance of the optimization algorithm when solving a particular problem, it can be of interest to analyse how the optimization converges. As a measure, it has been suggested to apply information theory to design optimization and to put the information gained during the optimization process in relation to the number of iterations required [14]. The information gained for a design $x$, $H_x$, also referred to as information entropy, is thereby expressed as

$$H_x = -n \log_2(\varepsilon_x)$$

with $n$ as the number of parameters, and $\varepsilon_x$ as the parameter tolerance. In this case of applying the Complex-RF algorithm $\varepsilon_x$ is interpreted as the parameter’s spread between the best and worst design achieved at the end of the optimization. $H_x$ can according to this definition be determined independently of whether an optimization run is stopped because of convergence in the parameters below a pre-defined tolerance or because the maximum number of iterations previously set is reached. The performance index $\phi$ can then be calculated from the information content $H_x$ of a design $x$ and the number of iterations $N_m$ needed to obtain it:

$$\phi = \frac{H_x}{N_m}$$

Compared to a unimodal hump as presented in [14], it can be seen that the design problem for a hydraulic hybrid vehicle is considerably more complex (Table 3, for the hydraulic hybrid optimization the average is given for the best three resulting designs): the information gained per iteration is lower, as more interdependencies between the variables have to be considered. In comparing the different design optimizations, single
drive cycle optimization appear to be easier to solve, similarly when more emphasis is put on the NYCC than the UDDS fuel consumption. It is to be noted that the success rate of the optimization can be included in the definition of $H_x$, but is not considered beyond the initial limitation to the three best results achieved for each case considered.

Table 3. Optimization performance index for unimodal hump and series hydraulic hybrid vehicle optimization

<table>
<thead>
<tr>
<th>Optimization problem</th>
<th>Information entropy $H_x$</th>
<th>Number of iterations $N_m$</th>
<th>Performance index $\phi$</th>
<th>$\phi$UnimodalHump / $\phi$Hybrid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unimodal hump [14]</td>
<td>19.8</td>
<td>99</td>
<td>0.2</td>
<td>-</td>
</tr>
<tr>
<td>SHHV with Individual Drive Cycle Requirements</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>NYCC</td>
<td>81.1</td>
<td>1924</td>
<td>0.0421</td>
<td>4.75</td>
</tr>
<tr>
<td>UDDS</td>
<td>75.4</td>
<td>2668</td>
<td>0.0282</td>
<td>7.08</td>
</tr>
<tr>
<td>SHHV with Combined Drive Cycle Requirements</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\lambda_{NYCC} : \lambda_{UDDS} = 1.0 : 0.0$</td>
<td>80.5</td>
<td>2381</td>
<td>0.0338</td>
<td>5.91</td>
</tr>
<tr>
<td>$\lambda_{NYCC} : \lambda_{UDDS} = 0.7 : 0.3$</td>
<td>76.6</td>
<td>2043</td>
<td>0.0375</td>
<td>5.33</td>
</tr>
<tr>
<td>$\lambda_{NYCC} : \lambda_{UDDS} = 0.5 : 0.5$</td>
<td>75.8</td>
<td>2533</td>
<td>0.0299</td>
<td>6.69</td>
</tr>
<tr>
<td>$\lambda_{NYCC} : \lambda_{UDDS} = 0.3 : 0.7$</td>
<td>78.5</td>
<td>2802</td>
<td>0.0280</td>
<td>7.14</td>
</tr>
<tr>
<td>$\lambda_{NYCC} : \lambda_{UDDS} = 0.0 : 1.0$</td>
<td>69.2</td>
<td>2744</td>
<td>0.0252</td>
<td>7.93</td>
</tr>
</tbody>
</table>

5. DISCUSSION

The results presented in the previous section gave insight into how the design of a series hydraulic hybrid vehicle is affected by different performance requirements, with a focus on suburban and urban driving. Optimizing the design for only one of the drive cycles considered leads to unsatisfactory results: a design optimized for the NYCC has the potential to fail at completing the UDDS, while the UDDS-based designs’ fuel efficiency suffers when exposed to more aggressive driving requirements. Beyond these extreme cases, too much focus on the NYCC does neither lead to considerably lower fuel consumption during this cycle, nor does it appear desirable for a vehicle of that size. Instead, performance requirements might also need to cover highway driving and thus a higher maximum vehicle speed, as for example the case in the New European Driving Cycle, leading potentially to new design challenges.

There are some limitations to both the model and the optimization. Of the optimization parameters, the maximum system pressure is the one converging to its upper limit. Further increase would not be desirable. It could, however, be of interest to study the effect on the designs if some further restriction was imposed on this parameter. A number of parameters have not been addressed in this optimization study, especially vehicle-specific ones as well as various controller parameters other than the pressure levels corresponding to the accumulator’s state-of-charge. For the components considered here mostly linear scaling has been applied, neglecting both positive and negative effects on aspects such as weight, efficiency and operating range. Including these aspects might shift the results obtained. Lastly, it has to be pointed out that the fuel efficiency has been optimized utilizing a fairly straight-forward control strategy; even better results could be obtained when optimizing the various controls, as demonstrated in [4]. In this context, one aspect that has not been addressed yet is the accumulator’s state-of-charge upon drive cycle completion, which might account for some of the design differences.
6. CONCLUSION

In this paper, a series hydraulic hybrid vehicle model was subjected to optimization with the Complex-RF algorithm. For suburban and urban driving with varying emphasis on aggressive acceleration requirements, the fuel consumption was minimized through variation of the basic components’ sizes as well as pressure parameters corresponding to the control of the system’s charging. When both drive cycles were considered in terms of fuel consumption and fulfilling the driving requirements, different designs could be obtained with nonetheless similar performance in terms of fuel consumption. Future work could focus on the one hand on the robustness of these parameter sets, and on the other hand take cost aspects into account as well.

REFERENCES


ABSTRACT

A cut-to-length (CTL) forest harvester is used for felling, delimbing and cross-cutting trees into the dimensions and assortments required by the forest industry. Hydraulically actuated primary functions of the studied machine, such as the cross-cutting and feeding of the tree stem, are commonly known to create a highly fluctuating load for the diesel engine. In order to manage these power demands, a hydraulic hybrid concept system is implemented into a full-scale experimental machine to discover the initial functionality and to collect experimental results. This paper covers only the very first tests related to the harvester head feed function. A detailed hydraulic hybrid system configuration is presented and discussed. In conclusion, the tested system can be said to operate as planned; however, measurements show that the dynamics of a standard over-center, closed-circuit pump that is normally used in hydrostatic drive transmissions is insufficient to meet the demands of the work cycle. A direct bypass valve between the energy storage and the work hydraulics is introduced to the system in order to overcome the somewhat slow pump response. Significant hybrid power was demonstrated in a longer work cycle using pump–motor control only, but in the actual work cycle this power was clearly lower than 50 kW. The dynamical behaviour of the system is discussed further and future work, such as studies about possible ways to improve the dynamics of the pump, is proposed. Longer duration tests in actual logging work are recommended to evaluate the possible performance advantage and the fuel efficiency. The control system also needs to be tuned in order to improve the hybrid drive.

KEYWORDS: hydraulic hybrid, forest machinery, cut-to-length harvester, power management

1. INTRODUCTION

A cut-to-length (CTL) forest harvester uses a harvester head to fell, delimb and cross-cut trees into the dimensions and assortments required by the forest industry. A harvester head is carried by a hydraulic crane mounted on a terrain vehicle chassis, normally a purpose-built forestry carrier. Hydraulically actuated primary functions of the harvester head of the studied machine, such as cross-cutting, feeding of the tree stem and crane movements, are commonly known to create a highly fluctuating load on the diesel engine during mechanised logging work.

Therefore, the diesel engine for this application needs to be dimensioned for a much greater load than is usual during a work cycle. Based on earlier measurements, the average load is typically significantly less than half of the peak loads and depends on the specific conditions and trees processed. The energy needed
to cover a single power peak in a typical CTL forest harvester work cycle has rarely been found to be greater
than 100 kJ, but higher cumulative energy demands up to 1200 kJ during longer work cycles have been
measured. The above-mentioned power peak should not be a problem for a reasonably sized hydraulic
accumulator, but the higher stored energy quantity is very difficult to accommodate with hydraulic
accumulators; for these situations, the available power (e.g. the rotational speed of the diesel engine) should
also be increased before the energy storage is depleted too much. A hydraulic accumulator storing up to
300–500 kJ of hydraulic energy is proposed to serve as the energy storage device for the discussed
hydraulic hybrid drive. In this paper, the first measurements related to the harvester head feed function alone
are considered.

The preceding work included a general prestudy on various hybrid systems relevant for the application,
measurements and analysis of the actual work cycle of a CTL forest harvester, and a novel hydraulic hybrid
system was proposed [1]. This system is now studied in more detail, and the first experimental results are
discussed. A simplified diagram of the studied hydraulic hybrid system is shown in Fig. 1.

![Figure 1. Simplified presentation of the studied hydraulic hybrid system.](image)

The hydraulic hybrid system test setup consists of the diesel engine (7), driving the powertrain in general,
pump 1 (1), which is also used in a standard harvester configuration and pump 2 (2), which is introduced as
a new component needed for the hydraulic hybrid drive. Pump 1 is an open circuit, variable displacement
pump, and pump 2 is a closed-circuit, over-center design variable displacement pump, a commercial
component normally used in drive transmissions of mobile work machinery, such as CTL harvesters. The
work ports of pump 2 are connected via shut-off valves (4 and 5) from one side to the harvester head work
hydraulics (6) and from the other side to the hydraulic accumulator (3), serving as the energy storage device
of the hybrid system. In order to meet the needs of the control system, the swivel angles of both pumps 1
and 2 are measured with angle sensors (9 and 11), and the accumulator pressure and harvester head work
hydraulics pressure are measured with pressure sensors (10 and 12). Information about the state of the
diesel engine, including SAE J1939 messages as load percent and rotational speed, is acquired through a
Controller area network (CAN) bus in the engine control unit (ECU) (8).

The functionality of this hydraulic hybrid system and its circuitry have already been described in some detail,
and a semi-empirical simulation model has been used to roughly evaluate its capability to manage the power
demand peaks that exist in the application [2], [3]. This simulation also took the fuel efficiency of the
dynamically loaded diesel engine into account and shows some fuel-saving potential. The initial
dimensioning of the main components of the hybrid drive and its initial control approach has also been
proposed.
A hydraulic hybrid system similar to that proposed for a CTL forest harvester can be used in several kinds of mobile machinery where working hydraulics have a significant role and power demand in the overall work cycle. A universal implementation of such a system with similar circuitry has been presented [4].

A number of studies and prototypes of various hybrid systems in CTL forestry machines like harvesters and forwarders have recently been published. These systems include both parallel and series hybrid drives. Some of the discussed systems are also capable of recuperating otherwise lost energy in either hydraulic or electric form [5].

More generally, various hybrid systems for off-highway applications have recently been actively studied, and some products have been launched onto the market. Both hydraulic and electric hybrids are now commercially available on, for example, earth moving machines like tracked excavators and wheel loaders with energy storage consisting of hydraulic accumulators and ultra capacitors, respectively. The added cost of a hydraulic hybrid is said to be lower than the cost of an electric hybrid, which means that the return on investment (ROI) is higher with a hydraulic hybrid as the fuel savings are considered equal (25% with both technologies) [6].

This means that the payback time for a lower-cost hydraulic hybrid drive would be shorter. Based on a manufacturer’s press release, the payback time for an investment in a hydraulic hybrid option for a large tracked excavator could be as short as one year, depending on the work cycle, fuel prices, etc. [7].

2. EXPERIMENTAL TEST SETUP

To study the performance of the hydraulic hybrid system proposed earlier [1], [3], a full-scale experimental test setup (seen in Fig. 2) was assembled on an actual CTL forest harvester. The assembly of this setup directly on a real machine environment was considered reasonable because this way, the real-life load and work cycle is easier to demonstrate. In addition, the practical feasibility of the machine layout and the application can be better evaluated. This way, it is also easy to arrange the possibility of operating the machine with and without the hybrid power assist to judge the functionality directly from the operator’s station in the later phases of the experiment. This test setup assembly was implemented in a way that later makes it possible to drive longer-term fuel efficiency and performance measurements in real logging conditions. However, mounting of the hydraulic accumulator (shown in Fig. 2), for example, does not meet the requirements of serial production, considering operators’ visibility and the durability of mounting. The dimensions of the 50 l hydraulic accumulator are rather large at 230 mm in diameter, 1930 mm in height and 120 kg in weight. For a dynamic energy storage application like this, a bladder accumulator is recommended to be mounted in a vertical position, which does not make it easy to find good mounting space for it on the machine layout. A bladder design hydraulic accumulator was chosen to the application for its competitive cost level and the possibility to use the foam-filled technology.
The discussed hydraulic hybrid system is quite simply integrated to the existing standard working hydraulic system driving the harvester head. Hydraulic circuits feeding the harvester crane and drive transmission are excluded from the scope of this experimental test setup, even though technically they could also be using the same hybrid power. In practice, these circuits are driven by pumps mounted on the same pump divider gear, which naturally means that they also create a load on the same diesel engine during actual work. Stored hydraulic power discharged directly to the machine’s hydraulic systems and to assist the diesel engine to drive the powertrain, referred to as hybrid power, is discussed later in this paper.

The hydraulic accumulator, the control system of such a hydraulic hybrid drive, plays an important role when trying to quickly change the operation mode of the hybrid drive between charging and discharging the energy storage. This control system is described in more detail in section 2.2.

2.1. Hydraulic hybrid drive

The main components initially introduced to the hydraulic hybrid drive are a hydraulic accumulator, an over-center closed-circuit pump, on/off valves on both sides of the pump, two pressure sensors and two pump swivel-angle sensors. The hydraulic accumulator is connected to the hydraulic hybrid drive by a safety and shut-off block, as required by the European Pressure Equipment Directive 97/23/EC. The experimental setup was built on a standard PONSSE Ergo harvester by adding an over-center closed-circuit variable displacement pump with maximum displacement of 100 cm³. The mounting of the pump was possible using a through-drive flange on the standard swash plate design drive transmission pump already existing in the powertrain. Only minor modifications were needed to accommodate these newly introduced components on the front frame of the machine under the operator’s cabin. The hydraulic accumulator is an exception and was placed standing on the side of the engine compartment. Compared to other known hybrid drives, such as electric ones where the majority if not all the power is transmitted hydraulically, the ease of installation (apart from the large bladder accumulator) is a clear advantage of a hydraulic hybrid in this kind of application. In terms of purchasing and lifecycle costs, a hydraulic accumulator seems to be quite competitive compared to ultracapacitors that are the relevant counterparts in electric hybrid drives.
The nominal volume of the installed hydraulic accumulator is 50 l, which is somewhat larger than needed for the final application based on the earlier studies. This dimensioning is supposed to give more than sufficient accumulator capacity for the tests and the application in general [2]. For the first experimental test setup, this accumulator dimensioning was considered a good starting point.

In order to evaluate the state of charge (SOC) and energy level of the hydraulic accumulator based on accumulator pressure, a simple accumulator model assuming an adiabatic charge and discharge process was created in the control system [3]. The energy and pressure in the hydraulic accumulator can be seen in Fig. 3 and was plotted based on the accumulator model.

Figure 3. Hydraulic accumulator energy and pressure plotted from the simple adiabatic model used in the control system

With the used maximum accumulator pressure level used, 33 MPa, and the precharge pressure of 13 MPa, the theoretical available oil volume in a fully charged accumulator is approximately 24 l and stored hydraulic energy is approximately 495 kJ assuming an adiabatic process estimated for a foam-filled bladder accumulator. It was decided that the accumulator would use a foam-filled bladder technology designed to improve its performance in highly dynamic applications. The foam-filled bladder technology is said to significantly increase the energy capacity of the accumulator and to reduce the heat exchange with the environment, which leads to improved efficiency [8].

The maximum displacement of pump 1 is 193 cm³, which is the standard displacement normally available in the harvester head circuit. The maximum displacement of pump 2 is 100 cm³, which means that 100 cm³ more pump capacity is installed for the harvester head circuit. Without using the stored energy, simultaneous full flow from both pumps at the set maximum harvester head work pressure of 28 MPa could not be used as the diesel engine power would not be sufficient.

2.2. Hydraulic hybrid drive control system

On a very general level, the parallel hybrid drive studied in this paper can be used for at least two different purposes in mobile working machine applications. First, energy can be stored during lower load work phases and can be released when the diesel engine needs help to equalize the power demand during the work cycle. This kind of power peak management can be used, for example, to make the downsizing of the primary mover possible or in order to shift the operation point of the diesel engine into a more optimal area, for example, a lower rotational speed and higher but more constant torque. Both of these targets aim at improved fuel efficiency, which is becoming increasingly important in forestry and in other mobile working machine markets. Noise, vibration and exhaust gas emissions can also be reduced.
Second, a hybrid drive with available stored energy can be used as a peak power source to overcome power demands that exceed the capability of the diesel engine, even at its highest available power output. This functionality gives the highest possible performance and productivity, but does not optimise efficiency. It is also possible to use both functionalities in the same system, depending on the situation.

In this paper and experimental tests, a control approach intended to equalize the power demand is studied. This means that the overall performance of the standard machine was kept at the original level without loading the diesel engine with as high power peaks as usual.

For this first experimental test setup, a constant, predetermined load level of the diesel engine was set as a target. Typically, it is advantageous to take power from the diesel engine with a reasonably high constant load, such as 70% of the available torque or power. During the test runs in this study, a load level of 60% was used in order to better demonstrate the functionality of the system. With a higher target load level, the hybrid drive would not have been activated within the test conditions for the logs to be fed.

Initially, a hybrid power control with pump 2 and the shut-off valves only was in the scope of the experiment. The control of shut-off valves was implemented very simply based on the prevailing state of the control system state machine, ‘neutral’, ‘charge’ or ‘discharge’. The actual measured swivel angle of pump 2 was also taken into consideration. As the control system was entering the ‘charge’ state, the shut-off valve between pump 2 and the harvester head work hydraulics was opened immediately. As the system was entering the ‘discharge’ state, the shut-off valve between pump 2 and the hydraulic accumulator was opened. The other shut-off valve was closed only after the swivel angle of pump 2 was measured to have passed the zero value and a small displacement in the needed direction was ensured. In the first phase, the controller type for hydraulic power of pump 2 was chosen to be a simple P-control, which seems quite adequate for the application as the low pump dynamics were much more of a concern than pump controller behaviour.

Some other limitations were also built into the control system, such as limited pump displacements during the ‘charge’ and ‘discharge’ states when the accumulator pressure was approaching the set lower or higher limits of the accumulator pressure. In addition, functionality that starts to linearly limit the pump 2 swivel angle in the ‘discharge’ state when harvester head pressure exceeds 28 MPa and sets the swivel angle to zero at 30 MPa was implemented. This functionality was found to be necessary because of the sudden stop of feed function, especially with the slowly reacting pump 2.

Later on, as the insufficient dynamics of pump 2 were becoming more obvious, an additional bypass valve (shown in Fig. 4) was introduced to the system.

Figure 4. Simplified presentation of the experimental test setup with added bypass valve and functionality
To overcome the clear dynamics issue with pump 2, in this modified circuit an on/off 2/2 bypass valve (13) and later an adjustable throttle valve (14) were added as new components to make direct flow between the accumulator and the harvester head work hydraulics possible.

This bypass valve was then controlled based on work cycle feed forward data to give the hybrid a boost at the beginning of the feed function for of a log. With this bypass valve a simple step response test was carried out to see the capability in order to overcome the slow response of pump 2. During the first test runs, an on/off valve with high flow capacity without any throttle valve or orifice was used, and the hybrid power was not easily controllable. As oil flow from the hydraulic accumulator was very high, the pressure in the harvester head work hydraulics remained high during the whole feed function, which caused even more load on the diesel engine than without the hybrid drive. This resulted in quite large losses as excess pressure differences between the hydraulic accumulator and the harvester head work hydraulics and energy could not be reasonably used. It was not possible to adequately control the maximum allowed flow to the harvester head feed motors.

An adjustable throttle valve was introduced in series with this bypass valve, and this hybrid boost controlled by the bypass functionality was now better controllable and could be used to assist the diesel engine at the beginning of a feed function, for example. The bypass valve was controlled based on the feed forward data as a simple 300 ms step-like power boost. The duration of this step was chosen without further analysis and would be more variable and advantageous depending on the function and work cycle phase.

The described bypass functionality could also be used in order to overcome problems related to slow pump response and at the end of log feeding and the phase when the harvester head feed valve needs to be closed fairly quickly. Normally, this means that if a variable displacement pump is not quick enough to adjust to zero displacement, the excess oil flow is directed to a tank through a pressure-relief valve, and thus energy is lost. The discussed bypass valve could be used to direct this otherwise lost energy into the hydraulic accumulator, at least under some conditions. If there is a need for the bypass valve in the future, a proportional valve could be worthwhile.

As the work cycle is known to include dynamic phenomena such as step-like load changes and fluctuations, a responsive control system was needed. A control system step size of 10 ms was chosen because all related system components supported this. Next, a rapid prototyping control system was used to develop a flexible and easy-to-modify control environment. A simplified presentation of the control system architecture is shown in Fig. 5.
In Fig. 5, we can see the interaction between the drive and the control system, as well as some of the measured inputs. Feed forward data about the work cycle and currently active functions are received through the CAN bus from the higher level machine control system. This machine control system controls the harvester head functions, among other things. Inside the actual hydraulic hybrid control system we can see that some general signal pre-processing like scaling and filtering is needed, as well as the calculation of given derived signals based on measured inputs. The state machine is shown to have three different states in between which the system is switching based on inputs and a comparison of given conditions. Finally, based on the state machine higher level control, separate lower level controllers are implemented and needed outputs are generated and forwarded to the hydraulic hybrid drive HW. The control system furthermore serves as a platform for the data acquisition and visualization needed for following the hydraulic hybrid system operation as a whole and in order to make further analysis and development possible.

One of the challenges for the control system is that there can be a lot of variation between work cycles and indeed in one work cycle. The feeding of two trees equal in diameter and length can require very different power levels depending on, for example, tree branches and the interaction of the ground and the tree, as well as on the simultaneous operation of other functions of the machine. In order to make the system work well in different situations, further development of the control system and fine-tuning are needed. However, during these studies, tuning of the control system was not really possible because of time limitations.

Later in this paper, both calculated hydraulic powers and CAN bus logged engine load powers are shown. Load power on the diesel engine induced by the harvester head work hydraulics, including the studied hydraulic hybrid drive, can be defined based on the hydraulic system parameters as follows:

\[ P_{\text{load}} = P_{p1} + P_{p2}, \]  

where

- \( P_{p1} \) is load power on the diesel engine induced by pump 1 and \([\text{kW}]\)
- \( P_{p2} \) is load power on the diesel engine induced by pump 2. \([\text{kW}]\)

For the hydraulic hybrid system state ‘charge’, the equation can be written as follows:
\[ P_{\text{Dload}} = \frac{\varepsilon_{p1} V_{\text{maxp1}} n_{pdg} p_{hh}}{\eta_{HMp1}} + \frac{\varepsilon_{p2} V_{\text{maxp2}} n_{pdg} (P_{es} - P_{hh})}{\eta_{HMp2}} \]  

where

- \( \varepsilon_{p1} \) is the relative swivel angle of pump 1 (0 ... 1), [-]
- \( \varepsilon_{p2} \) is the relative swivel angle of pump 2 (-1 ... 1), [-]
- \( V_{\text{maxp1}} \) is the maximum displacement of pump 1, [m\(^3\)/rev]
- \( V_{\text{maxp2}} \) is the maximum displacement of pump 2, [m\(^3\)/rev]
- \( n_{pdg} \) is the rotational speed of the pump divider gear outputs (and pumps 1 and 2), [rev/s]
- \( \eta_{HMp1} \) is the hydromechanical efficiency of pump 1, [-]
- \( \eta_{HMp2} \) is the hydromechanical efficiency of pump 2, [-]
- \( p_{hh} \) is the pressure in the harvester head work hydraulics and [MPa]
- \( p_{es} \) is the pressure in the hydraulic accumulator. [MPa]

For the hydraulic hybrid system state 'discharge', the equation can be written as follows:

\[ P_{\text{Dload}} = \frac{\varepsilon_{p1} V_{\text{maxp1}} n_{pdg} p_{hh}}{\eta_{HMp1}} + \frac{\varepsilon_{p2} V_{\text{maxp2}} n_{pdg} (P_{es} - P_{hh})}{\eta_{HMp2}} \]  

The total hybrid power shown in Fig. 7 is calculated, neglecting the efficiencies, as follows:

\[ P_{\text{Htotal}} = Q_{es} \cdot p_{es} \]  

where

- \( P_{\text{Htotal}} \) is the total hybrid power, [kW]
- \( Q_{es} \) is the total flow rate out of the hydraulic accumulator (calculated for the bypass valve based on the used adiabatic accumulator model and for pump 2 based on measured displacement and rotational speed), [l/min]

and

\[ P_{\text{Hpc}} = Q_{p2} \cdot p_{es} \]  

where

- \( P_{\text{Hpc}} \) is the pump-controlled hybrid power, and [kW]
- \( Q_{p2} \) is the flow rate of pump 2, [l/min]

3. INITIAL TEST RESULTS

At the beginning of the first tests, pump 1 was used in parallel with pump 2 at a limited, fixed maximum swivel angle, but later (e.g. during the measurements plotted in Fig. 7) pump 1 was also actively controlled in order to be able to manage the summed flow delivered by both pumps together. In this application, pump 1 is in the standard machine configuration controlled by the higher level machine control system to reach the given pressure and maximum flow settings based on the current active harvester head functions. In the test setup, the standard control reference of pump 1's maximum swivel angle was measured and modified by introducing a new software-in-the-loop functionality. This feature compares the overall flow rate from the hydraulic accumulator to the harvester head working hydraulic system to the controlled flow of pump 1 based on the swivel angle control reference. Pump 1 is then controlled to produce only the needed amount of flow if
more flow than is currently taken from the hydraulic accumulator is needed. In potential real applications, pump 1 would be downsized or at least controlled to limited displacement.

During the first test without the bypass valve, only the maximum pump-controlled hybrid power, the power from the energy storage, was demonstrated to be as high as 100 kW. However, this was only possible with a longer preset feed length in order to give enough time for pump 2 to swivel from zero displacement to maximum positive displacement and thus release the full amount of stored energy, however this is not shown in the measurements of this paper. In addition, the bypass valve was used without the throttle valve in the beginning, and significant hybrid power was discharged, but the controllability of this flow rate and consequently the hybrid power was of course poor.

After testing the basic functionality of the control system and the whole installation on the machine in general, the step response of pump 2 was measured. Pump 2 was equipped with integrated control electronics and a CAN bus interface that allowed the reading of the pump swivel angle directly from the bus. The pump swivel angle and displacement control was implemented in side the pump controller as a closed-loop control; in other words, only the swivel angle reference was needed to be passed over to the controller through the CAN bus. The results of the step response measurement can be seen in Fig. 6.

![Pump 2 - Relative swivel angle step response](image)

**Figure 6. Pump 2 swivel angle step response**

The time needed for a full swivel angle change from the maximum negative swivel angle of -1 to the maximum positive swivel angle of +1 was measured to be approximately 1.4 s, which is far beyond what had been measured for open circuit pumps as, for example, pump 1, that is used for the harvester head in the standard machine configuration. The time needed for full swivel angle change is also more than used as an initial assumption of 1.0 s for this performance in earlier studies and simulations related to the same application [3]. It can be seen in Fig. 6 that there is a small dead band around the zero displacement, and swivel angle change speed seems to be higher towards zero displacement than toward maximum displacement. This is natural for a variable displacement pump, as system work pressure is trying to swivel the swash plate towards zero angle.

The step response of this over-centre, closed-loop pump was noticed to be a lot slower than expected, even though the pump was driven without orifices between the governor and swash plate angle adjustment cylinder chambers. To make sure the slow response was not caused by low feed pressure, the feed pressure was measured to be on the specified level and was reasonably constant all the time. Control currents for pump controller pilot valves were also measured to check that no excess delay was caused by the electric part of the governor.

Despite the clearly slow pump 2 response, it was decided to test the basic functionality of the control system with a set of test runs carried out inside the factory area rather than in the forest. These test runs were carried out with a 20 m long delimbed tree stem that had a butt end diameter of 55 cm. The preset feed
length was 5 m and the stem was processed with the automatic closed-loop feed length control normally used in actual production tree processing. This same stem and preset feed length were used for all measurements in order to standardize the conditions. Regardless of this, each test run may be slightly different considering the harvester head work pressure and therefore also engine loads. The time behaviour of work cycle phases can also be different. The diesel engine rotational speed was set to approximately 1750 rev/min, which corresponds to the pump rotational speed of approximately 1810 rev/min as the pump divider gear driving the pumps is geared up with a gear ratio of 0.966. The first test was driven without assistance from the hybrid system as the load visible on the diesel engine CAN bus was logged during feeding of the preset feed length.

Next, the same log was fed with the hybrid system activated and the pump-controlled hybrid power available to the harvester head work hydraulics system. The bypass valve and hybrid boost functionality was not used at this time. However, results from this test run are not illustrated in Fig. 7.

In the third test, the preset feed length was processed with the hybrid system active and in addition to pump-controlled hybrid power. Hybrid boost through the bypass valve was also now available to the harvester head work hydraulics during a 300 ms step in order to overcome the slow response of pump 2. The results of the measurements are shown in Fig. 7.

The measurements shown in Fig. 7 include a preset length feed of 5 m that has an overall duration of less than 2 s. At 117 kW we can see the constant target load level for the diesel engine, which is approximately 60% of the maximum engine power available at the given rotational speed. The pressure of the hydraulic accumulator and harvester head work hydraulics can be seen in the upper part of Fig. 7. At the beginning of the measurement \( t = 0 \) s, the hydraulic accumulator was fully charged to 33 MPa and it was not charged much during the tested short feed work cycle measurement.

From the beginning of the measurement and all the way to \( t = 0.8 \) s, the diesel engine load was below the target level, which means that the hybrid control system was controlling the system in the ‘charge’ state, except for the hybrid boost part at the beginning. In other words, as the control system gets the feed forward data from the harvester head control system (meaning that the feed function is started at \( t = 0 \) s), both the bypass valve and pump 2 are controlled for maximum discharge of stored energy towards the harvester head work hydraulics. This lasts 300 ms and then the bypass valve is closed and pump 2 is controlled again in a normal way. At \( t = 0.8 \) s, the diesel engine load exceeds the target level and pump-controlled hybrid

![Figure 7. Diesel engine loads, hybrid power and pressures shown during a preset length feed work cycle measurement. (Diesel engine load without hybrid is measured from a separate test run than the rest of the signals shown).](image-url)
power starts to be discharged to the system. However, the pump response is too slow to follow the control reference adequately. At the end of the log feed, the harvester head work pressure increases to over 28 MPa and the hybrid assist is controlled to zero by the hybrid control system. At \( t = 1.6 \) s, something unknown happens as pump 2 no longer seems to really try to follow the control reference. The reason for this remains unknown, as pump 2 feed pressure was not measured at this time. However, the lack of feed pressure could possibly be the reason. The highest total hybrid power discharged to the system was approximately 71 kW, and pump-controlled hybrid power was approximately 33 kW. With hybrid boost functionality implemented with the bypass valve, stored hybrid power can be discharged quickly enough. Actually, hybrid boost functionality via the bypass valve seems to be discharging stored energy into the harvester head work hydraulics roughly 0.1 s before the diesel engine is experiencing the load induced by the increasing harvester head work pressure at the beginning of the log feed.

The load power of the diesel engine was read from the diesel engine ECU via the CAN bus in order to plot the results in Fig. 7. This load power includes all load on the engine, including usual losses and potentially the load induced by hydraulic circuits other than the harvester head work hydraulics. During the measurements, harvester head feeding was the main load, as other functions were not intentionally used and the machine's drive transmission was at a standstill.

4. CONCLUSIONS

The first experimental tests carried out showed that the concept hybrid system works mainly as intended but delivers only limited performance, which is mostly related to insufficient pump 2 response. It seems quite obvious that a standard over-centre, closed-circuit driving transmission pump is not able to meet the high demands set by the aggressive work cycle of a CTL forest harvester. In order to overcome the recognized pump dynamics issues, a direct bypass valve between the energy storage and the work hydraulics was introduced to the system. This bypass valve makes it possible to react faster to the sudden load related to starting a log feed as a hybrid boost kind of functionality, but the controllability is poor. In addition, some of the known advantages related to pump-controlled hybrid power are lost when using a bypass valve. By taking advantage of feed forward data from the higher level machine control system, hybrid power through the bypass valve was available to the harvester head work hydraulics before the increased load was visible on the diesel engine's CAN bus. However, already during the first experimental tests and measurements it was possible to demonstrate a significant pump-controlled hybrid power during a longer work cycle and a pump-controlled hybrid power of less than 50 kW for an actual work cycle. Finally, it seems to be viable to continue developing the system further if a responsive enough over-centre, closed-circuit pump is available and then to continue with the measurements.

5. PROPOSED FUTURE WORK

The first experimental test setup on the proposed novel hydraulic hybrid drive has now been installed on a CTL forest harvester and the first limited test runs have been carried out. It is already obvious that the dynamics of the used over-centre, closed-circuit pump that is purely intended for vehicle drive transmission application is insufficient for use in the studied application and system configuration. One of the key questions is how to get a more responsive high pressure over-centre pump that meets the other requirements set by the system characteristics, for example, the ability to allow high pressure simultaneously on both ports of the unit. One of the current ideas is to replace or override the standard pump governor, possibly having quite limited capacity pilot valves with external fast control valves of larger flow capacity in order to get more responsive pump swivel angle control. Also, flow paths between the standard governor and the swivel angle adjustment cylinder chambers may limit the pump response; one option could be to use external pressure measurement ports found in swivel angle adjustment cylinder chambers on the pump...
housing as pilot pressure ports. By using external control, we can see the possibilities of the plain rotary unit and the swash plate actuation system.

When the pump dynamics issues have been overcome, the control system needs to be tuned to meet the requirements set by the work cycle. In addition, longer-term tests in actual logging work are needed in order to evaluate the possible performance advantage and fuel efficiency of such a hybrid machine. However, this was not possible within the scope of this study because of time limitations.

REFERENCES

GAIN SCHEDULING FULL STATE FEEDBACK WITH D-IMPLEMENTATION FOR VELOCITY TRACKING OF HYDROSTATIC DRIVE TRANSMISSION

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ABSTRACT

This paper presents a gain-scheduling based velocity controller for hydrostatic drive transmissions (HSD). We design our controller based on a model of the system which captures most of the nonlinear effects and parameter variation. Therefore, we can obtain much better performance compared to existing linear controllers. Our control strategy is based on full state feedback whose gains are scheduled on measured states which are speed and volume pressures, and estimated hydraulic flow. To implement standard state feedback, we would need to calculate operating points of all the states at all time. However, due to modelling uncertainty (specially unknown frictions) pressure equilibrium calculation will be very inaccurate. We will employ D-implementation methodology to remedy this problem.

For the proof of concept, we show the efficacy of the controller using a validated simulator of a wheel loader with real machine parameters. The experiments are performed both on flat terrain and slope. The results demonstrate that the performance of velocity tracking is high and the controllability of the machine is maintained in every situation.

KEYWORDS: Gain scheduling, state-dependent linear system, velocity control, hydraulic system, nonlinear control

1. INTRODUCTION

Gain scheduling is a very popular method for the control design of non-linear systems. Although, the concept originates from the 1960s, it was not until the 1990s when interest towards it started to increase in the academia [1], [2].

The most common procedure is to divide the problem into a number of linear sub-problems and utilize well-established linear control theory to these systems [2]. After designing a family of linear controllers, one has to somehow interpolate the gains as the operation point differs from the chosen linearization points. One method is the linear interpolation of the elements of controller transfer functions [3]. Other approaches are e.g. utilizing fuzzy logic for tuning the terms of PI-controllers [4], interpolating state feedback gain matrices based on pre-calculated ones [5] or linearizing the system about a specific trajectory [6]. It is common practice to ignore time-dependent variations in the system during the design. This is justified for slowly varying system parameters and scheduling [7], especially for linear parameter varying systems (LPV) [8].

The model we use in this work can be viewed as a state dependent parameter varying system in which the varying parameters are volumetric and hydro-mechanical efficiencies of the motors and pump of hydrostatic
drive transmission (HSD). Although, the efficiencies are functions of the states, the variation is not great and allows for the employment of gain scheduled pole placement using full state feedback. The successful implementation of state feedback requires the calculation of the operating points of all the states. These calculations are model based; thus, they are not accurate due to unknown disturbances and modelling uncertainties. A solution to this challenge is so called D-implementation developed by Kaminer et al. [9]. With a clever choice of an integral action on some of the state errors, one can guarantee zero steady-state error despite uncertainties. In the D-implementation, by placing a derivative and integral at certain points of the control loop, the need for the calculation of some of the operating points is lifted. In fact, the constant operating points vanish by derivation. Moreover, the addition of integral and derivative does not change closed loop properties of the original design.

Hydraulic systems are extremely nonlinear by nature, so controlling them with a linear controller cannot provide satisfactory performance through the entire operating range. In this paper, our aim is to design a gain scheduled velocity controller for hydrostatic drive transmissions utilizing full state feedback with D-implementation. For hydraulic systems, D-implementation means utilizing the derivatives of pressures instead of actual pressure values. This is analogous to how pressures are usually fed back in fluid power systems ([10], [11]), because otherwise the natural output disturbance rejection of these systems would be compromised [10]. The control strategy introduced in this paper for HSD (displacement control) can be easily extended to throttling types of control circuits. Other methodologies worth mentioning that address velocity tracking of HSD are model predictive control (MPC) [12], multi-mode H$_\infty$-controller [13], H$_2$ and H$_\infty$-controllers [14] as well as robust gain scheduling [15].

Reader may notice that closed loop velocity control of HSDs is not a critical functionality for manually operated machines, since the operator can always compensate for errors. However, cruise control function facilitates some tasks and guarantees the quality of work especially with inexperienced drivers. Autonomous and cooperative machines are the main motivation for this research work. Agricultural tasks that need regular speeds, combine-tractor synchronization, trucks convoying are few examples where accurate speed tracking is essential for safety and improved performance.

The paper is organized as follows. In the next section, our control design process is described in a rather general form. This is followed by the description of the example system, a simulated 3.5-ton wheel loader. In Section 4, we present simulation results that prove the concept and demonstrate the performance of the controller. In the conclusion, some ideas for further development are described.

2. CONTROLLER DESIGN

In this section, we present our control architecture and detail the design process for a rather general system model.

2.1. State-dependent linear systems

Traditional gain scheduling approaches utilize a number of linear controllers each tuned at different operating point covering entire operating range. On the contrary, the parameters of the controller devised in this paper are continuous functions of states and the controller uses state-dependent linear representation described in Equation (1). Dynamics of a hydrostatic drive system for velocity tracking control can be represented in Equation (1) below.

$$\dot{x} = A(x)x + B(x)u + d,$$  \hspace{1cm} (1)

where $x = (x_1, x_2, ..., x_n)^T$, $u = (u_1, ..., u_m)^T$ and $d = (d_1, ..., d_n)^T$ refer to state, input and disturbance vectors, respectively. In addition, $n$ and $m$ are the number of the states and inputs, respectively. In our case, $m = 1$. 
The HSD system is stable, consequently, for a constant input (operating point) \( u(t) = u_0 \) and constant disturbance \( d \), the states in steady-state will converge to a constant point, say \( x(s) = x_0 \). Thus, the pair \((x_0, u_0)\) defines the equilibrium which naturally satisfies \( A(x_0)x_0 + B(x_0)u_0 + d = 0 \). For a given \( u_0 \), state \( x_0 \) can be calculated if disturbances were known. Furthermore, accuracy of this calculation depends on the model accuracy. Notice that modelling accuracy is not as important for feedback control design as it is for steady-state calculations. It is worth mentioning that variation of \( A(x) \) and \( B(x) \) with respect to \( x \) is small in the region of interest for the HSD system. As it will be shown next, this will justify the scheduling of the control gains on the measured states.

### 2.2. Pole Placement with state feedback

We start by designing a full state feedback using pole placement techniques. For the pair \((x_0, u_0)\), the control law can be expressed by Equation (2).

\[
\begin{align*}
  u &= -K(x_0)(x - x_0) + u_0 \\
  \text{(2)}
\end{align*}
\]

Notice how the operating points contribute in the control signal. The control architecture is shown in Figure 1. Now, the closed loop system (Equation (1) together with the state feedback controller of Equation (2)) is given by Equation (3).

\[
\begin{align*}
  \dot{x} &= A(x)x - B(x)K(x_0)(x - x_0) + B(x)u_0 \equiv A_c(x)x + c \\
  \text{(3)}
\end{align*}
\]

where \( A_c(x) = A(x) - B(x)K(x_0) \) and \( c = B(x)(K(x_0)x_0 + u_0) \). State feedback \( K(x_0) \) will be designed such that the closed loop system matrix \( A_c(x) \) is stable for all operating points \( x_0 \) in the region of interest. As it is common practice in gain scheduling, the scheduling of the control gains \( K(.) \) is performed on measured states \( x \), and it can be shown that the system will remain stable if the variations are slow. In other words, we assume that extra dynamics, also known as hidden coupling, are negligible.

By equating the characteristic polynomial of the system of Equation (3) with a polynomial whose poles are located at desired locations \( \lambda_i, i = 1, \ldots, n \), we can determine the state feedback gains. This is described in Equation (4).

\[
\begin{align*}
  \det(sI - A_c(x_0)) &= \prod_{i=1}^{n} (s + \lambda_i) \\
  \text{(4)}
\end{align*}
\]

Solving Equation (4) yields the state feedback vector \( K(x_0) = [K_1(x_0) \ K_2(x_0) \ \ldots \ K_n(x_0)] \). We used the symbolic solver of Matlab to solve the equation above. The solution for our case is rather convoluted and it is not presented in here.

### 2.3. D-implementation

The D-implementation by Kaminer et al. [9] addresses the problem of gain scheduling controller implementation on nonlinear plants. It requires the same number of additional states to be added to the original system of Equation (1) as the number of inputs. Our control system is single-input single-output; therefore, we will need one extra state. This extra state is defined as an integral of a state error, by Equation (5), aimed at driving the steady-state error to zero.

\[
\dot{z} = \delta x_{n0} = x_n - x_{n0} \quad \text{(5)}
\]
Figure 1. Standard state feedback control implementation

Figure 2 illustrates D-implementation version of Figure 1. Notice how the derivatives are added to the loop right before the control gains and an integrator is added to the loop right after the gains. As explained in [9], this operation does not introduce any unstable pole-zero cancellations and the eigenvalues of the closed loop system remain unchanged.

Figure 2. State feedback with D-implementation.

Because the derivatives of the equilibrium values of the states \(x_0\) are zero, their effect on the system inputs \(u\) vanishes after D-implementation. This is a very useful result, since the utilized system model always has uncertainties and some of these equilibrium values cannot be determined accurately enough. Equilibrium control values \(u_0\) are considered as the initial condition of the integrator and their accuracy is not important. In fact, the integrator acts as an estimator, estimating the right steady-state control value. In Figure 2, only the equilibrium value \(x_{n0}\) is required. In our design, \(x_{n0}\) is speed equilibrium or equivalently the velocity reference, which is known.

However, differentiating the system outputs is not possible in practice, and in most cases the derivatives of the states cannot be measured, because they do not have a physical meaning. In real implementation, the derivatives are replaced by an approximate derivative, a causal transfer function \(s/(rs + 1)\) with small enough \(r\). [9]

3. EXAMPLE SYSTEM: WHEEL LOADER WITH HYDROSTATIC DRIVE TRANSMISSION

In this section, we describe the model of a hydrostatic drive transmission and specialise the control method presented in Section 2 to this system.
3.1. System description

The system used in this study is a 3.5-ton wheel loader with pure hydrostatic drive transmission. The hydraulic diagram of the HSD is presented in Figure 3. There is a hydraulic motor in each tyre of the machine to which the required volumetric flow is produced by a variable closed circuit hydraulic pump connected to an internal combustion engine. The direction of this flow $Q_p$, together with machine movement direction, can be changed with the displacement of HSD pump. $p_B$ and $p_A$ are the pressures of the main volumes of the HSD from which $p_B$ has higher value when the machine is moving forward.

![Hydraulic diagram of the modelled system.](image)

The control inputs (control commands) of the machine are

- the rotational speed of the engine ($n_{e,com}$).
- relative displacements of hydraulic pump ($\varepsilon_{p,com}$) and
- motors ($\varepsilon_{m,com}$).

From which the $\varepsilon_{m,com}$ has only two discrete values: 100 % and 50 %. Since the number of the inputs (3) is larger than the number of the outputs (1: the speed of the machine), there is a need for a strategy to distribute the control action among these control variables. This can for example be done using optimization [16] or based on expert knowledge and intuition [17]. In this study however, only the displacement of HSD pump is controlled while the other two are kept constant $n_{e,com} = 1500 \text{ r/min}$ and $\varepsilon_{m,com} = 100\%$.

In the middle of Figure 3, there is a flushing valve which ensures that some amount of the oil is recirculated back to the reservoir via filter. This flow, together with system leakages, is compensated by the flow of constant displacement boost pump $Q_B$. The integrated pressure relief valve limits the back pressure of hydraulic motors to a rated value.

3.2. Dynamic model of the translational motion of the machine

Equation (6) describes the equations of translational motion of the machine. The model includes viscous and coulomb frictions together with the traction forces. In deriving the model, we also assumed that the wheels do not slip.

$$n_m = \frac{b}{m} n_m + \frac{\varepsilon_m V_m}{\pi^2 r^2 m} \eta_{h,m} p_B - \frac{\varepsilon_m V_m}{\pi^2 r^2 m} \eta_{h,m} p_A - \frac{g}{2\pi r_t} \mu_{fric}$$

(6)

where

- $b$ is the viscous friction coefficient [Ns]
- $m$ is the mass of the machine [kg]
- $n_m$ is the rotational speed of the motor [1/s]
- $\varepsilon_m$ is the relative displacement of motor [-]
- $V_m$ is the displacement of motor $[m^3/r]$
The radius of the tyre [m]

η_{hm,m} is the hydromechanical efficiency of motors [-]

p_{B/A} is the pressure of volume B/A [Pa]

g is the gravitational constant [m/s^2]

μ_{fric} is the coulomb friction coefficient

The velocity of the machine is proportional to the rotational speed of the motor: \( v_{mach} = 2\pi r_t n_m \).

Equations (7) and (8) describe the pressure dynamics of volumes B and A, respectively.

\[
p_B = \frac{-\beta_{\text{eff}}}{V_B} 4\varepsilon_m V_m \frac{1}{\eta_{\text{vol,m}}} n_m - \frac{\beta_{\text{eff}}}{V_B} K_{v,fv} C_{pB} p_B + \frac{\beta_{\text{eff}}}{V_B} Q_p + Q_b C_{pB}
\]

\[
C_{pB} = \begin{cases} 
0 & p_A \leq p_B \\
1 & p_A > p_B + 0.5 \text{ MPa}
\end{cases}
\]

\[
p_A = \frac{\beta_{\text{eff}}}{V_A} 4\varepsilon_m V_m n_m - K_{v,fv} \frac{\beta_{\text{eff}}}{V_B} C_{pAP} - \frac{\beta_{\text{eff}}}{V_A} \frac{1}{\eta_{\text{vol,p}}} Q_p + Q_b C_{pA}
\]

\[
C_{pA} = \begin{cases} 
1 & p_A \leq p_B - 0.5 \text{ MPa} \\
0 & p_A > p_B
\end{cases}
\]

where

\( \beta_{\text{eff}} \) is the effective bulk modulus of the system [Pa]

\( V_B \) is the B side volume [m^3]

\( \eta_{\text{vol,m}} \) is the volumetric efficiency of motors [-]

\( K_{v,fv} \) is the flow coefficient of the flush valve [(m^3/s)/Pa]

\( C_{pB/A} \) is the coefficient defining the flush valve opening [-]

\( Q_p \) is the volumetric flow of the HSD pump [m^3/s]

\( Q_b \) is the volumetric flow of the boost pump [m^3/s]

\( \eta_{\text{vol,p}} \) is the volumetric efficiency of HSD pump [-]

A simplified first order dynamics of the flow of HSD pump is presented in Equation (9).

\[
\dot{Q}_p = -\frac{1}{\tau_p} Q_p + \frac{1}{\tau_p} Q_{p,\text{com}} \equiv -\frac{1}{\tau_p} Q_p + \frac{V_p}{\tau_p} \eta_{\text{vol,p}} u_Q
\]

where

\( \tau_p \) is the time constant of the HSD pump displacement [-]

\( u_Q \) is the control input of the system [1/s] and

\( u_Q = n_{e,\text{com}} \varepsilon_{p,\text{com}} \)

3.3. Velocity tracking control design

Equations (6) – (9) describe the model of our system of interest. The system has four states, namely, \((n_m, P_B, P_A, Q_p)\) and an input \( \varepsilon_{p,\text{com}} \) or equivalently \( u_Q \). We solve this problem in two stages. At the first stage, we assume that the flow can be controlled perfectly. To this effect, we consider the system of three states \((n_m, P_B, P_A)\) with \( Q_p \) as input, that is, the system defined by (6) – (8). To distinguish between state \( Q_p \) and control signal \( Q_{p,\text{com}} \), we will name the latter \( Q_{p,\text{com}} \). At the second stage, we will design a controller for the system defined by (9). The objective of the latter controller is to guarantee that \( Q_p \) tracks \( Q_{p,\text{com}} \) generated by the former controller. Thus, the control structure takes an inner-outer-loop structure as shown in Figure 4.

---

In this paper, we use the symbol "/" to compact the notation. In this case, B/A means B or A and p_{B/A} means p_B or p_A.
3.3.1. Control of speed with $\dot{z}$ as input, state feedback and D-implementation

In this section, we detail the design of the outer loop controller. Following the steps described in Section 2.3 for D-implementation, we define an additional integral state as follows

$$\dot{z} = n_m - n_{m,\text{ref}}$$

(11)

where $n_{m,\text{ref}} = n_m^0$, the equilibrium state. This state equation together with Equations (6) – (8) can be rewritten in a matrix form as in Equation (12), where $x = (n_m, p_B, p_A, z)$ is the state vector, $Q_{p,\text{com}}$ is the control input, and $d$ is the disturbance input.

$$\begin{bmatrix} \dot{n}_m \\ \dot{p}_B \\ \dot{p}_A \\ \dot{z} \end{bmatrix} = \begin{bmatrix} -\frac{b}{m} & \frac{\epsilon_m V_m}{\pi^2 \tau_1^2 m} \eta_{hm,m}(n_m, p_B, p_A) & -\frac{\epsilon_m V_m}{\pi^2 \tau_1^2 m} \eta_{hm,m}(n_m, p_B, p_A) & 0 \\ -\beta_{\text{eff}} & -4\epsilon_m V_m & -\frac{\beta_{\text{eff}}}{V_B} K_{v,fe} C_{pb}(p_B, p_A) & 0 \\ \beta_{\text{eff}} & \frac{4\epsilon_m V_m}{V_A} & 0 & -\frac{\beta_{\text{eff}}}{V_B} C_{pa}(p_B, p_A) \\ \frac{\beta_{\text{eff}}}{V_B} & \frac{4\epsilon_m V_m}{V_A} & -K_{v,fe} & 0 \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{\beta_{\text{eff}}}{V_B} \\ 1 \\ 0 \end{bmatrix} Q_{p,\text{com}} + d$$

(12)

where disturbance $d$ is given by

$$d = \begin{bmatrix} -\frac{g}{2\pi \tau_{\text{fric}}} \\ \frac{Q_b C_{pb}}{\eta_{\text{vol},m}(n_m, p_B, p_A)} \\ \frac{Q_b C_{pa}}{\eta_{\text{vol},m}(n_m, p_B, p_A)} \\ -n_{m,\text{ref}} \end{bmatrix}$$

Notice how the model is represented similar to (1). Notice also that since the aim is to control the velocity of the machine, the integral state is chosen as the speed error. Therefore, thanks to D-implantation, the equilibrium values of the pressures are not necessary. This increases the reliability of the controller, because the pressures are strongly related to frictions, which are the most uncertain part of the model of the machine.

The control signal is then given by

$$\dot{Q}_{p,\text{com}} = -K_1(x) \dot{n}_m - K_2(x) \dot{p}_B - K_3(x) \dot{p}_A - K_4(x)(n_m - n_{m,\text{ref}})$$

(13)

and $Q_{p,\text{com}}$ is simply integral of $\dot{Q}_{p,\text{com}}$. In other words, $Q_{p,\text{com}}$ and $\dot{Q}_{p,\text{com}}$ are outputs of the outer-loop controller which are in turn the inputs of the inner-loop controller. $K_i(x)$: $i = 1, \ldots, 4$ are designed by the pole
placement method described in Section 2.2. It is worth mentioning that the state vector $\mathbf{x}$ can be measured with high accuracy using simple and standard sensors available on modern machines.

The parameters of the model above are functions of the volumetric and hydromechanical efficiencies of the hydraulic components. In this paper, these hydraulic efficiencies are modelled as a function of pressure difference $|p_p - p_k|$, component displacements $\varepsilon_p$ and $\varepsilon_m$, and the rotational speed of the engine and motor axles $n_e$ and $n_m$, respectively. In addition, in our HSD system, the flow through the flush valve always comes from the volume that has the lower pressure and the flow of the boost pump is also directed to this volume. Also the flush valve model is simplified for the control design and the minimum 0.5 MPa pressure difference given by Equations (7) and (8) are neglected in (12).

The efficiencies of HSD pump and motors are estimated based on laboratory measurements and the data provided by the manufacturer. These values are calculated every iteration to determine control gains $\mathbf{K(x)}$ (see Equation (4)).

### 3.3.2. Control of flow $Q_p$ with $Q_{p,\text{com}}$ as input

Let $e = Q_p - Q_{p,\text{com}}$ be the error we are interested to drive to zero. It is easy to show that with the control signal

$$\varepsilon_{p,\text{com}} = \frac{1/\tau_p Q_p + \dot{Q}_{p,\text{com}} - K_5(Q_p - Q_{p,\text{com}})}{b_1},$$

where $b_1 = \frac{V_p}{\tau_p n_e \eta_{vol,p}(n_e, p_e, p_B, p_A)}$, the closed loop dynamic for $e$ is governed by $\dot{e} = -K_5 e$ that is, an exponentially fast dynamics. In the derivation above, we simply substitute the control law of Equation (14) in the flow Equation (9). To implement the controller above, we use state $x$ measurements pump displacement and Equation (15) to estimate $Q_p$

$$\dot{Q}_p = \varepsilon_p n_e V_p \eta_{vol,p}(n_e, \varepsilon_p, p_B, p_A)$$

(15)

However, for implementation purposes we use the control law below given by Equation (16).

$$\varepsilon_{p,\text{com}} = \begin{cases} 
\frac{1/\tau_p \dot{Q}_p + \dot{Q}_{p,\text{com}} - K_5 \text{sat}(\dot{Q}_p - Q_{p,\text{com}})}{b_1}, & \text{if } |v_{ref}| \geq 0.05 \frac{m}{s} \\
0, & \text{if } |v_{ref}| < 0.05 \frac{m}{s} 
\end{cases}$$

(16)

where sat() is a saturation function. In this case, the error dynamics is given by $\dot{e} = -K_5 \text{sat}(e)$, which is still stable and exponentially fast when $|e|$ is smaller that the saturation bound.

The method above is referred to as feedback linearization. We do not enter to more details in here and interested readers are invited to see [18].

### 4. SIMULATION RESULTS

#### 4.1. Description of the used simulation model

The experiments of this research have been conducted with an offline simulation model of a wheel loader. The machine is described in Section 3.1. The hydraulic and mechanical model is identical to the one validated in [16] and [19]. The main difference is that the model utilized here does not have terrain model. This means that the wheels do not slip and slopes are modelled as an additional force vector affecting the front frame of the machine. The sample time is set to 1 ms in the experiments.
The controller utilizes measured states to calculate the efficiencies and the flush flow coefficients \( (C_{pB}, C_{pA}) \) of Equation (12). Following choices were made for the controller.

- The poles of the closed loop system are all placed to \( s = -6 \) (i.e. \( \lambda_{1,2,3,4} = 6 \)) to calculate the gains \( K_{1,2,3,4}(x) \).
- The tuneable gain of Equation (16) \( K_5 = \frac{1}{\tau_p} \).
- The saturation function used in Equation (16) is saturated to 10% of the maximum \( q_p \).
- The resolution of \( \varepsilon_{p,com} \) is 0.1% (equivalent to the real machine).

4.2. Simulations

In the first simulation in Figure 5, the machine is driven on flat terrain. The velocity trajectory covers 6 different steady-state speeds in the range of 0.5 – 4.3 m/s. Response to both large and small reference changes are presented. These are generated as stepwise signals filtered with a 1st order transfer function (time constant 1 s). The graphs of the machine velocity and the displacement of HSD pump, as well as their commands, are presented in the simulation figures. Also the pressures of volumes A and B are shown.

From Figure 5, it is clear that the machine is able to follow the velocity reference with high accuracy and its controllability is preserved both during acceleration and deceleration phases. The integrative state of the D-implementation drives the steady-state error to zero as expected in Section 2.3.

Since the model is not valid for zero speed, during the first seconds of the experiment of Figure 5, the model is initialized and the state feedback gains are set to zero, i.e. \( K_{1,2,3,4}(x) = 0 \). At the beginning of acceleration (time = 5 – 5.6 s), the command of HSD pump oscillates mildly. However, this has such a minor effect on the displacement that there is not any indication of its effect on the velocity or the pressures graphs. See the magnification of the HSD pump displacement graph below. The performance of the controller is very good throughout the experiment and only minor undershoots can be detected at the end of decelerations. In steady-state \( (\dot{v}_{ref} = 0) \), \( \varepsilon_{p,com} \) varies only the amount of its resolution ± 0.1%.

![Figure 5. Simulation experiment on flat terrain.](image-url)
The only part that still needs improvement is at the end where the machine is brought to standstill. Complete stop is not reached as smoothly as e.g. reaching the 0.5 m/s-velocity at time = 51 s. Instead, this rough stop vibrates the velocity and the pressure difference changes sign which finally result in a brief 4 %-increase in $\varepsilon_{p,\text{com}}$ around time = 77.5 s. Clearly, the model of the controller is not able to capture the extreme nonlinearity of the system with $\varepsilon_p < 10\%$.

In Figure 6, the machine is driven to a hill (slope angle $\alpha = \pm 10^\circ$), which is modelled as an additional force $F_{\text{hill}} = mg \sin \alpha$. At first, the velocity reference is increased to 3 m/s and it is kept constant till the end of the experiment.

The uphill is simulated at time = 15 – 25 s and the downhill at time = 35 – 45 s. These intervals are marked in Figure 6 with vertical dotted lines and appropriate texts. The controller operates smoothly both under positive and negative disturbances. The only exception to this is the beginning of the downhill, where the velocity error increases and the displacement of HSD pump is decreased. This results in the rapid increase of $p_A$, which increases the $\varepsilon_{p,\text{com}}$ through the derivative of $p_A$ and gain $K_3(x)$. This is magnified in the HSD pump displacement graph below.

Similar problems during stopping the machine are also observed in this experiment. However, here the preceding steady-state velocity is higher than in the previous test, which results to increasing the command of HSD pump up to 8 %. Nevertheless, even the changes of this magnitude in the control signal do not have an effect on the velocity, because the machine practically stopped.

![Figure 6. Simulation experiment on a slope ($\alpha = \pm 10^\circ$).](image)

The controller compensates constant disturbances quite well, as the maximum velocity error when entering the uphill is approximately $v_{\text{err,max}} = 0.35 \text{ m/s}$ and the error of 0.05 m/s is reached in $\Delta t_{0.05} = 2.8 \text{ s}$. At the hill top, $v_{\text{err,max}} = 0.41 \text{ m/s}$ and $\Delta t_{0.05} = 3.2 \text{ s}$.

At the downhill, the effect of the disturbance is slightly larger. The same characteristic values for entering downhill are $v_{\text{err,max}} = 0.48 \text{ m/s}$ and $\Delta t_{0.05} = 4.0 \text{ s}$, and at the end of the hill $v_{\text{err,max}} = 0.54 \text{ m/s}$ and $\Delta t_{0.05} = 2.9 \text{ s}$. All these values are presented in Table 1.
5. CONCLUSION AND FUTURE WORK

In this paper, a gain scheduled full state feedback velocity controller was devised for hydrostatic drive transmission. Due to the nonlinearity of the system, a state dependent model was utilized in the calculation of the feedback gains, namely using the hydraulic efficiencies. Furthermore, D-implementation [9] was used to ensure zero steady-state error despite the uncertainties of the model and to provide adequate disturbance rejection. The system was simplified to a single-input single-output system by controlling only the displacement of a hydraulic pump.

The conducted simulations prove the concept as satisfactory velocity tracking was achieved in flat terrain in multiple operation points, and under positive and negative disturbance simulated as hills. However, some oscillation of the control signal was observed when the machine was brought to standstill. These variations were so mild that they were not visible in the velocity graph.

In the future, the presented controller will be integrated with the fuel optimal controller proposed by the authors in [16] in order to improve its velocity tracking under load. For this, the controller presented in this paper has to be modified to also consider the variable command of the engine and the reduced displacement of the hydraulic motors.

REFERENCES


Table 1. Characteristic values of hill simulation velocity errors.

<table>
<thead>
<tr>
<th></th>
<th>Uphill</th>
<th></th>
<th>Downhill</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Start</td>
<td>End</td>
<td>Start</td>
<td>End</td>
</tr>
<tr>
<td>Maximum error (v_{err,max}) [m/s]</td>
<td>0.35</td>
<td>0.41</td>
<td>0.48</td>
<td>0.54</td>
</tr>
<tr>
<td>Settling time (\Delta t_{0.95}) [s]</td>
<td>2.8</td>
<td>3.2</td>
<td>4.0</td>
<td>2.9</td>
</tr>
</tbody>
</table>


DEVELOPMENT OF A DIGITAL HYDRAULIC PUMP FOR HIGH TORQUE AND LOW SPEED APPLICATIONS IN HYDROSTATIC TRANSMISSION

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ABSTRACT

For the hydraulic drivetrain in the several renewable energy (e.g. wind, wave or tidal) applications, a kind of radial piston pump generating large torque at a low speed with flow distributor by ON/OFF valve control is introduced and its experiment has been done. General double acting cylinders are applied for the mechanism of pumping, and then the structure of this pump is simple and easy to extend for various specifications. Distribution of flow from the pumping cylinders is controlled by the ON/OFF valves, and it is possible to realize variable displacement function. To avoid flow leak and resistance on the ON/OFF controlled valve, a spool type valve with seal at the spool land is applied, so reduction of the valve operating force is compatible with non-leak characteristic. It is driven by an electric servo motor so that the valve operation is performed to exact timing. The pump that consists of these components mentioned above was built as an experiment, and the experiment which evaluates its characteristic was conducted. Its specification of hydraulic output power is 50kw by 5-cylinders, rated pressure is 20MPa and max flow-rate is 150liter/min. Finally, the pumping and variable displacement functions depending on the ON/OFF valve controls were confirmed, and the response and efficiencies in several operating conditions were estimated.

KEYWORDS: digital fluid power, digital pump, variable displacement pump, hydraulic transmission

1. INTRODUCTION

In the several renewable energy (e.g. wind, wave or tidal) applications, hydrostatic transmission is considered for its energy conversion systems [1] [2] [3]. And in order to reduce the energy loss and realize high power conversion efficiency, digital hydraulic concept is proposed [4]. In these applications, hydrostatic transmission system consists of combination of a high torque/low speed hydraulic pump and a low torque/high speed hydraulic motor. In general, when it is going to apply the digital hydraulic concept to these systems, it is stated that a fast switching valve is necessary, But if restricts to the high torque/low speed pump, it is considered that so much fast switching valve is not required. Then in this study, in the different approach from the former, how to realize a hydraulic pump with variable displacement function is proposed.
2. CONSTRUCTION OF PUMP

2.1. Plunger mechanism

Figure 1 shows the plunger mechanism for pump in this study. This pump is getting plunger movement in the same mechanism as a radial engine. But it has slight different structures from radial engine. The comparison of a radial engine and our mechanism is shown in figure 2. As a result, we can get some features described below.

Radial engine has an angle between the cylinder axis and the master rod axis. Therefor a friction is generated by pressure from piston to the cylinder wall. On the other hand, the central axis of the cylinder of this pump is always pointing the central axis of the crank pin. As the result, the piston does not press the cylinder wall, so a friction loss is not generated. And we can use a sealing system on rod side, it also becomes possible pumping in the rod side, therefor the pump became compact. The master rod of Radial engine is supported by the piston which is connected to the master rod and the crank pin. So the rotation force of the master rod transfers to the cylinder wall. On the other hand, to transfer the rotational force of the master rod to frame, the pump has a bearing to guide the rotation and translation mounted on the frame. As a result, it can be made by standard cylinders with same specification, so we were able to lower the cost of the pump. And the radial piston pump can make thinner, it is possible to increase capacity by stacking units in the axial direction with small space.

Figure 1. Plunger mechanism for pump

Figure 2. Comparison of a radial engine and our mechanism
2.2. Hydraulic circuit

In addition to the plunger mechanism, a distributor valve unit is installed for each cylinder chamber. The valve is 3-way port type and these are switched in 3-positions, suction/closed/discharge, according to the motion of cylinder for flow distributions. In order to cancel pulsations of flow, two kinds of accumulators are installed at the outlet port of pump, one is a piston type and the other is a bladder type. In general, a bladder type accumulator is restricted in the working range of pressure ratio for its durability, so it can’t work in a wide pressure range. So, different two types of accumulators are adopted, one is a piston type pre-charged in low pressure, and the other is a bladder type pre-charged in high-pressure. In the low pressure range, the piston type accumulator works but the bladder type one doesn’t work because of its pre-charging pressure level. On the other hand, in the high pressure range, the piston type accumulator doesn’t work much well but the bladder type one comes to work. So by the combination of different types of accumulators, it is able to cancel pulsations of flow in the wide range for working pressure.

2.3. Distributor valve

Figure 4 shows the structure of distributor valve unit. This is a three-way valve by spool type with seals at the spool land, and the seals work to shut off leaks in the centre closed position. On the other hand, in the each side positions, a large flow passage area can be produced to reduce pressure loss, and so much big force is not needed for valve operation. The spool is driven by an electric servo motor in the position control, so the spool switching position and its operating timing can be controlled accurately. Additionally, check-valves are installed in parallel to the three-way spool valve in order to prevent abnormal pressure in the cylinder chamber, when a compression or decompression process occurs in the situation of locked-in by the spool valve.
2.4. Working method

According to the movement of the plunger mechanism, each cylinder performs reciprocating movement and a pumping function is realized by switching the distributor valves installed with each cylinder. Each valve repeats the following process independently.

1. Suction process
   Cylinder chamber is connected to the suction side and the working fluid is supplied.

2. Compression process
   The valve is located in the closed position and pressurizes the cylinder chamber to the outlet pressure level.

3. Discharge process
   Cylinder chamber is connected to the outlet side and the working fluid is delivered to the outlet.

4. Decompression process
   The valve is located in the closed position and decompresses the cylinder chamber pressure to the suction pressure level.

In the above process, if the period of suction process is adjusted, variable displacement function is realized. To design the valve control strategy, hydraulic simulations about this process were carried out. The simulation model is shown in figure 5. It consists of all the components which construct the hydraulic circuit for pump. The movements of cylinders according to the plunger mechanism are inputted to the cylinders rods as velocities in this model. The switching responses of the distributor valves are taken into consideration.

![Simulation model](image)

**Figure 5. Simulation model**

Figure 6 shows the operating situation about certain one chamber of cylinder in the simulation, on the full or partial displacement control mode. The timing for the valve operations should be determined in consideration of its action delay, depending to the operating conditions such as pressure and drive speed.
**Figure 6. Working method**

### Full displacement control mode

- **Drive shaft rotation (°)**
- **Cylinder stroke (mm)**
- **Valve position (mm)**
- **Hydraulic pressure (MPa)**
- **Discharge flow rate (liter/min)**

### Partial displacement control mode

- **Drive shaft rotation (°)**
- **Cylinder stroke (mm)**
- **Valve position (mm)**
- **Hydraulic pressure (MPa)**
- **Discharge flow rate (liter/min)**
To function as a variable displacement pump and realize high efficiency, it is thought that each distributor valve should be operated independently in the following procedures.

- **Switch to suction process from decompression**
  In order to avoid valve leak loss, the valve should keep in the closed position until cylinder pressure falls to the suction pressure level. On the other hand, after the cylinder pressure falls to the suction pressure level, it must be opened as soon as possible in order to avoid valve pressure loss. Trigger for the valve operation is determined from pressure level in cylinder chamber so that it begins to open at the just timing which the pressure in cylinder chamber falls to the suction level.

- **Switch to compression process from suction**
  In the case of full displacement control mode, in order to avoid valve pressure loss, the valve should keep in the suction position until the expansion of cylinder chamber finishes. On the other hand, after the expansion of cylinder chamber finishes, the cylinder chamber behaviour shifts to shrinkage, so it must be switched to closed position as soon as possible in order to avoid leak loss by backflow from cylinder chamber to suction. Trigger for the valve operation is determined from rotation position of drive shaft so that it finishes traveling to the closed position at the just timing which the expansion process of cylinder chamber finishes.
  In the case of partial displacement control mode, trigger for the valve operation is determined from rotation position of drive shaft by setup for the pump displacement volume.

- **Switch to discharge process from compression**
  In order to avoid leak loss by backflow from outlet to cylinder chamber, the valve should keep in the closed position until cylinder pressure rises to outlet pressure level. On the other hand, after the cylinder pressure rises to outlet pressure level, the cylinder chamber behaviour must be shifted to discharging fluid to outlet, so it must be opened as soon as possible in order to avoid valve pressure loss. Trigger for the valve operation is determined from pressure level in cylinder chamber so that it begins to open at the just timing which the pressure in cylinder chamber raises to outlet side level.

- **Switch to decompression process from discharge**
  In order to avoid valve pressure loss, the valve should keep in the discharge position until the shrinkage of cylinder chamber finishes. On the other hand, after the shrinkage of cylinder chamber finishes, the cylinder chamber behaviour shifts to expand, so it must be switched to closed position as soon as possible in order to avoid leak loss by backflow from outlet to cylinder chamber. Trigger for the valve operation is determined from rotation position of drive shaft so that it finishes traveling to the closed position at the just timing which the shrinkage process of cylinder chamber finishes.

3. **EXPERIMENTS**

We built a testing machine for digital hydraulic pump by the components mentioned above, and the examination for a performance check was carried out.

3.1. **Test rig**

Configuration of test rig is shown in figure 7 and its specifications are shown in table 1 and 2. The pump is driven by an electric motor through reduction gear and the motor speed can be adjusted with inverter drive. The driving torque is measured by a load cell at the tip of torque arm and the driving speed and rotating position of drive shaft are detected from the signal of a rotation position encoder. The output flow-rate is measured by a flow-meter at the outlet side of pump, and the pressures at the outlet and suction are measured. To control the distributor valves and monitor operating situation, pressure transducers are also installed in the each valve unit at cylinder chamber side.
Table 2. Specification of distributor valve unit

<table>
<thead>
<tr>
<th>Feature</th>
<th>Valve spool</th>
<th>Actuation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter</td>
<td>25 mm</td>
<td>Rated power</td>
</tr>
<tr>
<td>Stroke</td>
<td>±10 mm</td>
<td>Rated force</td>
</tr>
<tr>
<td>Operation speed</td>
<td>±300 mm/s</td>
<td>Maximum speed</td>
</tr>
</tbody>
</table>
3.2. Operating situation

About each of the full or partial displacement control mode, certain operating situations of the pump are shown in figure 8 and 9.

3.2.1. Full displacement control mode

Figure 8 shows an operating situation of pump in full displacement control mode at the rated driving speed on full load. Each of the distributor valves are operated individually according to the rotating position and pressure. In the figure 8, the operation for only certain one of these valves is shown, but the other valves are also operated similarly with different phases. Ideally, the command for the servo motor to drive valve spool must be controlled so as to switch the valve position at the proper timing, but in actually, some error may occur. If a gap from the ideal timing occurs, the check valve installed with spool compensates for its gap. In figure 8, some commands for suction or discharge process lags form proper timing, but the operations of suction or discharge are working correctly because of the function of check valves.

3.2.2. Partial displacement volume control mode

Figure 9 shows an operating situation of pump in partial displacement control mode about 50% to the full displacement, and loaded outlet pressure is low. This condition must be most severe situation for the valve control and pulsations of flow, because each valve is switched when the pumping cylinder works at a maximum speed and a period of compressing pressure in cylinder chamber from suction level to outlet level is very short. Nevertheless, it’s working certainly without any serious problem. In such situations, since the valve traveling speed is slow compared with the speed of the pressure rise in the cylinder chamber, the compression process is skipped. By this operation sequence change, it is possible to avoid generating loss, by escaping quickly from the situation which the pressure loss generates when the valve places near the closed position.

3.2.3. Valve motions

Figure 10 shows response of the distributor valve. The valve travelling time is about 65msec and there is a dead time about 15msec from command to travelling start, so the total switch time lag from command to travelling end is about 80msec, but its response is stable irrespectively of load conditions and each valve has the almost same response, so it is easy to adjust the timing of valve command in consideration of the response delay. The force to operate the spool is estimated from motor torque, the maximum driving force is about 1200N including acceleration force.
Figure 8. Operating situation in full displacement control mode
Figure 9. Operating situation in partial displacement control mode
3.3. Efficiency

In this test equipment, the overall efficiency of pump was evaluated by the ratio of driving shaft power to generated hydraulic power.

\[
\text{Overall efficiency} = \frac{\text{Outlet flowrate} \times (\text{Outlet pressure} - \text{Suction pressure})}{\text{Driving torque} \times \text{Driving speed}}
\]

The volumetric efficiency was evaluated by the ratio of the volumetric capacity from cylinders strokes to outlet hydraulic flowrate.

\[
\text{Volumetric efficiency} = \frac{\text{Outlet flowrate}}{\text{Volumetric capacity} \times \text{Driving speed}}
\]

The mechanical efficiency was estimated form the overall and volumetric efficiencies. Figure 11 shows efficiencies in the full displacement control mode, and efficiencies in several displacement control modes is shown in Figure 12. Comparing with a conventional variable displacement pump, it is the same that efficiencies are worse in the partial conditions, but the worse seems to be comparatively small. It is thought that this system will be able to realize high efficiency in a comparatively wide operation range.
4. SUMMARY & CONCLUSION

Supposing the use of power conversion system for renewable energy applications, a novel digital hydraulic pump suitable for high torque/low speed uses was proposed. For the verification of its concept, we built the testing pump and trial experiments were conducted. It was verified that the testing pump functions normally without any serious problem and has the good efficiency characteristics. As a future work, we will not leave it only as a pump but studies for realizing as a motor function will be performed.

5. ACKNOWLEDGMENT

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REFERENCES


A LINEAR VALVE ACTUATED SWITCHED INERTANCE HYDRAULIC SYSTEM

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ABSTRACT

A Switched Inertance Hydraulic System (SIHS) makes use of a switching element, a hydraulic capacitance and an inertance in order to achieve theoretically lossless control of hydraulic flow and pressure. This paper reports on experimental results of a SIHS which is set up to control pressure and makes use of a novel linear valve as its switching element. The control and dynamic performance of this valve are characterised before presenting experimental results of the flow booster circuit. It was found that the SIHS worked as expected at a limited range of switching widths and frequency before a failure within the instrumentation precluded experimentation on a wider range of conditions.

KEYWORDS: Digital Hydraulics, SIHS, Digital Valve, Control

1. INTRODUCTION

The majority of hydraulic systems in the world today use throttling valves as control elements. This method is very simple and provides a smooth and infinitely variable response. However, this comes at a cost to the system efficiency. Even with component efficiencies around 90% it is not uncommon for system efficiencies to drop below 50% as a result of this throttling [1]. Given the tightening of emissions legislation and increasing cost of energy, these losses, which were once of no concern, are now firmly in the spotlight. Switched Inertance Hydraulic Systems (SIHS), which work in a similar way to switched mode power supplies in electronics, are one of the new technologies being developed to try and improve this efficiency. Suggested by Brown in 1987 [2], the SIHS has, in general, only a single active component – a fast switching valve. A fluid volume provides a capacitance effect and generally a thin pipe acts like an inductance. Figure 1 shows one possible form of the SIHS and its electrical parallel.

Figure 1. Flow booster SIHS and electrical equivalent(from [1])
In Brown's initial experiments the design of the fast switching valve proved to be the limiting factor [3]. Since Brown's work there have been many further attempts to realise a SIHS [4] [5] [6] [7] [8] [9]. In almost all cases the fast switching valve has, again, been the limitation. For example, De Negri [8] used an off-the-shelf valve which was capable of switching at 16Hz to validate a SIHS model. Based on this model, it was suggested that a switching frequency of 100Hz, or settling time of 1ms, was required to maximise efficiency. This need for fast switching valves with low pressure drops is not unique to SIHS but is common across the whole field known as digital hydraulics. It is for this reason that, when discussing the future of fluid power, Scheidl et al [10] suggest that these valves could be the same across multiple functions and industries, drastically reducing their unit cost. There is no agreed specification for this standardised valve but many designs have been published, each with slightly different specification and function [11] [12] [13] [14] [15] [16] [17]. All these valves have two things in common. Firstly, they all use a linear switching motion and secondly, they all rely upon multiple low flow elements switching synchronously to achieve a fast switching time and high flow rates. This is at odds with the only published SIHS result where the valve was able to achieve the frequencies De Negri recommended. Pan et al used a rotary valve to test a SIHS of the form seen in Figure 1 at a range of switching ratios and frequencies [7]. It was found that for a flow rate of 7L/min, an efficiency of over 83% could be achieved for all pulse widths when the High Pressure (HP) supply was held at 30bar and the Low Pressure (LP) supply at 20bar. These results are very promising and this paper seeks to emulate these results but using a linear valve instead of a rotary one. The reasons for this decision are twofold. Firstly, whilst a rotary valve is well suited to the SIHS, it is not applicable to digital hydraulics in general and the standardisation of a valve would be beneficial. Secondly, if a linear valve is position controlled then it would be possible to use it as a throttling valve at the extreme pulse widths, where the losses of the SIHS are greater than the throttling losses would be. This paper will give some details of the linear valve before proceeding to present the experimental set-up and results from the SIHS.

2. VALVE DESIGN

The design of the first iteration of the linear valve discussed herein is documented in [18]. Since then the valve's design has been iterated in order to aid with assembly and manufacturing accuracy, though the overall concept remains unchanged. Therefore this paper will only discuss the salient and novel design features of the valve. The valve is a three stage design. A Moog E050-899 servovalve directs flow into an equal area hydraulic cylinder. Together with the connecting flow galleries these form what is referred to as the actuation module, which can be seen in Figure 2.

![Figure 2. Sectioned view of actuation module](image)

This arrangement provides proportional control of the main stage instead of the bang-bang actuation used by other switching valves. However, in a similar way to other designs, the main stage is made of multiple switching elements – in this case multiple grooves on a single spool. Figure 3 shows the geometry. The main stage is a 3-way valve: it switches two inputs to a single output. The design of the spool means that at
±0.1mm the valve is fully open to either high pressure (HP) or low pressure (LP) and at 0mm is closed to both. Between ±0.1mm and ±0.5mm there should be no difference in the valve’s performance as other losses dominate.

![Figure 3. Main stage geometry](image1)

![Figure 4. Spool locations](image2)

Whilst proportional spool position control increases the complexity of the controller design task and instrumentation requirements, it offers two key advantages. The first is that it overcomes concerns about valve robustness. With operating frequencies in the hundreds of cycles a second, a day of use could equate to over six million end-stop collisions. Using position control means that these collisions are avoided, placing significantly less stress on the valve’s components. The other advantage is that De Negri’s suggested settling time of 1ms can be swapped for a switching time. Due to the spool design, as long as the spool remains between ±0.1mm and ±0.5mm it can be considered as settled. Thus any overshoot, within a reasonable margin, does not affect the switching time. With this consideration the valve is able to achieve a minimum switching time of 0.5ms as shown in the next section.

3. CONTROLLER DESIGN

The controller is a combination of a State Variable Feedback (SVF) controller and an Iterative Learning Controller (ILC). A block diagram representation of this can be seen in Figure 5.
The SVF controller was designed as a Linear Quadratic Regulator using the guidelines given in [19]. The state feedback was provided by a combination of position sensor and accelerometer from which the signals were fused to provide high bandwidth measurements of position, velocity and acceleration. The method for doing this is documented in [20]. The ILC was designed using a similar technique to that described in [21] but using a non-causal error calculation and modified stop-learning conditions. The response of the valve to a 200Hz square wave using this combined approach can be seen in Figure 6, it can be seen that a switching time of 0.5ms was achieved.

4. EXPERIMENTAL SET-UP

The experimental set-up for testing the linear valve in a SIHS can be thought of as two separate hydraulic circuits, one for the actuation stage and another for the main stage. For the main stage the entire valve is shown as a three-way, two-position switching valve. These two circuits are shown in Figure 7.
A combination of standard bladder type accumulators and inline attenuators were used to control pressure pulsations. The attenuators are effectively small bladder accumulators but instead of being branched off the flow they circumferentially surround the main flow path. This provides a more direct connection to the main flow path and provides reduction of pressure ripple over a much higher bandwidth. A standard symbol for these devices does not currently exist, therefore in Figure 7 they are depicted by an accumulator sitting on the flow path rather than being connect to it. The measurement and control was provided by a single xPC system via a National Instruments PCI6220 card. This operated at 10kHz and received inputs from the position sensor, accelerometer, three flow meters and five pressure sensors. Details of the sensors used can be found in Table 1 below.

Using this test setup it was possible to confirm whether the flow characteristics of the manufactured valve agreed with the design. The first test conducted was to ensure the switching behaviour was as intended. Figure 8 shows the flow from the HP and LP ports as the spool’s position is moved from one extreme to another and the pressure compensated, flow control valve is fully open. The sweeps lasted for 10s and all results are lo-pass filtered at 10Hz.
It is clear that the zero position mentioned above is 0.4mm with HP being effectively fully open at 0.5mm and LP at 0.3mm. It is also clear that at this zero position there is leakage to the A port. This is likely to be the result of manufacturing tolerance resulting in the spool being underlapped rather than zero lapped. It can also be seen that there is less flow from LP than from HP, which is believed to be a result of manufacturing tolerances. In order to quantify the leakage between the two ports the outlet was blocked and the position was again swept from one extreme to the other with a pressure drop of 10bar from HP to LP, as shown in Figure 9.

There is a clear hysteresis in the results. This is believed to be due to a slight lag caused by the accumulators in the circuit. Thus, a true representation of the leakage is provided by the inner curve. This agrees with the results above, showing little to no leakage past ±0.1mm. When it is considered that the valve is capable of moving from the HP to LP ports in 0.5ms with no undershoot this leakage flow is of little consequence. To confirm the design flow rate, the outlet was unblocked, the position set to 0.1mm and the pressure at the HP port increased until a drop of 10bar was registered. The flow rate at 30°C is shown in Figure 10, and is sufficient for the purposes of testing a SIHS. These initial static tests, along with the promising dynamic results, show that the valve is well suited to operating in a SIHS.
5. SWITCHED INERTANCE HYDRAULIC SYSTEM

The theory of SIHS is well documented in [3] [6] and [8]. Pan et al provide a modelling approach for SIHS which is able to account for the performance of the valve used in the SIHS [7]. The same paper also presents the concept of flow loss. This is the flow supplied from the HP port in excess of that predicted from the switching ratio.

\[ Q_{\text{loss}} = Q_{\text{HP}} - Q_{\text{out}} \cdot \alpha \]  

(1)

Where \( \alpha \) is the switching ratio and \( Q_{\text{HP}} \) and \( Q_{\text{out}} \) are the HP and output flow respectively. This flow loss is dimensionless and indicative of efficiency, it therefore provides a good performance index for SIHS circuits. Theory suggests that the ideal switching frequency for each switching ratio should have the same minimum pulse width [6] which is related to the pressure pulse reflection delay time within the Inertance tube. Thus with longer inertance tubes slower switching times can be accommodated. If losses within the valve are neglected then a simple formula can be used to find the optimal switching frequency, \( f \), using the speed of sound, \( c \), and the length of the inertance tube, \( L \) [6].

\[ f = \begin{cases} \frac{\alpha c}{2L} & 0 \leq \alpha \leq 0.5 \\ \frac{(1-\alpha)c}{2L} & 0.5 < \alpha \leq 1 \end{cases} \]  

(2)

If valve losses are included then the relationship becomes more complex, this relationship is also documented in [6]. For the purpose of this paper an inertance tube of length 1.6m and 7mm diameter was used as documented in Figure 7 and the speed of sound within the inertance tube was assumed to be 1350m/s. Figure 11 below shows the optimal switching frequency predicted from both the simple lossless model and the more complex one including valve losses along with the flow loss predicted by the more complex model. The more complex optimal switching frequency was found by using MATLAB’s inbuilt bounded optimisation routines to minimise the flow loss at each switching frequency using a model of the SIHS build upon the work of [6]. It was bounded to having a minimum pulse time of 2.5ms as this was the smallest switching time for which the control system had been validated.
Figure 11. Switching ratio against optimal switching frequency

Between 0.1 and 0.2, and 0.8 and 0.9, the optimal value is the fastest possible. This suggests that a little bit more bandwidth may be required from the controller. One assumption made in [6] is that the valve is zero-lapped, Figures 8 and 9 show that this is not the case with an overlap of around 0.15mm arising from manufacturing error. Given that the valve switches through this region of overlap in 0.5ms this should not unduly affect the models accuracy. However in order to validate the findings of the model, and also calculate the minimum flow loss of the SIHS over the entire operating range the same optimisation routine used with the model was applied to the test rig as shown in Figure 7. This sought to minimise the flow loss and by comparing the measured flow loss (which was calculated in accordance with Equation 1), and optimal frequency at which it was achieved, against those predicted it is possible to validate the model. Figure 12 shows the measured flow loss and frequency at which this was achieved for switching ratios between 0.1 and 0.45 with the resulting efficiency shown in Figure 13.

Figure 12. Experimental optimal switching frequency and corresponding flow loss
Above a switching ratio of 0.45 it was found that the flow loss greatly increased and the efficiency, correspondingly, decreased. This was believed to be a result of a slippage in the position sensors location by -0.07mm as a result of the numerous control tests conducted between the steady state and SIHS testing. This meant that there was a sizeable leakage path between HP and LP which naturally had a greater effect at high switching ratios. The results from the lower switching ratios however agree well with the model and show a good efficiency over a considerable range. To confirm that the problem did lay with a slippage in the position sensor an initial test at 100Hz and switching ratio of 0.5 was conducted with an increased valve opening.

Figure 13. Efficiency against switching ratio

Figure 14. Demand and realised position for valve at 100Hz and $\alpha=0.5$

Figure 15. Pressure at inlet and outlet of valve at 100Hz and $\alpha=0.5$
Figure 16. Pressure at inlet and outlet of SIHS at 100Hz and $\alpha=0.5$

The square wave demanded in Figure 14 can clearly be seen reflected in the outlet pressure of the valve in Figure 15 with the oscillations from the LP and HP port superimposed. These oscillations would ideally be removed from the supply lines. The oscillations are probably due to the resonance of the valve passageway from the attenuator to the valve. A difference can be seen in the LP natural frequency when the valve is open and closed. However the mean HP and LP pressure doesn’t change with the load pressure – the oscillations are at a higher frequency. The smoothing effect of the inertance tube and attenuator can clearly be seen in Figure 16, it can also be seen that the pressure at the end of the inertance tube sits almost equidistant from the mean pressures of the HP and LP lines as the switching ratio of 0.5 would imply. In order to avoid further problems with slippage in the pressure sensor the position sensor holder was remade in PVC to provide interference fit, rather than relying upon a grub screw. Sadly whilst being installed this failed within the valve preventing the flow loss experiments from being conducted with the correct zero.

6. CONCLUSION

Though it fell short of its design flow rate, the linear fast switching valve exceeded the required switching time. Its 0.5ms minimum is believed to be the fastest published. The use of position control meant that even after an estimated three million cycles there is little sign of wear, with the slip in the position sensor being the only noticeable effect.

The results from the SIHS are promising, showing the expected behaviour, and give confidence that a universally high efficiency system could be achieved. The high efficiencies and low flow loss at low switching ratios are of particular note as it is in these conditions that most hydraulic systems are at their least efficient. Sadly, this promise was not converted into results due to the untimely failure of a critical component. However, it is believed that enough has been done to prove the applicability of the valve in question to use in SIHSs.

In order to further develop the SIHS, the valve in question is in the process of being rebuilt with the intention of rerunning the tests conducted above with the aim of moving forward to use the SIHS as a control element in a larger system. In order to achieve this goal it is also necessary to develop a better means of attenuating the supply pressure pulsations.


ON EFFICIENCY OF SWITCHED INERTANCE CONTROL FOR HYDRAULIC SYSTEMS


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ABSTRACT

In hydraulic drives, throttling valves are usually used for flow and pressure control. The disadvantage of this control method is poor efficiency - typically 50% of the input energy is dissipated into heat, especially at low loads. The use of switching valves controlled by pulse-width modulation (PWM) is an efficient alternative to analog components. In combination with an inertance tube the switching valve forms a switched inertance device (SID) which can be used as a flow or pressure booster. The switching valve is a critical component of the SID. It must be fast, have low resistance and low leakage. Although research on fast switching valves has been conducted intensively over the last decade, the performance of commercially available valves is not good enough for digital hydraulics. To use valves available on the market one needs to match the valve performance to the switching requirement. The paper presents research results in the application of typical servo valves for switching control of flow in a flow booster configuration. Sizing of the hydraulic system is carried out with the objective to reach the highest possible efficiency. Mathematical modeling and simulation are performed to reveal the impact of the SID variables (duty cycle and phase delay between high and low pressure valves openings) on the system dynamics. The simulation results show that the energy efficiency of the system under switching control is higher by 10-15 % as compared to the throttle control. The mathematical model and results of the simulation are verified by experimental study.

KEYWORDS: switched inertance control device, inertance tube, pulse-width modulation, energy efficiency

1. INTRODUCTION

Throttles are widely used in modern hydraulic drives to control parameters of output component (hydraulic cylinder/motor, hydraulic actuator). Energy losses during flow throttling make up to 50 % of input power. One of the ways to increase the drive energy efficiency is to apply switching control that has been extensively developed in recent years. Application of such technologies allows to improve efficiency by tens percent. The operating principal based on fluid inertance and its advantages compared with throttle control are well described in [1, 2 and 3]. The switching technology is most prospective for those applications where high efficiency and high power density are key requirements, e.g., in mobile robotics [4].

Depending on the application, a SID can be configured in a flow or pressure boost arrangements. Its performances in both configuration has been studied in [5] (a flow booster) and in [6] (a pressure booster).
The critical component of the switched inertance hydraulic systems (SIHS) is a switching valve [7]. To achieve high efficiency the valve must be fast (time of switching <1 ms), has low resistance and low leakage. Research on fast switching valves for digital hydraulic has been conducted intensively in last decade. As result some successful designs of the valves having effective performance were proposed. A seat type valve, that switches on and off within 1ms at a flow rate of 100 lpm at 5 bar pressure drop, was developed in [8]; a spool valve, having the same switching time at 65 lpm and 10 bar respectively, was presented in 9].

The theoretical study on the SIHS operating in the mode of flow booster using Simulink software was described in [10]. In that research a hydraulic cylinder moving a weight has been used as a load. Two typical 4/3 directional valves formed control unit. To reduce pulsations at the inertance tube outlet there has been installed an oscillation dampener. Maximum efficiency gain was (10...15) % depending on load, valve operation rate and pulse duty factor. Further to that theoretical study the work [11] presented some experimental results received on a test rig developed for experimental study on switched inertance hydraulic drives. The experiments proved sufficient quality of the stand installation, its control and measurement systems but because of poor performances of the SID based on the single servo valve (MOOG 76), the proposed energy efficiency has not been achieved.

The present work studies the SIHS operating in the flow booster mode. The hydraulic system parameters have been adjusted with a goal to obtain maximum efficiency. Such parameters as valve operation rate, pulse duty factor, valve effective passage area, line sizes have been considered in the studies. Mathematical modeling and simulation has been used for studying the transient processes in the hydraulic drive under pulse-width modulation control and for its comparison with conventional throttle control.

2. THEORETICAL STUDIES OF SWITCHED INERTANCE HYDRAULIC SYSTEM

The SIHS simulation has been performed with the LMS Imagine.Lab AMESim software. The model of the systems is shown in Figure 1. Pressure in high-pressure line is equal to \( P_{HP} = 3.7 \text{ MPa} \), pressure in low-pressure line \( P_{LP} = 1.5 \text{ MPa} \). Frequency of control signal delivered to the valves is 11.6 Hz. Duty cycle of the high pressure (HP) line valve is \( D_{HP} = 50 \% \), of the low pressure (LP) line valve \( D_{LP} = 20 \% \). Delay in operation of the LP valve is \( \tau = 0.05 \text{ s} \).

![Figure 1. The switched inertance hydraulic system model in the AMESim software package:](image)

**Figure 1. The switched inertance hydraulic system model in the AMESim software package:**

HP - high pressure line (delivery), LP - low pressure line (suction), HP Valve - high pressure valve, LP Valve - low pressure valve, Ac1 - high pressure line accumulator, Ac2 - low pressure line accumulator, Ac3 - tube outlet accumulator, \( dP \) — throttle orifice pressure drop
The calculations have been made for the following fluid parameters (oil HLP-46): density \( \rho = 884 \text{ kg/m}^3 \) at \( t=30 \, ^\circ\text{C} \); sound speed \( c = 1320 \text{ m/s} \); kinematic viscosity \( \nu = 74 \text{ cSt} \) at \( t=30 \, ^\circ\text{C} \); bulk modulus \( B = 15420 \text{ bar} \); oil temperature \( t=30 \, ^\circ\text{C} \). Parameters of the valves and accumulators are described in subsection 3.1.

Figure 2 presents rectangular control signal of the HP valve and theoretical pressure response at the inertance tube inlet. Axis x shows the time and the signal phase in degrees. The rising edge of the control signal corresponds to 0° and 360°, and the falling edge corresponds to 180°.

Figure 2. Control signal of HP valve and pressure at the inertance tube inlet

In this research the duty cycle of HP valve is constant and equal to 50 %. The duty cycle and the phase delay of LP valve are variable. The duration of LP valve opening can be variable between 180° and 360°, when HP valve is closed. Maximum efficiency can be achieved if the low pressure line opens at the moment when pressure at the inertance tube inlet reaches minimum after closing HP valve. As can be seen from the Figure 2, there is a time lag between the moments of switching off HP valve and settling the minimum pressure. Soft pressure reduction at the tube inlet is due to inertance of HP valve and the system hydrodynamic processes. Consequently, at the opening LP valve it is necessary to provide some phase delay. After a while, before the opening HP valve at the inertance tube inlet the pressure begins to increase due to oscillation processes in the system. So, to avoid flow rate leakages to the suction line LP valve should be closed long before the phase 360°. As a result, the effective pulse duration of LP valve and its duty cycle are formed.

As a result of simulation Figure 3 shows variations of total flow rate, pressure at the tube inlet and outlet, as well as the valve control signals. In this simulation \( P_{HP} = 8.0 \text{ MPa} \), \( P_{LP} = 1.0 \text{ MPa} \). The total flow rate at the tube inlet consists of two components: flow rates coming from the HP and LP lines.
Figure 3. Simulated flow rates (a) and pressure (b) responses to rectangular control signals (c):

a) - flow rate out of high and low pressure lines and total flow rate at the inertance tube inlet,
b) - pressure at the inertance tube inlet and outlet;
c) - control signal of high and low pressure valves

As shown in Figure 3, the LP valve opening phase begins from the moment when the tube inlet pressure achieves minimum value. As a result, the tube inlet pressure equals to the pressure in the LP line (10 bar). The total flow rate is defined by this pressure value and by suction from the low pressure line due to inertance of the liquid column in the tube.

To compare efficiency of switched inertance control and traditional throttle control, the hydraulic system with a throttle at the inlet (meter-in system) has been modeled. Efficiency of the meter-in system has been calculated by

$$\eta = \frac{P_{\text{Load}} Q_{\text{Load}}}{P_{\text{HP}} Q_{\text{HP}}} = \frac{P_{\text{Load}}}{P_{\text{HP}}}$$

thereby assuming $Q_{\text{Load}} = Q_{\text{HP}}$, where $Q_{\text{Load}}$ and $P_{\text{Load}}$ are flow rate and pressure at the loading throttle inlet.

Simulation results for both systems are presented in Figure 4.
Figure 4 shows that energy efficiency of the SIHS operating at the constant duty cycle of LP valve 20% is larger than efficiency of the throttling meter-in hydraulic system by 5-16% at a different phase delay of the LP valve opening.

3. EXPERIMENTAL STUDIES OF SWITCHED INERTANCE HYDRAULIC SYSTEM

In order to verify theoretical results of energy efficiency of the SIHS with PWM control the experimental studies have been performed using the test rig described below.

3.1. Experimental installation

The test installation diagram for SIHD in flow booster configuration is shown in Figure 5. The main elements of the system are two valves, an inertance tube and hydraulic accumulators. Proportional valves Bosch Rexroth 4WSE2EM6 referred to below as BR25 (25 lpm) and 4WSE2ED10 - BR75 (75 lpm) have been used in the PWM mode. The inertance tube consists of two parts with length 4.8 and 3.1 m, pipe bore 8 mm. To reduce pressure oscillations the following bladder hydro-pneumatic accumulators have been used:

- Ac1 for HP line (volume 1 l, charge pressure 2.7 MPa)
- Ac2 for LP line (volume 4 l, charge pressure 0.7 MPa)
- Ac3 at the tube outlet (volume 3 l, charge pressure 0.8 MPa)
The poppet type check valve (CV) has been installed at the valve BR25 outlet to prevent back-flow through the valve when it switches from the upper to the middle position. The metering orifice with the pressure drop sensor has been set up at the inertance tube inlet to determine the instant flow rate by the indirect method. On the assumption of quasi-stationary process the instant flow rate is defined by measurement pressure drop $\Delta P$ over known orifice effective area. The adjustable throttle installed at the inertance tube outlet has been used as a load.

To reduce hydraulic resistance of the valve BR75 it has been converted into a 2/2-directional valve. An arrangement where a 4/2-directional valve is modified into a 2/2-directional valve is a good alternative, instead of increasing the size of the valve, in providing a small resistance while keeping a high dynamics for a digital control.

The idea of 4/2-directional valve conversion to 2/2-directional valve has been proposed in [12]. It was experimentally verified in [13] that this technique can be used to achieve a high flow capacity digital valve.

As a result, there is a doubled flow rate through the valve at its full open position.

The tests have been conducted with the use of the Hydac hydraulic power station in the hydraulics laboratory of the Samara State Aerospace University. The experimental unit is shown in Figure 6.
During experiments the following parameters have been registered: pressures and flow rates in the HP and LP lines, pressures at the tube inlet, midpoint and outlet, the metering orifice pressure drop at the inertance tube inlet. Pressure has been measured by the pressure pulsation sensors Vt 206 of various nominal pressures (Fig. 5): PT₁ (8 MPa), PT₂ (2.8 MPa), PT₃ (8 MPa), PT₄ (4 MPa), PT₅ (4 MPa), PT₆ (8 MPa). Static flow rate has been measured with the turbine flow meters: FT₁ and FT₂ (3-15 lpm), FT₃ (7.2-36 lpm). Signals from the pressure sensors have come to the module NI 9237, from the flow meters – to the module NI 9215. The modules have been connected to the chassis NI cRIO-9023.

For control of hydrodynamic processes, experimental data review, record and processing there has been developed a special program in NI LabVIEW. In order to measure pressure drop ΔP the sensors at the orifice inlet and outlet have been differentially connected. The experimental data have been recorded with the sampling frequency 10 kHz during 10 s with the error no more 64 byte. While signal processing, the following operations have been performed: scaling of the signals, filtration of high-frequency disturbances from the flow rate sensor, expanding them in a Fourier series for extraction of separate signal components.

3.2. Experimental study results

Natural valve frequency $f$ and damping factor $\xi$ have been determined by transient response. Parameters of the valve BR25: $f_1=196.1$ Hz, $\xi_1=0.78$. Parameters of the valve BR75: $f_2=78.4$ Hz, $\xi_2=0.78$. Flow rate though the fully open valve BR25 at pressure drop 25.5 bar is equal to 20.6 lpm, flow rate through the valve BR75 at pressure drop 7.04 bar is equal to 32.95 lpm.

Flow rate at the inertance tube inlet for the duty cycle $D_2=20\%$ and the phase delay 190.88° of LP valve is shown in Figure 7.

![Figure 7. Theoretical and experimental flow rate at the inertance tube inlet](image)

The flow rate plots for the most efficient operating mode of the system (duty cycle $D=20\%$, phase delay 190.88°) at frequency 11.6 Hz is shown in Figure 8. There is a small phase delay of pressure and flow rate relative to rectangular control signals. This is due to the additional short tubes between the high and low pressure valves installed in the test rig.
Figure 8. Experimental flow rate, pressure at the inertance tube inlet and the valve control signal for the most efficient mode

Comparison of theoretical and experimental energy efficiency of the hydraulic unit at different control duty cycle of the LP valve is shown in Figure 9.

Figure 9. Experimental and theoretical energy efficiency of a hydraulic unit at various phase delay of the LP valve control signal at different duty cycle
One can see that the most profitable mode of the system operation is realized at the duty cycle value $D_2=20\%$. In this case the efficiency comes up close to its theoretical value.

This research was conducted at a constant value of the HP valve duty cycle equal to 50\%. By variation of the HP valve switching ratio one can change the output flow in the system.

4. CONCLUSION

On the basis of the developed mathematical model of the switched inertance hydraulic system acting as a flow booster it has been proved that there is an increase of its energy efficiency in comparison with the throttle control which can reach 12-15\% depending on load, frequency and duty cycle of the low pressure valve. The developed mathematical model and experimental equipment allows evaluating the influence of the system’s and the valve parameters on the total efficiency of the switched inertance hydraulic system.

The concept of opening the LP valve at different timings to the HP valve proposed in the paper can be considered as a sound alternative to a common usage of check valves [2, 3, 4] especially of those having poor dynamics. In a case of fast acting check valves this control method gives additional opportunity to enhance efficiency. In future work the both alternatives should be compared thoroughly.

In future work it is also intended to apply an oscillation dampener reducing resonance phenomena in the system and, thus, improving the control process quality.

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ABSTRACT

Digital valve system has some unique features concerning fault tolerance. The system is able to continue operation despite a valve fault with small degradation in performance. The increased level of fault tolerance does not require any additional components and is completely software-based. Static analysis, simulations and experimental measurements show that although operation can be continued despite of a single valve fault, the tracking accuracy may suffer and the amount of needed switchings may increase. Accurate and fast fault detection and identification is very important part of fault tolerance in such system. Also the controller must be designed properly so that maximum performance can be achieved even if a valve is acting faulty. This study shows the effect of valve fault and the potential of a fault tolerant digital valve system in a high dynamic tracking control system.

KEYWORDS:    Digital hydraulics, Fault tolerance, Servo system

1. INTRODUCTION

This study focuses on fault tolerance of a high dynamic servo system with high requirements for static and dynamic accuracy. Studied application consists of a symmetric high dynamic cylinder with position feedback and pressure sensors. Such systems have traditionally been made with either servo or high performance proportional valves [1]. These high dynamic valves are fast enough for this type of application from performance point of view but they lack characteristics concerning fault tolerance. A single spool valve is a single-point-of-failure item in hydraulic servo system and therefore it must be doubled if good fault tolerance is required. Such system can be seen e.g. in aerospace applications [2] with the cost of extra components, increased space requirements, higher cost and increased complexity. This field is well studied and technology exists but it has not been utilized in traditional applications mostly due to these reasons.

Digital hydraulic Valve System (DVS) utilizes several on/off valves connected in parallel in order to achieve good controllability and fully programmable system [3]. Valves can be set to different kinds of coding schemes depending on various factors such as maximum and minimum velocity, available valve types and their characteristics. Mostly studied schemes are binary PCM and PNM systems [4, 5]. DVS has also some unique features concerning fault tolerance. It can detect faults during normal operation, with separate fault detection cycles and it can react into faults [6, 7]. On/off valves can have various faults but most usual ones seem to be
faults where valve does not open at all despite of control signal. The reason for this is that cable, connector and electronic faults usually lead into this kind of problem. Although this study focuses on such faults, DVS is also capable of reacting into faults where valve is jammed to open position [8].

Previous work on fault tolerance of digital hydraulics has focused on slower dynamic “mobile hydraulic” applications that utilize commercial valves in the range of 20...50 ms response time. Siivonen et al. published first paper on this topic that showed the basic idea of special features concerning fault tolerance in digital hydraulics [6]. The results showed that it is possible to continue operation even if one of the valves does not open despite of control signal. Further studies included fault reaction for situations where a valve is jammed to open-position [8]. This paper continues the study in the area of high dynamic tracking control systems using much faster components.

1.1. Studied system

The studied system is a high-dynamic axis with a symmetric 57/50 servo cylinder, short hydraulic transfer lines, sub two millisecond prototype 2/2 poppet valves and a constant supply pressure system. DVS is controlled with dSpace DS1103 system and feedback is taken from three pressure sensors; one for supply line and two for cylinder chambers (Bosch Rexroth sensors with voltage signal), and a linear magnetostrictive position sensor. Special features concerning this application are the high requirements for both static and dynamic performance. The maximum velocity in real application is around 0.7 m/s, the system has to be able to position itself accurately, preferably with ±1µm accuracy, and perform accurate trajectory tracking. The studied system can perform velocity tracking with 1% accuracy of maximum velocity and the Bandwidth of the test system is 65 Hz with ±2mm Sinusoidal input (<3 dB magnitude). The natural frequency of the system is approximately 110 Hz. The studied system has only a small inertia and no load on it. The hydraulic circuit diagram is shown in fig 1.

![Hydraulic circuit diagram](image)

*Figure 1. Hydraulic circuit of digital flow control unit (left) and the test system*

The valve flow rates are set with orifices to match approximately a binary PCM coding scheme. Fig 2 shows the assumed pQ curves of each valve-orifice-block combination. The shown pQ curves are based on measurements in a couple of operation points and do not take into account possible cavitation choking. The given curves are used by the valve model of the controller. Small deviations between components is typical for orifice-valve combinations and is caused by manufacturing tolerances, position in valve block and orientation of the valve in cavity.
1.2. Control method

A conventional model-based digital valve controller is utilized in this study. Basic functionality and working principle was developed by Linjama et al. [9] and it has been tailored for the studied application. Basic control principle is shown in fig 3. The modifications included a new and computationally fast valve model [10], performance optimized parameters and a simplified driving mode selection algorithm due to lack of flow regeneration capabilities [11].

The controller is a closed-loop type and utilizes three pressure sensors to measure supply pressure and both chamber pressures and a position sensor. This information is used to calculate the load force of the actuator and then calculate steady-state velocity and chamber pressures by using a valve model and simplified cylinder model. This calculation is done for a subset of all possible opening combinations \( U \), so called search space. The studied DVS consists of 5 valves in four Digital Flow Control Units (DFCU) and this leads to \( 2^{20} = 1 \, 048 \, 576 \) different opening combinations and this is limited to a subset of 16 most promising ones (4 combinations for PA + AT and 4 for PB + BT) with correct velocity direction and rough estimate of flow rates. The best candidate is selected by utilizing a cost function with possibility for tuning the system to avoid e.g. velocity error, pressure level error or number of valve switchings. The outer part of the controller is a motion controller.
that uses upper level velocity and position references \( (v_{ref}, x_{ref}) \) and modifies these signals with two gains; feed forward gain for velocity reference and P-gain for position error signal.

1.3. Controller modifications to make the system fault tolerant

Some small modifications are needed in order to utilize fault reaction capabilities of DVS. If a faulty valve is detected and identified, it must be “deactivated”. The DVS controller is designed to utilize this information so the solution is to remove the faulty component from search space completely. A couple of ways are available for this task and probably the easiest one is to change the valve parameters in such way that the valve is never selected for use. In this case the nominal flow rate of the valve is multiplied with very big number. Since the resulting valve will never fulfill controller tracking requirements it is therefore not available anymore. This method, however, requires some information of the controller architecture because in some applications and with a very small pressure difference the resulting flow rate could still be reasonable. In this studied system the minimum pressure difference is so high that this will never happen.

Other possible ways to deactivate a valve are e.g. using NaN or inf to define valve characteristics or to remove the corresponding valve completely from the matrix of all possible opening combinations \((U)\). The first suggestion may be vulnerable with some compilers or microcontrollers that do not support the use of such data types and the second method requires slightly bigger changes to controller code.

1.4. Fault detection and identification with DVS

Fault detection and identification (FDI) is important part of fault tolerance. Siivonen et al studied the topic with DVS and developed two approaches where fault is detected during normal operation by utilizing current and voltage measurements [12] and a method that utilized a separate fault detection cycle that studies various faults when system is off-line [13]. These methods lead into accurate FDI and could be utilized also in high dynamic servo applications.

Other ways to detect valve faults include e.g. model based fault detection that has been widely studied. An example of this approach is a nonlinear model studied by Tan et al. [14]. Another way to detect and identify a valve fault accurately is to utilize position measurement of valve poppet. LVDT sensors are commonly used as position feedback with spool valves and this technology could also be utilized with on/off valves [15].

2. EFFECT OF FAULT IN HIGH DYNAMIC DIGITAL VALVE SYSTEM

2.1. Passive fault accommodation of studied system

If the system is simplified into one flow edge, a single DFCU, the effect of a fault is very clear. If a faulty acting valve is commanded open and it stays closed, the total opening of that DFCU is either exactly correct or too small depending on the control signal. The magnitude of the faulty flow rate depends on the nominal flow rate of the corresponding valve and the pressure difference over it. Eq 1 shows how to calculate the flow rate \( Q \) of one DFCU with \( i \) valves with given flow rate coefficient \( K_v \) over a pressure difference \( dp \). The power-of-\( x \) function is used instead of theoretical square root equation because it is more accurate when compared to real \( pQ \) curves of valves. In faults where a valve stays open despite of control signal the same equation applies.

\[
Q = \sum_i K_v(i) \, dp^{x(i)}
\]  

In a complete digital valve system the combined effect of all DFCUs, chamber pressures and piston velocity must be taken into account. Too small flow rate in a DFCU will decrease the flow rate in or out of the cylinder chamber and depending on the load it will affect the piston velocity and corresponding chamber pressure. If the system is expected to be in steady state situation both before and after the fault becomes effective, the
Magnitude can be calculated if some assumption are made. Deriving analytical equation, the cross-flow from pump to tank is neglected although it is possible to be used in order to improve control accuracy. Also the tank pressure is assumed zero but it could also be some other constant number. In practice the tank pressure is seldom measured because it has only a small effect on control performance and it’s usually relatively constant. The following equations use 0.5 as the power of pressure difference in order to keep the study in general level.

The basic equations of hydraulics determine the dependence between chamber pressures, cylinder areas, valve parameters and piston velocity as presented in equations 2-5. Movement direction is assumed as extending and load to be force zero. The same equations also apply to faults where a valve is open despite of control signal.

\[ p_B = p_A \frac{A_1}{A_2} \]

\[ v = \frac{Q_{PA}}{A_1} = \frac{Q_{BT}}{A_2} \]

\[ v = \frac{K_v,PA\sqrt{PP-PA}}{A_1} \]

\[ v = \frac{K_v,BT\sqrt{PA}}{A_2} \]

Although the external load force is assumed to be zero, it could just as well be some other constant and known value.

\[ F = p_A A_1 - p_B A_2 - F_{Load} \]

These equations have only two unknown values which are chamber pressure \( p_A \) and piston velocity \( v \) and they can be solved analytically to following equations 7 and 8.

\[ p_A = \frac{A_2^3K_v,PA^2PP}{A_1^2K_v,BT^2 + A_2^2K_v,PA^2} \]

\[ v = \frac{K_v,PA\sqrt{PP-PA}}{A_1} \]

These equations can be used to calculate chamber pressure and piston velocity with any given valve opening and used to study the effect of valve fault into pressure and velocity error in steady-state conditions. Equations for retracting direction and differential connection can be calculated in similar way.

According to these equations an example can be made with a 57/50 symmetric cylinder, no external load and five binary coded valves [1:2:4:8:16] in each DFCU dimensioned for ~0.7 m/s max velocity. The example in Fig 4 uses 10 MPa constant supply pressure and it also shows typical states marked in blue and divided with red dashed lines. These states are used in normal operation in order to track chamber pressure references and avoid chamber pressure fluctuations. The system without any faults can achieve any of the "velocity – chamber pressure" –pair while a system with a fault disables some of the combinations leading to reduced search space. If the controller is operating in area that has disappeared due to the fault, the closest point is automatically selected instead and the difference is the error caused by the fault.

Smallest valves (a valve with smallest orifice) create a disturbance and bigger valves (valves with bigger orifice) removes complete section. On the other hand a fault in big valve only makes a limitation to some area while a fault in the smallest valve hinders the controllability in all pressure and velocity regions.

If a fault is unknown, the effect when activating a faulty valve is same as setting wrong control signal. From single DFCU point-of-view the flow rate changes depending on what was the original opening and what is the new. The relative error is smallest when a small valve is faulty and total opening of DFCU is close to maximum. E.g. with 5 parallel valves the two biggest states 30 and 31 are exactly equal if the smallest valve is not opening at all. According to equations 7 and 8 the biggest possible change in velocity occurs when the biggest valve suddenly closes during run and the pressure difference is big. With maximum positive velocity a fault in valve PA5 decreases velocity approximately 0.25 m/s which is roughly 40%. The other difference is that the A
chamber pressure changes from 5 MPa to 1.9 MPa because the system requires more pressure drop over the remaining four valves in the DFCU\textsubscript{PA}. There is hardly anything the controller can do in this phase if supply pressure can’t be controlled. Also, a closed loop controller is never at its best with close to maximum velocity because the possibilities are limited and therefore it’s more interesting to study situations where system is more flexible.

![Figure 4. Velocity vs A chamber pressure with and without DFCU\textsubscript{PA} faults in extending direction](image)

If a slower velocity region is studied the behavior of closed-loop controller is clearer. As an example with velocity reference of 0.35 m/s and A chamber pressure reference of 9 MPa the control signals could be for example [1 1 1 1] in DFCU\textsubscript{PA} and [0 1 1 0 1] in DFCU\textsubscript{BT}. The actual velocity would be 0.351 m/s and A chamber pressure 8.6 MPa. Fault in valve PA5 will cause the DFCU\textsubscript{PA} to be in state [1 1 1 1 0]. The resulting velocity would be 0.259 m/s and chamber pressure would be 7.5 MPa.

2.2. Dynamic effect of valve fault

This chapter studies effect of a valve fault on system dynamics. Basically if a fault is introduced to the system while the corresponding valve is used, it acts in same way as a deliberate step would be made to control signal from one state to another. The result is a transient between two states that will cause pressure change and velocity change depending on inertia, flow rates, pressure levels, hydraulic capacitances and change rate of the disturbance. If the fault is known, the system can avoid using the valve with decreased control performance and otherwise the performance is hindered by dynamic effects of suddenly choosing a faulty state.

The behavior of the system depends strongly on the initial velocity and pressure level because the DVS control accuracy is limited by resolution. An example of dynamic behavior of studied system with a valve PA3 fault is shown in Fig 5. The cyan colored line shows the time when fault is introduced to the simulation model. System has constant restricting load of 3 kN in this example. Left plot shows system when fault is unknown and in right
plot the controller knows about the fault and disables the valve. In left plot the closed loop controller tries to fix the increased error by selecting other states and then it return to faulty state every time. This control shows as oscillation in both velocity and pressure levels. The simulation uses constant pressure source so there is no supply line dynamics involved which might also worsen the situation. In right side plot the controller reacts into fault in correct way and removes the faulty valve from search space. This leads to some disturbance in both velocity and pressure levels but shortly a steady state is found. The disturbance is caused by sudden step in control signal and is practically unavoidable. It must also be noted that the shown example is one of the worst cases that was found during the study. However it seems important that the fault is detected and identified so that the controller can react into the fault.

![Figure 5. Closed-loop simulations with fault in valve PA3 at time 0.2s – Controller doesn’t know (left) and when controller knows and compensates the fault (right)](image)

3. MEASUREMENTS

Fault tolerance of the studied DVS is tested with faults in all valves so that the fault is either known or unknown. The faults were caused by disabling the valve from software so that the controller knows about the fault and so that the controller does not know about the fault. This study uses no sensors to detect faults during operation but suitable on-line detection method could be e.g. current measurement.

Fig 6 shows an example where system is run without any faults versus a system with a fault in valve PA5. The trajectory is trapezoidal with short acceleration, deceleration and some amount of constant velocity region. Typical position tracking accuracy during constant velocity phase is roughly ±200 µm. The pressure levels are controlled with a mathematical pressure compensator that tries to maintain relatively high pressure levels at all times. Also it must be noted that the valves used in this test were prototypes with somewhat problematic delay variations that cause some peaks when changing states. The test run with a fault PA5 shows the effect of fault on tracking performance. Position error reaches levels of 2-3 mm, pressure levels drop to 2 MPa region and dynamic behavior is very bad. The difference is very similar as simulations showed in Fig 5. Also it is notable that when the faulty acting valve is not used, the system performance is similar to the normal run. If
the fault is known and compensated the performance is close to normal run. Some differences can be seen in pressure behavior during the faster extending movement due to lack of flow capacity (DFCU\textsubscript{PA} control signal has saturated to 15 [1 1 1 1 0]).

![Figure 6. Comparison run without faults and run with unknown fault in valve PA3](image)

Fig 7 shows another example from measurements where the studied valve is PB1. The smallest of the DFCU\textsubscript{PB} valves is not causing such big problems during trajectory tracking because of small flow capacity of the valve but it causes a different type of problem. If the fault is not known the system is unable to stop in two occasions. This is happening because the smallest valves are almost every time used in last part of positioning. When the fault is known, the system is able to stop almost normally.

In most of the cases the system is able to operate almost with maximum performance despite a fault if it’s known. Fig 8 shows an example where deactivating the faulty valve AT4 does not solve the problem completely. The slower part of the trajectory is going well if the fault is known but the faster part would need such opening that valve AT4 is capable of delivering and avoiding this leads to situation where DFCU\textsubscript{BT} is switching between states 7 [1 1 1 0 0] and 16 [0 0 0 0 1]. The missing bit of BT4 decreases the resolution in just this velocity region and the DVS is not able to compensate this very well.
Figure 7. Test with valve PB1 fault

Figure 8. Test with valve BT4 fault
4. CONCLUSIONS

The fault tolerance of a digital valve system and a high-dynamic servo axis was studied. Analysis on static characteristics showed that a single fault will effect on system performance but controllability can be maintained. The tracking accuracy and maximum velocity of the system may decrease but operation can be continued if the fault is detected, identified and compensated. Simulations were used to study dynamic effects of a fault on closed loop control performance and the results showed that system may suffer from increased switching frequency and tracking error if the fault is now detected. Therefore it is very important to detect and identify the fault so that the compensation measures can be performed and operation continued. Measurements were done in a test bench where valve faults were introduced to the system. Experimental results showed that although operation can be continued despite of a single valve fault, the controllability may suffer. Some faults seem to be more difficult to compensate if the control principle is kept the same.

Future research topics include better use of mathematical pressure compensator that utilizes wider range of pressures in case a fault that can't be properly compensated with normal pressure differences. Other important topic is on-line fault detection and identification.

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ENERGY SAVING USING A MULTI-CHAMBER ACCUMULATOR: EXPERIMENTAL RESULTS AND PROOF OF CONCEPT

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ABSTRACT

Energy storage and energy recovery are subjects of major importance in mobile hydraulic systems. The implementation of hydraulic storage solutions seems natural but often such solutions fail to meet the requirements due to restrictions such as limited pressure ranges. A possible solution to overcome these restrictions is the multi-chamber piston-type accumulator. In contrast to conventional hydro-pneumatic accumulators, the multi-chamber accumulator has several fluid chambers in parallel which can be pressurised separately. This paper is concerned with the experimental investigation of the energy recovery potential of the multi-chamber accumulator. An experimental set up involving a four-chamber accumulator and a forklift mast is presented. Investigations for a simple lifting and lowering cycle yield energy savings up to 45% and a reduction of the peak supply flow rate by 56% depending on the operating mode.

KEYWORDS: multi-chamber accumulator, energy recovery, forklift, energy efficiency, digital fluid power

1. INTRODUCTION

Energy efficiency is one of the most addressed topics in contemporary research papers on fluid power. This is the case not least because fluid power research is classically an industry-driven discipline and energy is a major factor when considering the operating costs of hydraulic machines. One key approach to improve the energy efficiency in hydraulic machines, which are often operated in cyclic regimes, is the implementation of an energy recovery system. The basis of such a system is an energy storage device. As Achten pointed out in his call for innovation [1], "there is no fundamental reason why energy recuperation [...] could not be realized by means of hydraulic accumulators". Given the fact, that there are some technical limitations related to hydro-pneumatic accumulators as discussed in section 1.1 of the present article, innovative solutions for hydraulic energy recovery have to be brought to the application. One such solution, first presented by Beachley and Fronczak in [2], is a multi-chamber piston-type accumulator. These authors used such an accumulator for energy recuperation in an automotive application. A similar multi-chamber accumulator, referred to as the Digital Accumulator, was used in the simulation study [3] in the context of digital fluid power. The present article represents the continuation of the work concerning this accumulator.

A prototype of the multi-chamber accumulator is investigated experimentally with regard to its qualification for recovery of the potential energy of a forklift mast. To this end, the energy efficiency of the system with respect to an exemplary load cycle is determined in different operating modes. Additionally, measurements are used to analyse the energy dissipation in the system in order to point out possible approaches for further
optimisation. Similar work based on conventional hydraulic accumulators is presented in [4], where hydraulic energy recovery is compared to electric energy recovery. The results of the latter article are referred to in the discussion section of this contribution.

The paper is organised as follows: In the following subsection, the problem of energy recovery using hydro-pneumatic accumulators is discussed. Section 2 gives an overview of the test rig used for the experimental investigations. The results of the experiments are presented in Section 3. The paper ends with a discussion of the obtained results and some suggestions for future work in Section 4.

1.1. Problem statement

The main advantage of a hydro-pneumatic accumulator over other types of energy storage devices is a high power-to-weight ratio. Additionally, the use of a hydraulic storage device in a hydraulic machine is a natural choice since it avoids energy conversions such as hydraulic-mechanical conversions, which are usually related to energy losses. However, there is a major drawback connected with hydro-pneumatic accumulators, which is the fact, that the accumulator pressure has always to be smaller than the system pressure in order for the accumulator to be able to store energy from the system. Conversely, to withdraw energy from the accumulator, the accumulator pressure has always to be greater than the system pressure. Additionally, the accumulator pressure varies strongly depending on the state of charge of the accumulator which is a consequence of the nonlinear gas characteristics. By contrast, in a typical forklift, the load pressure is nearly constant within one load cycle, since it is mainly used to compensate the gravitational force of the load mass. Fig. 1 illustrates the problems described above. Regarding the lowering case, there is a significant gap between the load pressure and the accumulator pressure. In practice, this pressure gap must be compensated for by throttling the flow in the system. Ignoring the purple line for the time being, using the accumulator for lifting is impossible for the above-mentioned reasons.

Summing up these facts, it seems rather unfeasible to use hydraulic accumulators in a reasonable energy recovery system, at least without taking additional measures. One approach to overcome these problems is the multi-chamber accumulator. It is equipped with several separate fluid chambers, each of which can be switched independently between two hydraulic ports, typically one being a connection to the reservoir and the other one being a connection to the hydraulic system. It could be interpreted as a piston-type accumulator with an integrated hydro-static transmission with a step-wise variable transmission ratio. The purple lines in Fig. 1 schematically illustrate the solution to the aforementioned challenges: By choosing the appropriate combination of chambers to be connected to the load port, the gap between the load pressure and the gas pressure can be decreased. In fact, it also becomes possible to lift the load even when the accumulator gas pressure is significantly lower than the desired load pressure.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure1.png}
\caption{Mismatch between accumulator gas pressure (\textcolor{purple}{\textasteriskcentered}) and load pressure (\textcolor{cyan}{\textasteriskcentered}) and solution using the multi-chamber accumulator (\textcolor{cyan}{\textasteriskcentered}).}
\end{figure}
2. ACCUMULATOR TEST RIG

A sketch of the hydraulic circuit of the test rig is depicted in Fig. 2. It can be divided into the four main parts drive unit, valve manifold, accumulator prototype, and forklift mast.

The accumulator prototype is equipped with four oil chambers in annular arrangement. The dimensioning of the chambers is based on a binary coding scheme, i.e., the quotient of the volumes of two adjacent chambers is approximately 2. The chambers are separated by the moving accumulator piston and a fixed base plate. Readers interested in a more detailed description of the multi-chamber accumulator may be referred to the preceding article [3]. In the present prototype, the piston position is measured by a cable sensor mounted at the inside of the accumulator cap.

The valve manifold provides two main functions. First and foremost, its purpose is to connect each accumulator chamber either to the load port or to the tank port. To this end, a pair of 2/2 directional seat valves is connected to each chamber (V1-V8). Due to the high flow rates occurring in certain switching configurations, at least the valves connected to chamber 4 of the accumulator need to have a high flow capacity at low pressure drops. For the sake of simplicity, the same type of valves has been used for all chambers in the prototype valve manifold. The valves are doubly piloted seat valves with a flow capacity of about 200 l/min at 5 bar pressure drop, which are optimised for fast switching times of less than 15 ms. In order to ensure leakage free operation not only for the main stage but also for the pilot stages, special measures have to be taken as depicted in the valve sketch in Fig. 2.

The second function of the valve manifold is to control the load velocity of the forklift. For functional safety
reasons, two individual flow paths are implemented for lifting and lowering, respectively. Both contain a proportional throttling valve (V9, V12) which is normally closed and a check valve (V11, V14) for load protection. Additionally, a downstream pressure compensator (V10, V13) is used for pressure independent control of the flow rate. An extra benefit of these valves is a partial decoupling of the load pressure and the system pressure, which is limited by the bandwidth of the pressure compensator valves.

The drive unit basically consists of a variable displacement pump with a maximum displacement volume of $28 \text{ cm}^3/\text{rev}$ driven by an induction motor. Additionally, the maximum of the supply pressure $p_S$ can be set via a proportional pressure control valve (V15). The supply flow rate $q_S$ from the drive unit is measured by an internal gear flow meter.

2.1. Control system

The overall control system structure is depicted in Fig. 4. The central processing unit (CPU) is an industrial real-time fieldbus master based on a single core PowerPC architecture with a clock frequency of 1 GHz. The C++-based control software is executed in a cyclic task at 8 kHz frequency. Sensor data are acquired via 16bit analog-digital-converters (ADC) operating at 16kHz, which are connected to the CPU via bus. The same bus is used to connect the power output stages (POS), which allow for closed loop control of the valve currents. The control software divides into three parts which are the load control, the drive control, and the accumulator control.

2.1.1. Accumulator control

The main task of the accumulator control is to keep the pressure difference between the load pressure $p_L$ and the system pressure $p_S$ as small as possible in order to reduce the throttling losses, and, by the same time, as large as necessary to allow for the velocity control of the load. This is achieved by choosing an appropriate switching configuration $c \in \{-1, 0, \ldots, 15\}$ depending on the pressure difference $\Delta_p = |p_S - p_L|$, the gas pressure $p$, and the desired mode $\mu \in \{\text{lift}, \text{lower}, \text{off}\}$. The switching configuration $c$ determines whether a
The chamber is switched to system pressure $p_S$, to tank pressure $p_T$, or is blocked. In the special case $c = -1$, all chambers are blocked preserving the state of charge of the accumulator. For non-negative values of $c$, the $i$-th bit (starting from the least significant bit) of the binary representation of $c$ indicates the switching of the $i$-th chamber. For example, if $c = 13 = 1101_{\text{binary}}$, the chambers 1,3, and 4 are switched to system pressure, whereas the chamber 2 is switched to tank pressure.

The switching scheme is depicted in Fig. 5. At the beginning of a lifting or lowering phase, the accumulator is switched off ($c_{\text{act}} = -1$). Then, an appropriate switching configuration $c_d$ is calculated:

$$c_d = \begin{cases} \text{floor} \left( \frac{15p_G}{p_{S,\downarrow}} \right) & \text{or} \quad c_d = \text{ceil} \left( \frac{15p_G}{p_{S,\uparrow}} \right) \\
\end{cases}$$

in the lifting case and in the lowering case, respectively. The parameters $p_{S,\downarrow}$ and $p_{S,\uparrow}$ depend on the load of the forklift mast. Within the moving phases, it is checked whether the pressure difference $\Delta p$ is smaller than the required pressure difference $\Delta p,d$. If this is the case, the desired switching configuration $c_d$ is increased in the lowering case or decreased in the lifting case. Finally, after checking the feasibility of the desired switching configuration with respect to the state of charge of the accumulator, the switching configuration is set. It may be noted, that due to the separation of the accumulator control and the load control by using pressure compensated throttling valves, the switching algorithm presented here is quite plain when compared to the switching algorithm in [3].

2.1.2. Load control

Since the focus of this contribution is the proof of concept of the digital accumulator, the load control is kept as simple as possible. Therefore, the goal is to set a constant load velocity in the lifting and the lowering phase. Neglecting\(^1\) the dynamics of the throttling valves and the pressure compensators, a model for the plunger cylinder of the forklift mast may be written in terms of

\begin{align}
 m_L (\ddot{z} + g) + d_p \dot{z} &= p_p A_p \\
 (V_p + A_p \dot{z}) p_p &= \beta (q - A_p \ddot{z}) \\
 q &= f(u).
\end{align}

The total moving mass of the forklift mast is denoted by $m_L$, the area of the plunger cylinder by $A_p$, the friction coefficient by $d_p$, and the lifting height by $z$. The dead volume $V_p$ includes the volume of the piping

---

\(^1\)A consequence of this simplification is that the load dynamics is fully decoupled from the system pressure $p_S$.\n
connection and the equivalent bulk modulus is given by $\beta$. The virtual control input $u$ is related to the flow rate $q$ by a nonlinear odd function representing the quasi-static behaviour of the two pressure compensated throttling valves. The valve currents are then calculated as

$$i_1 = \begin{cases} u & \text{if } u > 0 \\ 0 & \text{else} \end{cases}, \quad i_2 = \begin{cases} -u & \text{if } u < 0 \\ 0 & \text{else} \end{cases}. \quad (2)$$

From (1), an input-output-relation is obtained by eliminating $p_p$ and $q$:

$$m_L \ddot{z} + d_p \dot{z} + \frac{A_p^2 \beta}{V_p + A_p} \dot{z} = \frac{A_p \beta}{V_p + A_p} f(u). \quad (3)$$

It may be noted that the model is differentially flat and the load position $z$ is a flat output of the system. Accordingly, it would be possible to use flatness based trajectory-tracking control for the load. Nevertheless, since a constant load velocity is the only control objective in the experiments, a simple feed forward control shall be sufficient as a first approach. The control law reads

$$u = f^{-1}(A_p \dot{z}_d), \quad (4)$$

where $\dot{z}_d$ is the desired (constant) load velocity$^2$.

2.1.3. Drive control

The task of the drive control software is to set the appropriate supply pressure and supply flow rate depending on the actual operating mode. For the basic experiment defined in section 3, three different operating modes are used, which are described in the following.

In the non-regenerative mode (NRM), the accumulator is not used for energy storage and recovery. When the load is lowered, the relief valve is opened and the load velocity is controlled by the throttling valve. In the lifting phase, the supply pressure is set to the value $p_S,\uparrow$ in order to allow for load velocity control using the valve PV2.

In the partial support mode (PSM), the accumulator control described in section 2.1.1 is active. In the lowering phase, the supply flow rate is set to zero and the accumulator is charged by the lowering load. If the gas pressure reaches $p_S,\downarrow$, i.e. the state of charge of the accumulator cannot be increased further by the load, the accumulator is locked ($c = -1$) and the relief Valve V9 is opened until the load reaches the lower end position. In the lifting phase, the accumulator control is activated again, until the accumulator is fully discharged. Then, the pump displacement is set to maximum and the supply pressure is set to $p_S,\uparrow$ again, until the load reaches the upper end position.

The third operating mode is the full support mode (FSM), where the goal is to achieve a higher state of charge of the accumulator. In this mode, the drive unit is active also in the lowering phase. Consequently, the accumulator is charged both from the lowering load and the hydrostatic drive unit. The maximum supply pressure is set to $p_S,\downarrow$ and the supply flow rate is set based on an estimation of the additional fluid volume required to achieve the desired state of charge. In the lifting phase, the maximum supply pressure is set to the upper pressure limit and the supply flow rate is limited depending on an estimation of the volume demand. Here, the supply flow supports the lifting movement of the mast.

$^2$In order to avoid abrupt stop of the mast when reaching the upper and lower end position, the valve current is continuously reduced to zero before reaching an end position.
3. EXPERIMENTAL RESULTS

In this section, the results of the experiments conducted on the test rig are presented and discussed. Most of the experiments are based on an exemplary load cycle as depicted in Fig. 6, which consists of a lowering phase, where a constant load of 1421 kg is lowered from the maximum lifting 2390 mm height to zero followed by a lifting phase, where the same load is lifted to the initial point.

3.1. Validation of the accumulator model

An additional goal of the experimental investigations is the development of mathematical system models that can be used in numerical simulations. Since the simulation model used in [3] has shown not to be in a satisfactory accordance with measurement results, a more sophisticated gas model has to be taken into account. The modelling, based on the work in [5], is summarised below. According to the first law of thermodynamics, the internal energy $u(v, T)$ of the accumulator gas satisfies

$$ du = dq - p_G dv, \quad (5) $$

where $v$ denotes the specific volume, $T$ the temperature, $p_G$ the absolute pressure and $q$ the specific heat of the gas. At the same time, the total differential of the internal energy can be written in terms of

$$ du = \left( \frac{\partial u}{\partial v} \right)_T dv + \left( \frac{\partial u}{\partial T} \right)_v dT = \left( T \left( \frac{\partial p_G}{\partial T} \right)_v - p_G \right) dv + c_v dT \quad (6) $$

denoting the specific heat capacity by $c_v(v, T)$. The heat exchange between the accumulator gas and the environment is modelled in a simplified manner as

$$ m_g dq = \frac{1}{R_{th}}(T - T_u) dt \quad (7) $$

where $R_{th}$ is the equivalent thermal resistance and the environmental temperature is denoted by $T_u$. The combination of (5), (6), and (7) yields the differential of the gas temperature:

$$ dT = -\frac{1}{m_g c_v R_{th}}(T - T_u) dt - \frac{T}{c_v} \left( \frac{\partial p_G}{\partial T} \right)_v dv. \quad (8) $$

The momentum balance of the accumulator piston yields

$$ m_k \dot{w} = \sum_{i=1}^{4} A_i p_i - A(p_G - p_A) + F_t(v, w). \quad (9) $$
Figure 7. Validation of the accumulator gas dynamics model.

Table 1. Model parameters.

<table>
<thead>
<tr>
<th>name</th>
<th>symbol</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>gas mass</td>
<td>( m_g )</td>
<td>1.567 kg</td>
</tr>
<tr>
<td>filling temperature</td>
<td>( T_0 )</td>
<td>301 K</td>
</tr>
<tr>
<td>filling pressure</td>
<td>( p_0 )</td>
<td>60 bar</td>
</tr>
<tr>
<td>total accumulator volume</td>
<td>( V_0 )</td>
<td>23.32 l</td>
</tr>
<tr>
<td>environmental temperature</td>
<td>( T_u )</td>
<td>308.9 K</td>
</tr>
<tr>
<td>equivalent thermal resistance</td>
<td>( R_{th} )</td>
<td>0.0223 K/W</td>
</tr>
</tbody>
</table>

The piston mass is denoted by \( m_k \), its velocity by \( w \), the gas-sided area by \( A \), the area of the \( i \)-th fluid chamber by \( A_i \), the chamber pressure by \( p_i \), and the ambient pressure by \( p_A \). The friction force \( F_i(v, w) \), which depends at least on both the position and the velocity of the accumulator piston, is not modelled in detail here\(^3\).

An overall model of the accumulator can be written in terms of

\[
\dot{v} = -\frac{A}{m_g} w
\]

\[
\dot{T} = -\frac{1}{m_g c_v(v, T) R_{th}} (T - T_u) + \frac{A}{m_g c_v(v, T)} T \frac{\partial \phi}{\partial T}(v, T) w
\]

\[
\dot{w} = -\frac{F_i(v, w)}{m_k} + \sum_{i=1}^{4} \frac{A_i}{m_k} p_i - \frac{A}{m_k} (p_G - p_A)
\]

by using the generalised real gas equation \( p_G = \phi(v, T) \). The validation of the accumulator gas model represented by (10b) and (10c) is illustrated in Fig. 7. The model has been simulated using the real gas equations developed in [5]. The simulation results are in good agreement with the measurements, especially when compared to the ideal gas model \( \phi(v, T) = \frac{RT}{v} \). The parameters are provided in Tab. 1. It is worth noticing, that the characteristic time\(^4\) \( \tau_{th} = \frac{m_g c_v R_{th}}{T} \) is often referred to as the time constant of the accumulator. This is valid only as an approximation, since the specific heat \( c_v(v, T) \) depends on the accumulator state.

---

\(^3\)The modelling of the (dynamic) behaviour of the accumulator sealings is beyond the scope of this contribution. Nevertheless, this does not pose significant restrictions to any further considerations in this paper since the maximum value of the friction force is small in comparison to the pressure forces.

\(^4\)The characteristic time \( \tau_{th} \) is often referred to as the time constant of the accumulator. This is valid only as an approximation, since the specific heat \( c_v(v, T) \) depends on the accumulator state.
3.2. Energy recovery

As mentioned in the introduction, the main focus of this contribution is to deliver an experimental proof of concept for energy recovery using the digital accumulator. To this end, the basic experiment is conducted repeatedly using the different operating modes explained in section 2.1.3. The result of these experiments is presented in Figure 8. In the first plot, the lifting height is depicted. The variation of the three trajectories is a consequence of the open loop control of the load velocity. The second plot shows the supply flow rates. It can be seen that in the partial support mode, the full supply flow rate \( q_s \) of approximately 27.5 l/min is needed from the moment when the accumulator is fully uncharged at \( t = 18 \) s. In the full support mode, by contrast, the maximum supply flow rate \( q_s \) is reduced to 15.5 l/min. The total hydraulic energy supplied from the drive unit is calculated by

\[
E_{\text{hyd}}(t) = \int_0^t p_s(\tau)q_s(\tau)d\tau. \tag{11}
\]

The comparison of the three operating modes with respect to the supplied hydraulic energy is depicted in the third plot of Fig. 8. In the non-regenerative mode, a total energy of 58.09 kJ is needed to lift the load to its maximum position. In the fully supported mode, the amount of energy supplied from the drive unit is reduced to 38.35 kJ. A further reduction is achieved in the partial support mode, where only 32.23 kJ are needed to lift the load to the same height. The corresponding energy saving ratio is obtained as

\[
\Gamma_{\text{FSM}} = \frac{E_{\text{hyd, NRM}} - E_{\text{hyd, FSM}}}{E_{\text{hyd, NRM}}} = 0.445 \tag{12}
\]

\[
\Gamma_{\text{PSM}} = \frac{E_{\text{hyd, NRM}} - E_{\text{hyd, PSM}}}{E_{\text{hyd, NRM}}} = 0.34. \tag{13}
\]

3.3. Analysis of the energy dissipation

The energy saving ratios achieved show the qualification of the digital accumulator for energy recovery in lifting applications. However, a more detailed analysis of the energy dissipation in the system is needed to identify possible approaches for a further increase in energy saving. To this end, the energy exchange at different points of the system is taken into account. A sketch of the balancing envelopes used for analysis can be found in Fig. 9a. The first energy exchange to be balanced is the one between the total moving mass of the forklift mast and the plunger cylinder. The input power and the output power are defined as

\[
P_{\text{in},1} = \begin{cases} -m_L g \dot{z}, & \text{if } \dot{z} < 0 \\ 0, & \text{else} \end{cases}, \quad P_{\text{out},1} = \begin{cases} m_L g \dot{z}, & \text{if } \dot{z} > 0 \\ 0, & \text{else} \end{cases} \tag{14}
\]

The second energy exchange considered involves the cylinder powers

\[
P_{\text{in},2} = \begin{cases} -p_p A_p \dot{z}, & \text{if } \dot{z} < 0 \\ 0, & \text{else} \end{cases}, \quad P_{\text{out},2} = \begin{cases} p_p A_p \dot{z}, & \text{if } \dot{z} > 0 \\ 0, & \text{else} \end{cases} \tag{15}
\]

The third exchange is the one at the load port of the accumulator with powers

\[
P_{\text{in},3} = \begin{cases} \sum_{i=1}^{4} \xi_i p_i A_i w, & \text{if } \dot{z} > 0 \\ 0, & \text{else} \end{cases}, \quad P_{\text{out},3} = \begin{cases} -\sum_{i=1}^{4} \xi_i p_i A_i w, & \text{if } \dot{z} < 0 \\ 0, & \text{else} \end{cases} \tag{16}
\]

where \( \xi_i = 1 \) if the \( i \)-th chamber is connected to the load port and \( \xi_i = 0 \) otherwise. Finally, the total energy exchange at the hydraulic ports of the accumulator is regarded in the fourth case, where the powers are

\[
\text{The peaks that can be observed in the flow rate measurement result from the switching between the accumulator configurations.}
Figure 8. Comparison of different operating modes.
written as

\[
P_{in,4} = \begin{cases} 
\sum_{i=1}^{4} p_i A_i w, & \text{if } w > 0 \\
0, & \text{else}
\end{cases}
\]

\[
P_{out,4} = \begin{cases} 
-\sum_{i=1}^{4} p_i A_i w, & \text{if } w < 0 \\
0, & \text{else}
\end{cases}
\]

Following [4], the corresponding energy efficiencies with respect to the considered load cycle may be calculated as

\[
\eta_i = \frac{\int_0^T P_{out,i}(t) \, dt}{\int_0^T P_{in,i}(t) \, dt}
\]

for \(i = 1, \ldots, 4\),

where \(T\) is the total cycle time. For the dissipation analysis, a load cycle as depicted in Fig. 9b is used, where the drive unit is deactivated and the load is lifted using only the energy that can be recovered. The computation of the energy efficiencies of the system parts for this cycle yields

\[
\eta_1 = 0.458 \\
\eta_2 = 0.527 \\
\eta_3 = 0.869 \\
\eta_4 = 0.981.
\]

The analysis shows that the dissipation due to heat exchange and friction in the accumulator is of minor importance as measured by the overall energy dissipation. The quantity \(1 - \frac{\eta_3}{\eta_4} = 0.114\) is a measure for the inherent energy losses due to the compression and decompression of the oil chambers switched from the load port to the tank port of the accumulator. In a similar manner, the losses due to the hydraulic resistances and capacities between the accumulator and the plunger cylinder can be quantified by \(1 - \frac{\eta_2}{\eta_3} = 0.394\). The energy dissipation due to the friction in the forklift mast can be judged by the quantity \(1 - \frac{\eta_1}{\eta_2} = 0.131\).

Summarising these results, it can be stated that the most significant part of the energy losses is generated in the hydraulic resistances in the valve manifold and in the piping of the test rig, not least because of the use of pressure compensated throttling valves for load control.

4. DISCUSSION AND FUTURE WORK

The article summarises selected results of experimental investigations concerning a prototype system of a multi-chamber accumulator. The accumulator is used for energy recovery in a forklift mast, which is a prime
example for a mobile hydraulic system suffering from a low hydraulic efficiency in the first place. The focus is laid on the proof of concept, i.e. to show the qualification of the accumulator for energy recovery in the system considered.

The test rig is described in detail regarding both its hardware and its software. The basic principle in the current setting is to keep the pressure differential at the load control valves as small as necessary for the pressure compensation to work properly and facilitate the load control. The load control itself is kept very simple in the experiments presented. Nevertheless, the differential flatness of the load model discussed provides a line of approach for future work on load control which is more a topic for the interaction between the machine and the operator than for energy recovery.

In order to reduce costs for prototypes and testing, it is desirable to have mathematical models of high quality that can be used in numerical simulations. Therefore, part of the experiments aimed on the identification of model parameters and the validation of the models. It is shown, that the use of a sophisticated gas model is indispensable when the dynamic behaviour of the accumulator is of interest. However, the proper modelling of the sealing friction in the accumulator remains an open point for the time being.

The qualification of the accumulator system is judged by comparing different operating modes when running the same basic load cycle, where a fixed load is lowered to the minimum position and lifted again to its starting position. The experiments yielded a maximum energy saving ratio of 44.5%. In [4], a similar peak energy saving ratio was observed using conventional hydraulic accumulators with preload gas pressures adapted to the load cycle. In the same article, a system based on hydraulic accumulators combined with a hydraulic transformer is pointed out as possible approach for future work. The multi-chamber accumulator might be interpreted as such a system in a special sense, where a digital hydraulic transformer such as the one proposed by Bishop in [6] is integrated within the accumulator.

In the dissipation analysis, it is pointed out that a major part of the energy losses is a result of the resistance control based on pressure compensated throttling valves. At the same time, this is the part of the system which is qualified most for taking measures to increase the overall energy efficiency. One possible approach is to replace the resistance control by a displacement control system such as an internal gear unit driven by a servo motor. This brings about several advantages. The most obvious one is that the pressure difference due to the mismatch of the accumulator pressure and the required load pressure could partly be recovered by using an appropriate inverter in combination with an electrical energy storage instead of being dissipated completely in the resistance control. Another advantage lies in the load control itself, which can then be realised with the servo motor. The approach aims to combine the advantages of both electric and hydraulic energy recovery systems.

Summarising, it can be stated that the multi-chamber accumulator is well qualified for energy recovery at least in the presented lifting application. Beyond the proof of concept provided by the results presented, future work must involve the further characterisation of the energy efficiency with respect to varying load cycles that are characteristic for mobile hydraulic machines. Furthermore, the possibility to adapt the accumulator pressure to the load pressure to some extent motivates the use in hydraulic systems other than forklifts. Within the discussion in this section, more links to future work are pointed out such as development of mathematical system models, the implementation of a more sophisticated load control, and not least the use of a servo motor based load control.
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REFERENCES


RESEARCH ON KEY TECHNOLOGY OF HIGH-PRESSURE HYDRAULIC IMPULSE TESTING SYSTEMS

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ABSTRACT
In recent years, hydraulic transmission systems are widely used in construction and agricultural machinery owing to their high power density, high compactness, and flexible control. In the operation of hydraulic systems, hydraulic impulse pressure, also known as water hammer wave, may arise. The impulse pressure force not only damages hydraulic components and impacts the function of systems, but also causes noise and vibration. To guarantee the reliability and performance of hydraulic systems, it is of vital importance to do impulse testing for hydraulic components.

In this paper, a hydraulic impulse testing system based on the pressure-boost cylinder is designed, the output pressure of which can be as high as 42 MPa. The pressure-boost cylinder is designed to work in a differential mode, which is under the control of a three-way servo valve. Mathematical model of the system is built, and indicates that it is difficult to obtain satisfactory control effect by traditional PID control method. A fuzzy repetitive controller is designed and applied to the system. Both simulation and experiment results show that the fuzzy repetitive control system can achieve a better performance than traditional PID control system, with the trace error reduced by nearly 50%. In addition, common pressure waveforms including sine-waves and peak-waves can be produced by the system, which can satisfy the demand of most of existing hydraulic impulse testing equipment.

Key words: hydraulic impulse testing; pressure-boost cylinder; fuzzy repetitive control; mathematical model; high pressure

1. INTRODUCTION
In recent years, due to high power density, durability, accuracy and flexible control, the electro-hydraulic pressure control systems are widely used in various industrial fields, such as the pressure impulse testing systems, the vehicle anti-lock brake systems, the hydraulic injection molding machines, and so on.\textsuperscript{1,2} The pressure tracking capabilities of electro-hydraulic systems are widely discussed, and various control methods are proposed, including self-tuning proportional-integral-derivative (PID) control, fuzzy control, quantitative feedback theory method, inverse control, and neural network based control.\textsuperscript{3} In some cases, the reference signal pressure to be tracked may be periodic, such as the reference signal for pressure impulse testing systems. To utilize this specific characteristics of the signals in control system design, the repetitive control is proposed, which is related to the learning control and originated from the idea of the internal model principle.\textsuperscript{4,5} The internal model principle is attached inside the feedback loop and behaves as the generator of a periodic signal so as to achieve zero tracking error. Various repetitive controllers have been designed and applied to a wide variety of systems. A repetitive model dealing with disturbances with a
known period is originally formulated by Inoue. A two parameter robust repetitive control is designed by Jianwu Li using structured singular values. Another adaptive repetitive controller is designed by N Uchiyama and applied to feed drive systems. And this is followed by some successful work by Edi Kurniawan in which a robust repetitive control with time-varying sampling periods is designed.

Theoretically, a repetitive control system is able to trace any high-frequency components in a periodical reference signal. However, it is difficult to stabilize such a system. To solve the problem, a low-pass filter $F(s)$ is adopted to cutoff high frequencies, and the improved repetitive control system is shown in Figure 1. And various filters are designed and studied by researchers. Nevertheless, while the filters stabilizing the systems, the control accuracy is reduced and a time delay arises since most of the low-pass filters may produce a phase lag. In other words, there is a trade-off between stability and accuracy.

![Figure 1. Schematic diagram of improved repetitive control system](image)

In this paper, a fuzzy repetitive controller is designed and applied to the electro-hydraulic periodic pressure servo system (EHPPSS) to achieve high precision with little phase lag, the principle of which is that the introduce of fuzzy controller to the repetitive control system can cutoff high-frequency components while causing no phase lag for low-frequency components. Simulation results show that the controller is convergent, with the control error decreasing with the increase of cycle times and the relative error reduced to 0.1% after 10 cycle times. Experiments are carried out on the EHPPSS, showing that the fuzzy repetitive control system can achieve a much higher precision than the PID control system for both sine-wave input and peak-wave input, with the relative error reduced by 54% and 40%.

2. SYSTEM DESCRIPTION

The schematic diagram of the EHPPSS is shown in Figure 2. It mainly consists of the pressure source, servo valve, pressure-boost cylinder, check valve, and the load. The pressure source is used to provide hydraulic power for the system, including a variable displacement pump, a check valve, a relief valve, and an accumulator. The load of the EHPPSS may be hydraulic components to be tested in a pressure impulse testing system, chambers of a hydraulic injection molding machine, and so on. To achieve a pressure higher than that of the pressure source, which may be required in certain circumstances, a pressure-boost cylinder is utilized.

![Figure 2. Schematic diagram of the electro-hydraulic periodical pressure servo system](image)

The pressure-boost cylinder is designed to work in a differential mode, under the control of the three-way servo valve, and the working principle is as follows. When a positive signal is provided for the servo valve, the piston of the hydraulic cylinder moves to the right, so the hydraulic oil in the load is compressed and
correspondingly the pressure rises. In contrast, when a negative signal is provided for the servo valve, the piston of the hydraulic cylinder moves to the left, so the hydraulic oil in the load is relaxed and correspondingly the pressure decreases.

3. MATHEMATICAL MODEL

The frequency method is used to analyze the dynamic characteristics of the EHPPSS. In order to facilitate the analysis, the servo valve is assumed to possess a frequency bandwidth much higher than other parts of the system. Accordingly, the transfer function of the servo valve can be defined as

\[
\frac{X_s}{U} = K_{sv}
\]  

where \( X_s \) is spool displacement of the servo valve, \( U \) is the demand input signal, and \( K_{sv} \) is the displacement gain of the servo valve.

For small perturbations, the linearized equation of the servo valve flow rate can be described as

\[
Q = K_q X_s - K_c P_c
\]  

where \( Q \) denotes flow rate of the servo valve, \( K_q \) and \( K_c \) denote the flow gain and the pressure coefficient of the servo valve, \( P_c \) denotes the pressure of control chamber of the hydraulic pressure-boost cylinder.

The flow continuity equation of the control chamber of the hydraulic pressure-boost cylinder can be described as

\[
Q = A_h s^2 X + C_{tp} P_c + \frac{V_t}{\beta_c} s P_c
\]  

where \( A_h, X \) denote the control area and piston displacement of the hydraulic pressure-boost cylinder, \( C_{tp} \) denotes the leakage coefficient of the control chamber, \( V_t \) denotes the total volume of the control chamber, and \( \beta_c \) denotes the oil bulk modulus.

The force balance equation of the piston of the hydraulic pressure-boost cylinder can be described as

\[
A_h P_c = m s^2 X + B_p s X + K X
\]  

where \( m \) and \( B_p \) denote total mass and viscous damping coefficient of the piston of the hydraulic pressure-boost cylinder, \( K \) denotes the equivalent stiffness of the load. And by combining equations (2)-(4),

\[
\frac{X}{X_v} = \frac{K_q / A_h}{mV_t / \beta_c A_h^2 + (mK_{cc} / \beta_c A_h^2 + B_p V_t / \beta_c A_h^2) s^2 + (1 + B_p K_{cc} / \beta_c A_h^2 + KV_t / \beta_c A_h^2) s + KK_{cc} / A_h^2}
\]

where \( K_{cc} = K_c + C_{tp} \) is the total flow-pressure coefficient of the system. For the fact that \( B_p \) is so small that

\[
\frac{K_{cc} B_p}{A_h^2} \ll 1,
\]

so the above equation can be simplified as

\[
\frac{X}{X_v} = \frac{K_q / A_h}{mV_t / \beta_c A_h^2 + (mK_{cc} / \beta_c A_h^2 + B_p V_t / \beta_c A_h^2) s^2 + (1 + KV_t / \beta_c A_h^2) s + KK_{cc} / A_h^2}
\]
Define hydraulic natural frequency and hydraulic damping ratio of the hydraulic cylinder as

\[ \omega_h = \sqrt{\frac{\beta_h A_h^2}{V_t m}} \]  
(7)

\[ \xi_h = \frac{K_{cc}}{2A_h} \sqrt{\frac{\beta_m}{V_t}} + \frac{B_p}{2A_h \sqrt{\beta_m m}} \]  
(8)

Define the hydraulic spring stiffness of the hydraulic cylinder as

\[ K_h = \frac{\beta L A}{V_t} \]  
(9)

So the transfer function of the hydraulic cylinder from the servo valve can be shown as

\[ \frac{X}{X_r} = \frac{K_q/A_h}{\frac{s^3}{\omega_h^2} + \frac{2 \xi_h}{\omega_h} \frac{s^2}{\omega_h} + (1 + \frac{K}{K_h})s + \frac{KK_{ce}}{A_h^2}} \]  
(10)

For the booster chamber of the hydraulic pressure-boost cylinder, which is connected with the load, the pressure can be described as

\[ P_L = \frac{\beta A_L}{V_0} X \]  
(11)

where \( V_0 \) and \( \Delta V \) denote the booster chamber volume including the load volume and its changes, \( A_L \) denotes the effective area of the chamber, and \( \beta \) denotes the equivalent bulk modulus of booster chamber and the load.

Taking all the above into consideration, mathematical model of the EHPPSS is shown as Figure 3.

**Figure 3. Mathematical model of the electro-hydraulic periodical pressure servo system (EHPPSS)**

The transfer function can be translated to

\[ P(s) = \frac{P_L}{U} = K_{sv} \frac{K_q/A_h}{\frac{s^3}{\omega_h^2} + \frac{2 \xi_h}{\omega_h} \frac{s^2}{\omega_h} + (1 + \frac{K}{K_h})s + \frac{KK_{ce}}{A_h^2}} \frac{\beta A_L}{V_0} \]  
(12)

It can be seen that the transfer function consists of an inertial element and a second-order oscillation element with gain. Note that \( \frac{\beta A_L}{V_0} \) in the transfer function changes with the load of the system, so the gain of the system transfer function may vary in a large range. That is to say, it is difficult for the conventional PID control method to achieve satisfactory effect in such a system, because it is very easy to cause oscillation with large gain or large trace error with small gain. Therefore, a fuzzy repetitive control method is designed and researched for the EHPPSS to achieve stability and precision.
4. CONTROL METHOD

The block diagram of the fuzzy repetitive control system is shown in Figure 4, which is composed of a repetitive controller to eliminate periodic error, a feed-forward loop $a(s)$ to improve the speed of the system, a fuzzy controller to stabilize the system, a compensator $C(s)$ to match the controller, the system model $P(s)$, and a feedback loop to achieve high precision. The time delay $T$ in the repetitive controller depends on the cycle time of the reference input signal and can be adjusted online to keep consistent with the cycle time of the input signal.

![Figure 4. Block diagram of the fuzzy repetitive control system](image)

The fuzzy controller has two input variables, the error $e$ and the error derivative $\frac{de}{dt}$.[16] The linguistic terms of the error $e$ are negative big (NB), negative small (NS), zero (ZO), positive small (PS), and positive big (PB). The linguistic terms of the error derivative $\frac{de}{dt}$ are negative big (NB), negative medium (NM), negative small (NS), zero (ZO), positive small (PS), positive medium (PM), and positive big (PB). The linguistic terms of the fuzzy controller output $\alpha$ are zero (ZO), small (S), medium (M), and big (B). Membership functions for input and output variables of the controller are shown in Figure 5.

![Figure 5. Membership functions for input and output of the fuzzy controller](image)
The fuzzy control rules are expressed in Table 1. When the absolute value of the error derivative $de/dt$ is big enough, the output of the fuzzy controller $\alpha$ approaches 0 to stabilize the system. So the effect of the fuzzy controller here is the same as a low-pass filter. Nevertheless, there is no phase lag for the low-frequency component. When the absolute value of the error $e$ is small, the output of the fuzzy controller $\alpha$ is big to improve the precision. As shown in Equation (13), the weighted average method is adopted for defuzzification, with simple calculation and smooth output.\(^{[17-19]}\)

$$u = \sum_{j=1}^{n} k_j e_j / \sum_{j=1}^{n} k_j$$

where $u$ is the defuzzification result, $e_j$ is the centroid of membership functions, and $k_j$ is random weighting coefficient.

### Table 1. Fuzzy control rules

<table>
<thead>
<tr>
<th>$\alpha$</th>
<th>$de/dt$</th>
</tr>
</thead>
<tbody>
<tr>
<td>NB</td>
<td>NB</td>
</tr>
<tr>
<td>NM</td>
<td>ZO</td>
</tr>
<tr>
<td>NS</td>
<td>ZO</td>
</tr>
<tr>
<td>ZO</td>
<td>ZO</td>
</tr>
<tr>
<td>PS</td>
<td>ZO</td>
</tr>
<tr>
<td>PB</td>
<td>ZO</td>
</tr>
</tbody>
</table>

5. SIMULATION

To study the performance of the EHPPSS with fuzzy repetitive controller, simulation models are built in AMESim, as shown in Figure 6. In the simulation model, the pressure-boost cylinder is built using the hydraulic component design library in the AMESim. Other hydraulic components such as the pressure source, check valve, and accumulator are built using the standard hydraulic library. And the load is a hydraulic hose. The simulation parameters are shown in Table 2.

### Table 2. Simulation parameters of the EHPPSS with fuzzy repetitive controller

![Figure 6. Simulation model of the EHPPSS with fuzzy repetitive controller](image)
<table>
<thead>
<tr>
<th>Pressure source</th>
<th>system pressure, MPa</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>maximum flow rate, L/min</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td>oil bulk modulus, MPa</td>
<td>1100</td>
</tr>
<tr>
<td>Servo valve</td>
<td>natural frequency, Hz</td>
<td>350</td>
</tr>
<tr>
<td></td>
<td>spool mass, kg</td>
<td>0.06</td>
</tr>
<tr>
<td></td>
<td>maximum flow rate, L/min</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>corresponding pressure drop, MPa</td>
<td>1</td>
</tr>
<tr>
<td>Pressure-boost cylinder</td>
<td>piston mass, kg</td>
<td>10</td>
</tr>
<tr>
<td></td>
<td>area of control chamber, m²</td>
<td>1.23×10⁻²</td>
</tr>
<tr>
<td></td>
<td>area of system pressure chamber, m²</td>
<td>8.65×10⁻³</td>
</tr>
<tr>
<td></td>
<td>area of booster chamber, m²</td>
<td>2.8×10⁻³</td>
</tr>
<tr>
<td></td>
<td>maximum piston displacement, m</td>
<td>0.072</td>
</tr>
<tr>
<td>Load (hydraulic hose)</td>
<td>diameter, m</td>
<td>0.019</td>
</tr>
<tr>
<td></td>
<td>length, m</td>
<td>0.96</td>
</tr>
<tr>
<td></td>
<td>effective bulk modulus, MPa</td>
<td>850</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>cycle time</th>
<th>Maximum error (MPa)</th>
<th>Relative error</th>
<th>$E_p$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>8</td>
<td>40%</td>
<td>852</td>
</tr>
<tr>
<td>3</td>
<td>1.2</td>
<td>6%</td>
<td>38.2</td>
</tr>
<tr>
<td>5</td>
<td>0.3</td>
<td>1.5%</td>
<td>2.1</td>
</tr>
<tr>
<td>10</td>
<td>0.02</td>
<td>0.1%</td>
<td>0.02</td>
</tr>
</tbody>
</table>

The sine-wave-input response and trace error of the EHPPSS with fuzzy repetitive controller are shown in Figure 7. To evaluate the precision of the system, the system error in a period is defined as

$$E_p = \int_0^T e^2 dt$$

(14)

where $e$ denotes the trace error. The error data in the simulation of fuzzy repetitive control system is listed in Table 3, which displays the maximum trace error and relative error after different cycle times. According to the simulation curves and error data, the designed fuzzy repetitive controller is proved to be convergent, with the control error decreasing with the increase of cycle times and the relative error reduced 0.1% to after 10 cycle times.
6. EXPERIMENT

For further analysis of the performance and characteristics of the EHPPSS with fuzzy repetitive controller, a prototype is manufactured, and experiments are carried out. To compare the experimental results with the simulation results, the parameters of the prototype are set the same as the simulation model, as shown in Table 2.
The sine-wave demand input is the same as Figure 7 (a), the sine-wave-input response and trace errors after 10 cycle times of the EHPPSS with fuzzy repetitive controller and PID controller are shown in Figure 8 and Figure 9, and the key data are shown in Table 4. The experimental results show that the trace error of the fuzzy repetitive control system decreases with the increase of cycle times, which agrees well with the simulation results. It is noted that the trace error of the experiment results is larger than that of the simulation results, which may arise from the manufacturing errors and parameter setting errors. Although the trace error of the fuzzy repetitive control system is larger than that of the PID control system at the first few cycles, with the increase of cycle times, the trace error of the fuzzy repetitive control system decreases fast and can achieve an error much smaller than the PID control system. In conclusion, the fuzzy repetitive control system can achieve a much higher precision than the PID control system, with the relative error and the error in a period $E_p$ reduced by 54% and 99%.

(a) Demand input (peak-wave)  (b) Results of the system with fuzzy repetitive controller

(c) Results of the system with PID controller

Figure 10. Experiment results of peak-wave-input response of the EHPPSS
Table 4. Experiment results of the EHPPSS with fuzzy repetitive controller and PID controller

<table>
<thead>
<tr>
<th></th>
<th>Sine-wave-input response</th>
<th>Peak-wave-input response</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum error (MPa)</td>
<td>Relative error</td>
</tr>
<tr>
<td>Fuzzy repetitive</td>
<td>0.12</td>
<td>0.6%</td>
</tr>
<tr>
<td>control</td>
<td></td>
<td></td>
</tr>
<tr>
<td>PID control</td>
<td>0.26</td>
<td>1.3%</td>
</tr>
</tbody>
</table>

For the fact that the peak-wave, as shown in Figure 10(a), is widely used in EHPPSS, such as the hydraulic impulse testing systems, experiments based on peak-wave input are also carried out. The peak-wave-input response and trace errors of the EHPPSS with fuzzy repetitive controller and PID controller are shown in Figure 10 and Figure 11, and the key data after 10 cycle times are shown in Table 4. The same as the sine-wave input, trace errors of the fuzzy repetitive control system decreases with the increase of cycle times while trace errors of the PID control system remains unchanged. Due to the high-frequency components in the peak-wave input signal, the relative error is larger than that of the sine-wave input. Nevertheless, the fuzzy repetitive control system still achieves a higher precision than the PID control system, with the relative error and error in a period $E_p$ reduced by 40% and 87%.

7. CONCLUSIONS

To obtain better performance and characteristics of electro-hydraulic periodical pressure servo system, which is widely used in industrial fields, a fuzzy repetitive controller is designed in this study. The introduce of fuzzy controller to the repetitive control system can stabilize the system without decreasing the precision by cutting off high-frequency components while causing no phase lag for low-frequency components.

The electro-hydraulic periodical pressure servo system is designed, utilizing a hydraulic pressure-boost cylinder under the control of a three-way servo valve. Mathematical model of the system is built using frequency method, which describes the system with an inertial element, a second-order oscillation element and variable gain, indicating that it is difficult to achieve a satisfactory control effect with the conventional PID control. Then the control method based on fuzzy repetitive controller is designed and simulation results proves that the control system is stable and convergent, with the control error decreasing with the increase of cycle times and the relative error reduced to 0.1% after 10 cycle times.
Experiments are carried out on the EHPPSS, showing that the fuzzy repetitive control system can achieve a much higher precision than the PID control system for both sine-wave input and peak-wave input, with the relative error reduced by 54% and 40%.

FUNDING

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References:


HIGH RESPONSE OVERLOAD PROTECTION VALVE (HROPV) FOR HEAVY HYDRAULICS

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ABSTRACT

The paper web speed in paper machine can be 1800 m/min (30 mm/millisecond). In paper web breaks several layers of paper can go through the high loaded nip that can include soft covered rolls. Rolls and belts will damage if nip is not unloaded before the coming pressure peak. Similar quick unloading requirements; relieving load within few milliseconds, exist also in many other application areas of heavy hydraulics. In 2007 Metso Paper discussed with Moog GmbH (Finland and Germany) to get quick electrical on-off valve that will react on pressure rise velocity, the start of acceleration. Moog considered the challenge and after some time gave suggestion to develop cartridge valve based on pure hydraulic elements instead of electrical components. For Metso point of view, the simple technology could fulfill the requirements of robust system with high reliability. During 2007 a working solution for valve was invented by Moog GmbH. Work was divided and Metso Paper was responsible for test environment, pilot unit, measuring system and control cover hydraulic circuit. The test unit, impact test device, was built around a pendulum where the mass weight and falling height were adjustable. The target set point for pressure rise velocity was 10000 bar/s and target for 100 % step response was 2 ms. The HROPV fulfilled the dynamic requirements and results were much better compared to results with accumulator. On the other hand, the set value adjustability was poor and the repeatability of function was not best possible and needed future work.

KEYWORDS: paper machine, hydraulic loading, overload protection, damping, pressure relief valve, quick unloading

1. INTRODUCTION

The nip loading is the key application area for hydraulics in paper machinery. Inside this area big focus is directed to deflection compensation control of rolls. In steel industry, the gap control between rolls has analogy to nip linear force control in paper industry. Traditional roll nip loading arrangement in one nip calender is presented in Figure 1. Typical long nip in press part of paper machine can be seen in Figure 2. Figure 3 shows the pressure impulse directed to paper web in roll nip and in long nip.
Figure 1. Traditional roll nip and loading with auxiliary cylinder

Figure 2. Long shoe nip in paper machine press section. In the bottom left Figure, the roll below is the Belt roll with flexible rotating cover. The counter roll is deflection compensated steel roll. Both rolls have equal size loading elements inside and the rolls are loaded against each other with common control pressure.
In paper web breaks, several layers of paper going through the nip present the most general reason for pressure peaks in nips. Because of peaks, the overload protection of soft covered rolls in roll nips and the protection of flexible belt in belt nips has always had to be taken into account in loading system design.

2. OVERLOAD PROTECTION METHODS

In roll nips, the accumulator near cylinder port has been the traditional method to damp the quick peak (Figure 4). Partly because of lacks in design, (accumulator assembling place, accumulator size, wrong precharge pressure, dimensioning of pipes and fittings etc.) and the difficulty of task (very high pressure increase velocity), the damping effect has not been good in all cases. However, the accumulator has normally been much better element for peak damping compared to other techniques, for instance pressure relief valves.

Direct acting relief valves are usually the quickest acting among relief valves. Response times of screw type relief valves in SUN hydraulics program can be seen in Figure 5. The “kick down” relief valve releases the pressure totally instead of only limiting overpressure. The response time in many market available “kick down” versions is long. Cartridge (2/2-way) based pilot operated quick relief valves have also been developed by modifying standard cartridges (Figure 6). Also in these, the response time for nip application is not short enough. The break disc (Figure 7) is good example of insulator type irrevocable overpressure protection. They can not be used in continuously running processes where the shutdown costs are high.

In long nip application, the electrical web break indicators along the production line have been utilized to get the unloading signal to electrical valves. Unloading cartridge valves inside the roll have also been used (Figure 8) in most difficult cases. The electrical system works if the web break information is got early enough before the nip. This is not possible in all cases.

In 2007 started the discussion between Metso Paper and Moog GmbH to create system that would react on pressure increase velocity; the start of acceleration, the fastest possible reaction to be achieved (Figure 9). Electrical system was the first in minds of participants. The delays in data handling and transmission presented challenge for system where the response target was below 2 ms. After considerations Moog suggested to return to pure hydromechanical system.

The idea to design and apply pure hydromechanical time dependent elements to improve damping is naturally not new in hydraulics. However, the use have been forgotten in revolution of electrohydraulics. Fifty years ago M. Guillon /2/ has handled the subject in his book and in several articles and he has also utilized
hydromechanical damping devices in real systems. In Technical University of Aachen in 1975...1985, the theme had key research focus. The design rules and guidelines to apply time dependent elements were laid in several research projects. Many thesis works, including Weingarten /9/, Luhmer/5/ and Engelsdorf/1/, summarize the results. Examples of hydraulic circuits equipped with damping elements are shown in Figure 10. Quick acting dynamic pressure relief valves developed in research projects can be seen in Figure 11.

Figure 4. Typical "wad damping" accumulator in paper machine nip. The precharge pressure is often 10...20% higher than the maximum loading pressure. The accumulator is often undersized compared to flow peak. When using accumulator, the protection system does not need any resetting after the peak.

Figure 5. Screw type relief valves from SUN hydraulics Fig. 5a; direct operated relief valve, type RDJA, 0...760 lpm, catalog value for response time is 2 ms. Fig. 5b; pilot operated rapid response relief valve, type RPKE, 0...760 lpm, catalog response time is 2 ms. Catalog times are optimistic. Fig. 5c; pilot operated "kick-down" relief valve, type RQKB, 0...760 lpm, typical response time 25 ms. In "kick down" valve, the pilot chamber pressure is released to tank when main valve opens. The system pressure need to be cutted before the valve resets.
Figure 6. Pilot operated vented relief valve based on 2-way cartridge valve size NG40 with pilot valve inside the cartridge cone (Parker Lokomec). Assembling space is minimized. Typical response time is about 20 ms. Normally mentioned response time for pilot operated cartridge based relief valve in size over NG32 is over 100 ms.

Figure 7. Break disk (Berstscheibe) used in hot water and in hot oil systems to protect system from abrupt overpressures.
Figure 8. Loading cylinders inside the Belt roll. Hydraulically pilot operated 2/2-way cartridge valves release the loading pressure from two cylinder manifold.

Figure 9. The overload protection triggered by pressure rise velocity. Pressure increase velocity “represents” the derivative of acceleration, jerk, the rate of change of acceleration. Working example: if you hit cylinder piston with a hammer, the piston should escape and fall down. The principle includes the “kick down” feature. /8/
Figure 10. Circuits equipped with RC-elements to damp pressure peaks when lowering the mass. The pressure increase velocity defines the valve opening. The valve only limits the pressure by creating "dynamic leakage" but the pressure is not released totally as in "kick down" relief valves. Above a circuit utilizing spool valve. Below a circuit with cartridge valve. /5/

Figure 11. Above a direct operated quick acting "dynamic pressure relief valve" with RC element and stiff spring. Below a cartridge valve with RC circuit defining the triggering value to open the valve. The orifice inside the cartridge has linear characteristics (R behaviour) with several turbulent behaviour orifices connected in series. /9/

3. HROPV PRINCIPLE AND DESIGN FEATURES

Belt roll (Figure 2) was the target application for HROPV. One valve would unload two cylinders (Figure 8). Because of common loading pressure for both nip rolls, the unloading of one cylinder pair must lead to unloading of whole nip.

The following features guided the valve design:

- Valve is triggered by pressure set point (relief pilot)
- Valve is triggered by pressure rise velocity (RC element)
- High response, 100% step response of size 32 is 2ms
- Open position of valve spool is hydro-mechanically position controlled without touching the end stop. (end stroke damping, no self-destruction)
- Automatic reset of overload protection valve when flow is zero
- Simple robust design adoptable to different sizes
- Design allows standard ISO cartridge valve cavity
- Suitable for 350 bar operating pressure
- Low leakage main stage
- Robust hydro-mechanical solution, no need for cables and sensors

The principal drawing of valve in hydraulic circuit is presented in Figure 12. The first drafts of design are shown in Figure 13.

**Figure 12: HROPV principle.** The set value, pressure rise velocity leading to unload, is defined by the diameter of orifice (R) and control chamber volume. The seat type control edge (3) opens first and releases the pilot chamber pressure. The edge (1) controls main flow and edge (2) closes the pilot chamber connection to tank (end stroke damping). The area ratio is 1. The spring force keeps the valve closed and defines the pressure difference over the orifice. /8/

Mechanical disturbance acting on cylinder. Overload is characterized by high pressure rise velocities

The mathematical models of hydraulic time dependent elements have been compared in references /5/ and /9/. The RC-element in pilot circuit (Figure 14) is typical P-T₁-element. Constant value R (resistance factor) without viscosity effects would be the optimum. In orifices \((l/d<0.5)\) with turbulent flow, the R factor depends on pressure difference (Figure 15). Valve equipped with RC-element forms D-T₁-element (Figure 14). The more accurate valve modelling leads to D-T₂ and D-T₃-elements. It has been shown in reference /5/ that simple D-T₁-model leads to accurate results in many cases handling damping circuit valves.

The RC-circuit in pilot stage and the flow characteristics of main valve are in design focus. The basic equations of RC element are:
\[ R_{\text{orifice}} = A \alpha_p \sqrt{\frac{1}{2 \rho \Delta p_{\text{op, point}}}} \]  
\[ C_H = \frac{V_{\text{sys}}}{E_{\text{fluid}}} \]  
\[ \frac{\Delta p_{\text{sys}}}{p_{\text{sys}}} = \frac{1}{1 + \frac{R_{\text{orifice}}}{C_H s}} \]  
\[ \frac{\Delta p_{\text{orifice}}}{\Delta t} = \frac{C_H}{R_{\text{orifice}}} \]

In equations \( R_{\text{orifice}} \) is the linearized hydraulic resistance (flow-pressure coefficient), \( C_H \) is the hydraulic capacity, equation (3) represents the transfer function of RC-element and equation (4) is the sizing equation (\( \Delta p_{\text{sys}}/\Delta t \) is the pressure rise velocity triggering the valve). The discharge coefficient \( \alpha_0 \) for sharp edged orifice with \( l/d < 0.5 \) is normally 0.6 in calculations. The bulk modulus \( E_{\text{fluid}} \), normally assumed as constant, can vary considerably mainly because of free air in oil.

When modelling the main valve for quick pressure release application, the discharge coefficient and flow saturation need special attention. The typical behaviour of discharge coefficient in one type cartridge valve is shown in Figure 16. References [3,4,6,7,10,11/ deal with discharge coefficient and flow saturation behaviour in different type cartridge valves. Many of the investigations handle the behaviour in small openings or partly opened valves. In quick pressure release application with "kick down valve", the cartridge will open fully.

The inductivity and capacity of fluid in pipes affect the system response. The pipe represents low pass filter in the system (Figure 17). Especially on the low pressure side, the return pipe effects have often been forgotten though on the pressure side, the valve has been assembled near the actuator and pipes are short. All drillings and manifold connections need to be designed according the max. possible flow through the system. Figure 18 shows approximately the flow limits in respect with cartridge valve nominal size (inner diameter of flow area).

![Figure 13. The HROPV design draft.](image)
Figure 14. The step response of R-C₁- and D-T₁-element. /5/

Figure 15. Flow and resistance coefficient ($\Delta p/\Delta Q$) as a function of pressure difference. Laminar orifice (short pipe or capillary) represent linear resistance (R is constant) if viscosity remains constant. In turbulent orifice ($\lambda/d<0.5$), the R value depends on working point. /9/

Figure 16. Discharge coefficient behaviour as a function of Reynolds number. After reaching turbulent flow conditions, the coefficient has constant value. The measurements are normally done by keeping the incoming pressure constant and changing the return pressure. Depending on cartridge type, the max values are 0.55...0.85. /10/
mechanic equivalent circuit

\[ \begin{array}{cctc}
Q_0 & p_0 \\
\hline
m & R & C & \frac{1}{\xi} \\
\hline
Q_1 & p_1 \\
\end{array} \]

electro-

equivalent
circuit

Natural frequency of hydraulic pipe:
\[ \omega_{\text{pipe}} = \sqrt{\frac{1}{C \cdot L \cdot \rho}} = \sqrt{\frac{E_{\text{final}}}{\rho \cdot \xi^2}} \]

<table>
<thead>
<tr>
<th>length of pipe in m</th>
<th>natural frequency 2nd order system</th>
<th>rise time in ms</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>4945</td>
<td>787</td>
</tr>
<tr>
<td>0.4</td>
<td>2473</td>
<td>394</td>
</tr>
<tr>
<td>0.6</td>
<td>1648</td>
<td>262</td>
</tr>
<tr>
<td>0.8</td>
<td>1236</td>
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<td>105</td>
</tr>
<tr>
<td>2</td>
<td>495</td>
<td>79</td>
</tr>
</tbody>
</table>

The pipe is a low pass filter (dynamic resistor) between cylinder and HROPVI.
The rise time of the pipe effects the response of the overload protection negatively.

**Figure 17.** Above the pipe model (inductivity and capacity of fluid). Below natural frequency of different pipe lengths. [8]

**Figure 18.** Use recommendations for cartridge valves. Thumb rule: in order to prevent cavitation damage from occurring, the flow should not exceed a mean velocity of 30 m/s in valve manifold ports. Saturation effects will occur as well. [8]
4. SIMULATION DATA AND RESULTS

The system overload was simulated by forcing the loading cylinder to move fixed time 5 ms with constant velocity 1 mm/ms (Figure 19). The HROPV model with simulation data is shown in Figure 20. The results given by the model can be seen in Figure 21 and 22. Figure 23 shows clearly the difference between standard relief valve and HROPV.

![Figure 19. Load simulation model. Piston is forced to move the time 5 ms with constant velocity 1 mm/ms. The piston diameter is 200 mm (forced flow 1885 lpm). This leads to pressure rise velocities depending on hydraulic capacity $C_H$ (piston position).](image)

![Figure 20. The HROPV model (ITI Sim). Valve size is NG32 (rated flow 1100 lpm with 5 bar pressure difference, max. flow (saturation) is 1500 lpm. Operating pressure before overload is 100 bar, no pipe between HROPV and cylinder, cylinder mounting is stiff.](image)
Figure 21. Simulation results. Above the cylinder piston position, spool position (green) and spool velocity (blue). In the middle, the flow through the main valve (blue) and cylinder piston position. Bottom the pressure behaviour inside the cylinder (blue). The pressure inside cylinder changes 100->200 bar in 1.25 ms and the valve begins to open. The pressure reaches maximum value 265 bar when the piston movement stops. The pilot chamber volume was 120 cm³, main stage overlap was 0.5 mm, orifice diameter 1.7 mm and the spring rating was 4 bar. Pressure rise velocity is 80 000 bar/s. /8/
Figure 22. Comparison of size NG32 (blue) and 40, pressure inside the cylinder during overload. The rated flow of size 40 is 1700 lpm with dp=5bar, max. flow (saturation) is 2300 lpm. /8/

Figure 23. Comparison between pressure relief valve (red) and HROPV. Pressure rise velocity is 25000 bar/s. /8/
5. HROPV TESTS

The chapter presents:

- test arrangements, test devices, measuring system
- results with different variables
- comparing accumulators to HROPV in impact damping

5.1. Test arrangements

The mechanical construction of impact test device was based on pendulum (Figure 24) where the weight and height of falling mass were adjustable. Test valve (NG32) in test manifold with control cover can be seen in Figure 25. The manifold was mounted directly to the cylinder port. The important variable, size of pilot chamber orifice, was made changeable by 3/2-way valve (Fig. 25) for test purpose. This was not the optimal choice. In standard valve, the orifice will be in the bore of main spool. Activating the valve V2 solenoid (orifice diameter changes for instance from 0.6 mm to 1.3 mm) did had nearly no effect on the opening time. Later, the valve was used only in activated position where the restriction of valve is lower. In contrast with real application, the cylinder piston is not preloaded by mechanical spring forces (mechanical elasticities, shaft bending, elasticity of frames). The piston has only constant hydraulic loading in the moment of impact. Photos of test unit are shown in Figure 26. The quantities recorded in measurements are presented in Table 1.

Two measuring systems were utilized in measurements: Hydac HMG3000 data recorder with 0.2 ms sampling rate and pressure transducers about 2 kHz frequency band. The other system was Hioki 9335 wave recorder with 1 μs sampling rate and transducers with 400 kHz frequency band (dynamic pressure transducers). Cylinder piston position was measured by laser transducer and the valve spool position by LVDT transducer that showed not to be able to follow the changes quick enough.

5.2. Results with different variables

The variables in measurements affecting pressure increase velocity are:

- weight and height of falling mass
- the basement properties (elasticity)
- the stiffness of hydraulic system
- distance between cylinder and HROPV (part of stiffness)
- the static pressure level before impact
- orifice diameter in RC circuit

Without any damping element between the cylinder rod and the falling mass, the pressure increase velocity becomes very high. Aluminium plate or thick steel plate with rubber plates have been used between the rod and mass (Fig. 26) to affect basement stiffness.

One example of several tables presenting test results with different variables is shown in Table 2. Main results can be summarized to items:

1. Pilot orifice size had only slight effect on the valve opening. The complicated pilot line bores in the cover affected behaviour. The right place for orifice is in the bore of main valve.
2. The pressure increase velocity triggering the valve is rough to adjust and the valve remains closed only with small pressure rise velocities 1000...5000 bar/s (Fig. 29). Low basement stiffness leads to low pressure increase velocity. When variables weight, height, and basement properties remain constant between measurements, the system natural pressure response behaviour is constant and only the pilot orifice size defines the valve opening moment.
The smaller the orifice, the sooner the pressure difference opening the valve will develop and the smaller is the time to develop peak pressure value. This behaviour was not always logical. The valve opened also without orifice and valve V1 activated. However, logical was that with very low basement stiffness and with bigger orifice (1.5 mm), the valve didn’t open but opened with small orifice. Pressure rise velocities higher than 5000 bar/s opened always the valve but the orifice did had some effect on the opening moment (Fig. 27 and 28). Pressure rise velocities in different test situations were 1000...250 000 bar/s. Maximum peak value was about 250 bar measured with 200 kg mass and height 2 (Table 2). (actually the highest measured values were not catched because Hioki system was not in use when measuring them).

3. The valve fulfills the quick opening (1...3 ms) target.
4. Concerning measuring arrangements, the slower Hydac system (Fig. 28) could catch nearly the same peak value as quicker Hioki system but with time lag. The LVDT transducer measuring spool position was not quick enough.
5. Moving the cylinder piston to extended position (piston comes to end) will sometimes open the valve.
6. Noticed that the edge 3 of main valve (Fig. 12) was not sharp and contact area is bigger than planned.
7. The valve will open also when valve V1 is activated (Figure 25). Restrictions in cover bores will cause the pressure difference between the main and pilot pressure causing the opening.
8. The pipe (Ø 35x2.5, length 0.8 m, Table 3) between the cylinder and valve causes time difference between the peak measured at the cylinder and measured at the valve. The peak value is higher at cylinder. The peak length with pipe is also higher. In real system where the piston is like loaded spring, the effect of pipe is bigger when also the compressed oil has to be released from cylinder.
9. In real systems, the construction under the loading cylinder is not stiff and more oil flow is needed to unload the system. This affects the sizing of the main valve.
10. The length of pressure peak is normally 1...3 ms (Table 2). In this time, the pressure is on the original level as before the impact and falls down to tank pressure.
11. The accumulator in parallel with HROPV decreases radically the pressure peak and pressure increase velocity especially with bigger mass and bigger falling height (Table 2).

Figure 24. The impact test device with simplified geometrical drawing. The falling height 1 and 2 have been in use.
Figure 25. HROPV in test manifold with test cover. The valve V1 connects the P port pressure to spring chamber. The choice between orifice K1 and K2 is done by the valve V2. In standard valve, the orifice will be inside the main spool. The pilot circuit volume was fixed.

Figure 26. Above the mass failed against the cylinder. Yellow parts are pendulum arms. Above right is the valve manifold assembled directly in cylinder port. Dynamic pressure transducers are the blue wired parts. Hydac transducer is on the right side of block. Below is damaged aluminium plate on the piston.
Table 1. The recorded variables and transducers

<table>
<thead>
<tr>
<th>Number</th>
<th>Target</th>
<th>Transducer</th>
<th>Channel</th>
<th>Range</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Cylinder pressure</td>
<td>Hydac sensor</td>
<td>CH1</td>
<td>0...400 bar/0.5...4.5 V</td>
<td>2 kHz frequency band</td>
</tr>
<tr>
<td>2</td>
<td>Cylinder pressure</td>
<td>Dynamic sensor</td>
<td>CH3</td>
<td>5V=345 bar, 69 bar/V</td>
<td>400 kHz frequency band</td>
</tr>
<tr>
<td>3</td>
<td>Valve pilot pressure</td>
<td>Hydac sensor</td>
<td>CH2</td>
<td>0...160 bar/4...20 mA</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Valve pilot pressure</td>
<td>Dynamic sensor</td>
<td>CH6</td>
<td>5V=345 bar, 69 bar/V</td>
<td>400 kHz frequency band</td>
</tr>
<tr>
<td>5</td>
<td>Valve tank pressure</td>
<td>Dynamic sensor</td>
<td>CH5</td>
<td>5V=345 bar, 69 bar/V</td>
<td>400 kHz frequency band</td>
</tr>
<tr>
<td>6</td>
<td>Cylinder piston movement</td>
<td>Laser</td>
<td>CH5</td>
<td>± 100 mm/±5 V</td>
<td>either 5 or 6 in CH5</td>
</tr>
<tr>
<td>7</td>
<td>Valve spool position</td>
<td>LVDT</td>
<td>CH4</td>
<td>50 mm/6.1 V</td>
<td></td>
</tr>
</tbody>
</table>

Table 2. The effect of variables to maximum peak pressure, peak length and pressure change velocity. Low basement stiffness and bigger orifice does not open the valve (Impact 4, measurement number) but the valve opens when orifice is smaller (Impact 3). When stiffness is bigger, the valve will open despite the orifice size (Impact 1 and 5, Impact 7 and 10). The effect of accumulator in parallel with HROPV can be seen by comparing Impact 10, 11 and 12.

<table>
<thead>
<tr>
<th>Measurement number</th>
<th>Max. peak [bar]</th>
<th>Static level [bar]</th>
<th>Peak length [mm]</th>
<th>Cyl. piston movement [mm]</th>
<th>Spool stroke [mm]</th>
<th>Pressure change velocity [bar/s]</th>
<th>Figure number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impact 1</td>
<td>174</td>
<td>50</td>
<td>1.4</td>
<td>1</td>
<td>19.5</td>
<td>118000</td>
<td>1</td>
</tr>
<tr>
<td>Impact 2</td>
<td>220</td>
<td>50</td>
<td>1.4</td>
<td>1.7</td>
<td>19.8</td>
<td>200000</td>
<td>2</td>
</tr>
<tr>
<td>Impact 3</td>
<td>85</td>
<td>50</td>
<td>2.4</td>
<td>1.4</td>
<td>19.9</td>
<td>150000</td>
<td>3</td>
</tr>
<tr>
<td>Impact 4</td>
<td>60</td>
<td>50</td>
<td>10</td>
<td>1.4</td>
<td>1</td>
<td>1000</td>
<td>4</td>
</tr>
<tr>
<td>Impact 5</td>
<td>146</td>
<td>50</td>
<td>1.4</td>
<td>1.4</td>
<td>19.5</td>
<td>94000</td>
<td>5</td>
</tr>
<tr>
<td>Impact 6</td>
<td>135</td>
<td>25</td>
<td>2</td>
<td>1.4</td>
<td>18</td>
<td>136000</td>
<td>6</td>
</tr>
<tr>
<td>Impact 7</td>
<td>190</td>
<td>50</td>
<td>1.4</td>
<td>1.9</td>
<td>18</td>
<td>160000</td>
<td>7</td>
</tr>
<tr>
<td>Impact 8</td>
<td>75</td>
<td>50</td>
<td>10</td>
<td>1.5</td>
<td>1</td>
<td>5000</td>
<td>8</td>
</tr>
<tr>
<td>Impact 9</td>
<td>76</td>
<td>50</td>
<td>3</td>
<td>1.4</td>
<td>20</td>
<td>10000</td>
<td>9</td>
</tr>
<tr>
<td>Impact 10</td>
<td>195</td>
<td>50</td>
<td>1.8</td>
<td>1.6</td>
<td>19</td>
<td>145000</td>
<td>10</td>
</tr>
<tr>
<td>Impact 11</td>
<td>253</td>
<td>50</td>
<td>2</td>
<td>2.5</td>
<td>21</td>
<td>250000</td>
<td>11</td>
</tr>
<tr>
<td>Impact 12</td>
<td>130</td>
<td>50</td>
<td>2.5</td>
<td>2</td>
<td>19</td>
<td>80000</td>
<td>12</td>
</tr>
<tr>
<td>Impact 13</td>
<td>96</td>
<td>50</td>
<td>3.8</td>
<td>2</td>
<td>10</td>
<td>15700</td>
<td>13</td>
</tr>
<tr>
<td>Impact 14</td>
<td>115</td>
<td>50</td>
<td>3</td>
<td>2.3</td>
<td>19</td>
<td>30500</td>
<td>14</td>
</tr>
<tr>
<td>Impact 15</td>
<td>105.4</td>
<td>50</td>
<td>2.2</td>
<td>2.1</td>
<td>10</td>
<td>105000</td>
<td>15</td>
</tr>
<tr>
<td>Impact 16</td>
<td>146</td>
<td>50</td>
<td>2</td>
<td>2</td>
<td>10</td>
<td>120000</td>
<td>16</td>
</tr>
</tbody>
</table>

Measurement number | Remarks                                                                 |
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Impact 1</td>
<td>mass 100 kg / height 1 / orifice 0.6 mm / alum. plate</td>
</tr>
<tr>
<td>Impact 2</td>
<td>100 kg / height 2 / 0.6 mm / alum. plate</td>
</tr>
<tr>
<td>Impact 3</td>
<td>100 kg / height 1 / 1.2 mm / steel + rubber plate</td>
</tr>
<tr>
<td>Impact 4</td>
<td>100 kg / height 1 / 1.2 mm / steel + rubber, does not open</td>
</tr>
<tr>
<td>Impact 5</td>
<td>100 kg / height 1 / 1.2 mm / alum. plate</td>
</tr>
<tr>
<td>Impact 6</td>
<td>100 kg / height 1 / 1.2 mm / alum. plate</td>
</tr>
<tr>
<td>Impact 7</td>
<td>200 kg / height 1 / 1.2 mm / alum. plate</td>
</tr>
<tr>
<td>Impact 8</td>
<td>200 kg / height 1 / 1.2 mm / steel + rubber, does not open</td>
</tr>
<tr>
<td>Impact 9</td>
<td>200 kg / height 1 / 0.6 mm / steel + rubber</td>
</tr>
<tr>
<td>Impact 10</td>
<td>200 kg / height 1 / 0.8 mm / alum.</td>
</tr>
<tr>
<td>Impact 11</td>
<td>200 kg / height 2 / 0.6 mm / alum.</td>
</tr>
<tr>
<td>Impact 12</td>
<td>200 kg / height 2 / 0.6 mm / alum.</td>
</tr>
<tr>
<td>Impact 13</td>
<td>200 kg / height 2 / 0.6 mm / 1.4 l accum. chamber (prechar. 60 bar)</td>
</tr>
<tr>
<td>Impact 14</td>
<td>200 kg / height 2 / 0.6 mm / 1.4 l accum. / V1 activ.</td>
</tr>
<tr>
<td>Impact 15</td>
<td>200 kg / height 1 / 0.5 mm / 1.4 l accum. / V1 activ.</td>
</tr>
<tr>
<td>Impact 16</td>
<td>200 kg / height 1 / without orifice, V1 activ. / alum. / 1.4 l accum.</td>
</tr>
</tbody>
</table>
Table 3. The effect of bended pipe Ø35x2,5 (length 0,8mm) between the cylinder port and HROPV. Compare measurement Impact 2 to Impact 1 in Table 2 and Impact 5 to Impact 2 in Table 2.

<table>
<thead>
<tr>
<th>Measurement number</th>
<th>Max. peak [bar]</th>
<th>Static level [bar]</th>
<th>Peak length [ms]</th>
<th>Cyl. piston move [mm]</th>
<th>Spool stroke [mm]</th>
<th>Pressure change velocity</th>
<th>Remarks</th>
<th>Figure number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impact 2</td>
<td>193</td>
<td>50</td>
<td>2,49</td>
<td>1,3</td>
<td>17</td>
<td>143000</td>
<td>100 kg, height 1,0 mm, alum.</td>
<td>18</td>
</tr>
<tr>
<td>Impact 5</td>
<td>270</td>
<td>50</td>
<td>2,6</td>
<td>2,1</td>
<td>17</td>
<td>220000</td>
<td>100 kg, height 2,0 mm, alum.</td>
<td>17</td>
</tr>
<tr>
<td>Impact 6</td>
<td>90</td>
<td>50</td>
<td>3,4</td>
<td>1,4</td>
<td>16</td>
<td>160000</td>
<td>100 kg, height 2,0 mm, steel + rubber</td>
<td></td>
</tr>
</tbody>
</table>

Figure 27. Measurement with Hioki recorder. Pilot orifice 0.6 mm, aluminium plate on the cylinder piston. Dynamic transducer (red, CH3) and Hydac transducer (green, CH1) are measuring cylinder pressure. Hydac notices the change about 0.4 ms later than the dynamic transducer. Pilot pressure dynamic transducer (dark blue, CH6) begins to change some 0.5 ms later than the cylinder pressure. The peak measured by Hydac is 90.5 and with dynamic transducer 98 bar. The peak length is about 1 ms. The same measurement with pilot orifice 1.5 mm causes the pilot to change more but only slightly. The spool position transducer (CH4) is not quick enough showing the opening to begin after the pressure has gone down.
Figure 28. Hydac recording. Orifice 1.5 mm, mass 100 kg, height 1, aluminium plate on piston. Pressure increase velocity is 97400 bar/s (red), peak pressure 150 bar, peak length 1.4 ms, piston movement (yellow) 4, 3 mm, valve spool stroke 19.7 mm, spool velocity 4.7 m/s. Pilot pressure is green.

Figure 29. The valve does not open. Orifice is 1.5 mm, two rubber plates on the cylinder piston, pressure increase velocity is 1704 bar/s, pressure peak value 59 bar, peak length 7.8 ms, cylinder piston movement is 1.3 mm.
5.3. Comparing accumulators to HROPV in impact damping

The damping effect of membrane accumulators (0.16 l and 1.4 l) was compared to HROPV with gas precharge pressure (60 bar) over the static loading pressure (50 bar) and with precharge (30 bar) smaller than the static level. The accumulators were connected directly to cylinder port. As reference worked the measurement without accumulator and HROPV and only aluminium plate between the hitting mass and cylinder rod (Fig. 30a). Recording was done with Hydac system that could not catch the absolute peak values but the results are relative comparable. The pressure curves from part of the measurements are shown in Figure 30. The results can be summarized to following comments:

- the "overprecharged" accumulator near the cylinder decreases the max. pressure peak and the pressure increase velocity compared to a cylinder without accumulator. The cylinder rod movement does not change much (with or without accumulator).
- naturally with bigger accumulator, the peak is smaller (the pressure increase velocity nearly the same). The static pressure level drops remarkably after the impact (Fig. 30c) with bigger accumulator and lower falling height; with smaller falling height, the impact "goes better" inside the accumulator
- with "underprecharged" accumulator (Figure 30d), the damping is very good; however, the stiffness of loading system will be low and vibration exist after the impact
- the HROPV releases the pressure very quickly and unloads the system pressure to tank pressure. The maximum peak is higher but very short (1...3 ms) compared to accumulators (Figure 30e).

The tradition to use "wad" damping accumulators in nip loading cylinders has arguments. The damping devices have bigger importance when the wad is "longer" and not only short impact as in test device. With HROPV, the peak value is higher but the peak time is shorter. Adjustable precharge pressure can be one alternative in systems where the impacts exist only in certain working conditions and you don’t need to be prepared to impacts continuously.

6. CONCLUSION

Principally the valve was working and fulfilled quick pressure release target. The RC-circuit design in pilot section is crucial. The set value for pressure rise velocity was low compared to target value 10000 bar/s. The adjustability by changing the pilot orifice was poor mainly because the orifice location in the cover and cover construction. The modelling does not reveal the details for instance in flow restrictions in pilot circuit. The possible differences in bulk modulus between modelling and real system can influence RC circuit dimensioning accuracy.

In test device, the nominal pressure rise velocity was higher than original target value and special arrangements (aluminium and rubber plates) were needed to reduce the value. In real application with complicated mechanical structure, the pressure response behaviour is difficult to forecast and possibility to adjust the rise velocity triggering the valve is important. The target response time 1...3 ms to release loading pressure was reached.

The options to be included in valve were in discussions; for instance the valve should remain closed if wanted, it should be able to close the valve without slowing the whole system (cutting supply pressure), the valve should open by external pilot signal, the valve can be utilized as pressure rise sensing pilot valve for several main valves.
The tests were done at the beginning of 2008. At that time there was also some other alternatives to prevent impact damages; for instance increase the damping by modifying the shoe (Figure 2), utilize accumulators inside the roll. The most promising alternative in all respects is not always absolutely clear. The economical recession in 2008 changed many plans and put developments to wait future.

In the history of valve development we live today in phase where research and design work has mainly moved from fluid technology based valve design to software technology based design. Though the direction is natural, the basics is in fluid technology and we should not totally forget the alternatives the natural working pure fluid components can offer.

Figure 30 (a,b). Cylinder pressure (red) and rod position (blue) curves with accumulator and with mass 100 kg and falling height 2. The reference measurement without any damping can be seen in Fig. 30a. The effect of accumulator 0.16 l (precharge 60 bar) is shown in Fig. 30b.
Figure 30 (c,d). The effect of bigger accumulator (1.4 l) is presented in Fig. 30c. The effect of lower precharge pressure (30 bar) with accumulator 1.4 l is seen in Fig. 30d.
Figure 30 (e,f). The pressure peak when using the HROPV is presented in Fig. 30e. In Figure 30f, the material between the rod and the hitting mass is steel+thick rubber instead of aluminium plate.

REFERENCES


POWER PLANT FUEL VALVE CHARACTERISTICS
CONSIDERING HYDRODYNAMIC FORCE

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ABSTRACT
The work presents a mathematical model of power plant differential valve with exact regulation of fuel consumption rate during starting and stopping time. The throttle characteristics and hydrodynamic force affecting the valve head from the side of liquid flow have been calculated with the ANSYS software package. The valve mathematical model has been implemented with the help of the MATLAB/Simulink software package. Transient processes in the valve appearing at its opening and closing have been calculated, influence of hydrodynamic force and design parameters on the transient process quality has been studied.

KEYWORDS: differential valve, transient process, hydrodynamic force, mathematical model, Matlab/Simulink, ANSYS.

1. INTRODUCTION
In fuel pump-consumer feedlines of power plant pneumohydraulic systems the shutoff valves are installed for working medium supply directly at starting time and for shutoff at stop time. The advantage of the differential valve is that the pump-valve system does not require any additional control actions to open it at the start of operation and to close it at the end of pump operation. One of the requirements for a differential valve is to provide the preset response time. This is especially important considering the mainline length from the pump to the consumer and its hydraulic characteristics. There are calculation procedures for valve dynamic and static characteristics, e.g. [1, 2, 3]. However, a different valve diagram is considered here and hydraulic force affecting the valve head is not taken into account. The work presented by the authors eliminates the problem using the developed mathematical model of differential valve with the account of hydrodynamic force, necessary calculations with the ANSYS software package and comparison of calculation results with experimental data.
2. MATHEMATICAL MODEL

The diagram of the analysed valve in operation is presented in Figure 1. The following assumptions are accepted for equations of the valve moving part (valve head) motion: bulk compliance of the valve case and head are not taken into account due to their smallness; hydraulic losses in the throttling elements are considered according to quasistationary model; no dry friction in the moving elements; leakage between the valve moving parts is negligible; hydraulic losses in the feedback line is considered according to the liquid flow quasistationary model.

Taking into account the accepted allowances we can develop the valve algebraic and differential equations and by solving them we can obtain the valve static and dynamic characteristics [4, 5, 6].

While analyzing the valve dynamic characteristics the differential equation of the valve head motion is of special interest [7]:

\[
M \cdot \frac{d^2 x}{dt^2} + \lambda_{\text{frie}} \cdot \frac{dx}{dt} + \gamma_{\text{spr}} \cdot x + N_{\text{dry frie}} \cdot \text{sign}\left(\frac{dx}{dt}\right) = F_{\text{flow}} - F_{\text{spr}} - A_{\text{int1}} \cdot p_{\text{int1}} - A_{\text{int2}} \cdot p_{\text{int2}},
\]

where \( M \) – reduced mass of the valve movable part, kg; \( x \) – valve axial coordinate, m; \( t \) – time, s; \( \lambda_{\text{frie}} = \mu \cdot l_{\text{seal}} \cdot b_{\text{seal}} / \delta_{\text{seal}} \) – coefficient considering viscous friction; \( \mu \) – dynamic liquid viscosity, Pa·s; \( l_{\text{seal}} = \pi \cdot D \) – sealing gasket circumference, m; \( D \) – sealing gasket diameter, m; \( b_{\text{seal}} \) – sealing gasket width, m; \( \delta_{\text{seal}} \) – gap between the sealing gasket and the case, m; \( \gamma_{\text{spr}} \) – total spring stiffness, N/m; \( N_{\text{dry frie}} \) – the force of dry friction, N; \( F_{\text{flow}} \) – hydraulic force acting on the valve head from the valve flow part, N; \( F_{\text{spr}} \) – total preliminary spring tension, N; \( A_{\text{int1}} = \pi \cdot D^2 / 4 - \pi \cdot d^2 / 4 \) – area of cavity B under the valve head, \( m^2 \); \( p_{\text{int1}} \) – pressure in cavity B under the valve head, Pa; \( A_{\text{int2}} = \pi \cdot d^2 / 4 \) – area of cavity C under the valve head, \( m^2 \); \( p_{\text{int2}} \) – pressure in cavity C under the valve head, Pa.
The hydraulic force occurs as a result of the impact of liquid flow passing through the valve on the valve head and represents the sum of hydrostatic and hydrodynamic forces [8]:

\[ F_{\text{flow}} = c_{\text{hydrostat}} \cdot p_{\text{out}} \cdot A_p + c_{\text{hydrodyn}} \cdot A_p \cdot \rho \cdot v^2 / 2 , \]

where \( c_{\text{hydrostat}} \) – hydrostatic force coefficient; \( p_{\text{out}} \) – the valve outlet pressure, \( \text{Pa} \); \( A_p = \pi \cdot D^2 / 4 \) – the valve head area, \( m^2 \); \( c_{\text{hydrodyn}} \) – hydrodynamic force coefficient; \( \rho \) – liquid density, \( \text{kg} / \text{m}^3 \); \( v \) – mean velocity of liquid flow near the valve head, \( \text{m/s} \).

Coefficient \( c_{\text{hydrostat}} \) takes into account distribution of static pressure over the valve head surface. Coefficient \( c_{\text{hydrodyn}} \) is identical to the coefficients derived in hydromechanics for definition of forces acting on a streamlined body. Coefficients \( c_{\text{hydrostat}} \) and \( c_{\text{hydrodyn}} \) depend on the structure of liquid flow around the valve head.

Average liquid flow rate near the valve head is defined by the liquid flow through the valve:

\[ v = G_x / (\rho \cdot \mu \cdot A_x) , \]

where \( G_x \) – mass flow rate through the valve flow part, \( \text{kg/s} \); \( \mu \cdot A_x \) – average effective area of the flow part near the valve head, \( m^2 \).

Equations describing liquid flow in the cavities under the valve head when the valve head is in motion:

\[ A_d \cdot \rho \cdot \frac{dx}{dt} = \frac{V_{\text{int}2}}{c^2} \cdot \frac{dp_{\text{int}2}}{dt} + \mu_{\text{th}} \cdot A_{\text{th}} \cdot \sqrt{2} \cdot \rho \cdot (p_{\text{int}2} - p_{\text{int}1}) , \]

\[ A_p \cdot \rho \cdot \frac{dx}{dt} + \mu_{\text{th}} \cdot A_{\text{th}} \cdot \sqrt{2} \cdot \rho \cdot (p_{\text{int}2} - p_{\text{int}1}) = \frac{V_{\text{int}1}}{c^2} \cdot \frac{dp_{\text{int}1}}{dt} + G_{\text{pipe}} , \]

\[ L_{\text{pipe}} \cdot \frac{dG_{\text{pipe}}}{dt} + R_{\text{pipe}} \cdot G_{\text{pipe}} = p_{\text{int}1} - p_{\text{in pump}} , \]

where \( V_{\text{int}2} \) – volume of cavity C under the valve head, \( m^3 \); \( c \) – velocity of sound in liquid, \( \text{m/s} \); \( \mu_{\text{th}} \cdot A_{\text{th}} \) – total effective area of orifices, \( m^2 \); \( V_{\text{int}1} \) – volume of cavity B under the valve head, \( m^3 \); \( G_{\text{pipe}} \) – liquid flow rate through the pipeline, \( \text{kg/s} \); \( L_{\text{pipe}} = l_{\text{pipe}} / A_{\text{pipe}} \) – pipeline specific acoustic inductance, \( 1/\text{m} \); \( l_{\text{pipe}} \) – pipeline length, \( \text{m} \); \( A_{\text{pipe}} \) – pipeline flow passage area, \( m^2 \); \( R_{\text{pipe}} = 128 \cdot v \cdot l_{\text{pipe}} / (\pi \cdot d_{\text{pipe}}^4) \) – pipeline hydraulic resistance, \( 1/(\text{m} \cdot \text{s}) \); \( v \) – liquid kinematic viscosity, \( \text{m/s} \); \( d_{\text{pipe}} \) – pipeline flow passage diameter, \( \text{m} \); \( p_{\text{in pump}} \) – pre-pump pressure, \( \text{Pa} \).

Liquid flow rate through the valve flow part is defined by the dependency:

\[ G_x = \mu_x \cdot A_x \cdot \sqrt{2} \cdot \rho \cdot (p_{\text{in}} - p_{\text{out}}) , \]

where \( \mu_x \cdot A_x \) – valve effective area, \( m^2 \); \( p_{\text{in}} \) – valve inlet pressure, \( \text{Pa} \).

As a result, the system of differential and algebraic equations (1)…(7) represents a mathematical model of the valve on the basis of which we can calculate its dynamic characteristics and select the valve design parameters to meet specified requirements for the unit.
3. CALCULATION OF HYDRAULIC FORCE WITH THE ANSYS/FLUENT PROGRAM

Main difficulty in the valve mathematical model is the analytic description of hydraulic force acting on the valve head. It is due to the fact that hydraulic force depends on the flow structure near the valve head. Furthermore, the channel formed by the valve head profile and the case in real constructions can have a complex shape that makes calculations of hydraulic force acting on the valve head more complicated. Let's consider calculation of the valve hydraulic force and throttle characteristics using computing simulation. For this purpose a geometrical model of the valve has been constructed with the NX program package (Figure 2, a) and a hydraulic domain of the valve flow part has been constructed on its basis taking into account requirements for the design model [9, 10, 11].

![Sectional drawing of the valve (a) and hydraulic domain of its flow part (b)](image)

Figure 2. Sectional drawing of the valve (a) and hydraulic domain of its flow part (b)

The finite-element model (Figure 2, b) and hydrodynamic parameters calculation have been performed with the help of the ANSYS software package. A sample of calculated parameters is presented in Figure 3 - 5.

![Calculated velocity fields (a), m/s and pressures (b), Pa with the fully open valve](image)

Figure 3. Calculated velocity fields (a), m/s and pressures (b), Pa with the fully open valve
Figure 4.  Calculated velocity fields (a), m/s and pressures (b), Pa with the fully open or 1/3 open valve
1 – reverse-flow area.

Figure 5.  Bulk concentration of working medium fumes with the 1/3 open valve and minimum flow rate

On the basis of calculated parameters the throttle characteristics of the valve have been plotted for a fully open position and for two interpositions. In Figure 6 the dots show the valve experimental throttle characteristics, and the solid line shows the calculated parameters. As can be seen from the graphs, the calculated and experimental characteristics for a fully open and 2/3 open position of the valve are completely the same that proves the adequacy of the calculated model. Non-coincidence of calculated and experimental characteristics for 1/3 open position of the valve is explained by the cavitation (Figure 5) and secondary losses in the flow part behind the valve (Figure 4-a, element 1), absent in case of the fully open valve.
As the calculated throttle characteristics mentioned above are adequate to the real process, then the same conclusion can be done with reference to the other calculated parameters, including hydraulic force. Calculated dependencies of hydraulic force on the valve pressure drop for five positions are shown in Figure 7.

Approximating dependencies \( F_{\text{flow}} = f(\Delta p, x) \) (Figure 7), \( G_x = f(\Delta p, x) \) (Figure 8), enables to consider complex nature of liquid flow through the flow part while calculating its dynamic characteristics according to the above equations \((1)\ldots(7)\) with the numerical technique of the MATLAB/SIMULINK software package.
4. SIMULATION RESULTS

The valve transient characteristics are calculated with the account of pressure change at the valve inlet as a control action. The calculation has been done for different orifice diameters (Figure 1, element 7).

The valve is closed in the initial position. When pressure delivered to the inlet is higher than the opening pressure, the valve opens and the working medium flows to the consumer. The calculation has been performed with the stepwise increasing valve inlet pressure from 0 MPa to 20 MPa with the orifice diameters 1, 2 and 3 mm. The transient characteristics are shown in Figures 9…11.

![Figure 9. Transient processes of the valve movable element position variation](image)

![Figure 10. Transient processes of mass flow rate variation](image)

![Figure 11. Transient processes of pressure drop variation](image)
The analysis of the graphs presented in Figures 9…11 shows that the change of the orifice diameter from 3 to 1 mm causes the increase of the valve opening time from 6.5 ms to 11.8 ms. However, the peak mass flow rate at the outlet changes from 1.1 ms to 1.2 ms and the steady value of the mass flow rate occurs in 3 ms. The pressure drop transient process has a slight stratification. Therefore, by changing the orifice diameter it is possible to perform fine adjustment of the valve opening time within a fairly narrow range. For a wider range of the valve opening time control we can propose to change the valve springs and, if it is impossible, to install throttling orifice in the feedback pipeline. For this purpose the mathematical model is slightly modified with the account of the throttling orifice in the pipeline.

5. CONCLUSION

The developed mathematical model of the valve can be used to analyze power plant dynamic characteristics and to forecast a change of its properties with the change of valve parameters, e.g. to solve problems of optimization and to examine the influence of parameters spread in the process of manufacturing.

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FREQUENCY RESPONSE CORRECTION OF LAUNCH VEHICLE FUEL LINE

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ABSTRACT

Design engineers of liquid multistage launch vehicles (LV) always take into account the probability of longitudinal instability called Pogo-effect. One of the traditional measures to provide LV longitudinal stability is the frequency response correction of fuel lines with the use of dampening units. Notwithstanding the developed design solutions to provide LV longitudinal stability, the open literature gives little attention to calculation methods for correcting devices and to the analysis of frequency response of fuel lines with a dampener.

The report presents a mathematical model developed on the basis of the energy equation for gas and liquid cavities of the dampener, and of the equations of continuity and equilibrium of movable parts. This mathematical model allows estimating the influence of different design parameters on its dynamic properties. In particular, frequency dependent acoustic admittance of the dampener has been obtained with the use of MatLab/Simulink software package. Frequency response of the fuel-supplying mainline with a connected dampener clearly demonstrating the reduction of fuel line natural frequency has also been obtained. The method of frequency response calculation and parameter selection for a LV longitudinal stability dampener has been proposed herein.

KEYWORDS: pogo suppression device, fuel line, launch vehicle, mathematical model, dynamic properties, acoustic admittance, frequency response, MatLab/Simulink.

1. INTRODUCTION

Low frequency oscillations of launch vehicle bodies along a longitudinal axis (Pogo effect) appear as a result of interactions between oscillations of the launch vehicle elastic body and oscillations of liquid in the fuel lines and oscillations of the engine thrust [1, 2, 3]. Elastic oscillations of the vehicle body lead to pressure oscillations of the fuel tank components which are transmitted via transfer lines to the engine and cause its thrust pulsations. The last mentioned pulsations acting on the elastic body, in their turn, dampen or amplify structural oscillations depending on amplitude and phase relations in this closed loop. Analysis of a large number of works dedicated to the research and control of the vehicle longitudinal instability [1, 2, 4, 5, 6, 7, 8] shows that an effective way to suppress Pogo effect is the correction of frequency response of fuel lines realized by introduction of the element with elastic flexibility which will help to lower resonance frequency of
the fuel line in order to exclude its coalescence with the LV body frequency. Among the modern works on the research of correcting devices we can point out the work [9] where a simplified analytical model has been developed and the key parameters have been defined not theoretically but on the basis of experimental results. To design a gas damper with required parameters providing stable work of the launch vehicle pneumohydraulic system it is necessary to develop a calculation method of its dynamic characteristics. The research is based on the mathematical modeling of a pogo suppression device with the account of numerous factors affecting its characteristics. This method will allow predicting the pogo suppression device dynamic characteristics in order to find out its malfunction due to engineering changes and technological spread of parameters.

2. MATHEMATICAL MODEL

The pogo suppression device (Figure 1 [10]) is installed in the oxidizer fuel line and designed for correction of its frequency response to increase the LV longitudinal stability margin and to avoid cavitation at the inlet to the centrifugal pump with screw of the liquid propellant engine.

The principle of the dampener action is based on the introduction of local flexibility with a damping element in the oxygen line that enables to lower its resonance frequency and to provide additional liquid inflow at the pump inlet during the liquid-propellant engine starting.

Let's consider the above mentioned dynamic system consisting of the shutoff and control element 8 with the mass \( m \), fixed by the elastic coupling with the stiffness \( J \) (the bellows 5), and linear damping \( D \).
While modeling the pogo suppression device the following allowances have been accepted: working medium in the gas cavity is the ideal gas; pressure drop at the shutoff and control element (in section 1-1) – supercritical; input impedance of the downstream system is equal to zero ($p_a = const$); no heat exchange with the environment through the pogo suppression device walls; the whole mass of oxygen in the liquid cavity of the dampener and the bends is in liquid state.

Assuming that the gas state in the gas cavity is changed according to adiabatic relationship we can write [11]:

$$C \cdot \frac{dp_g}{dt} = G_{g,in} - G_{g,out},$$  \hspace{1cm} (1)

where $C = \frac{V_g}{k \cdot R \cdot T_g}$ - acoustic capacitance; $p_g$ - gas cavity pressure; $G_{g,in}$ - mass flow rate at the gas cavity inlet; $G_{g,out}$ - mass flow rate at the gas cavity outlet; $R$ - gas constant; $V_g$ - gas cavity volume; $T_g$ - gas cavity temperature; $k$ - adiabatic index.

Gas flow rate through the throttling section defined by the shutoff and control element position is expressed by Saint-Venant – Wenzel formulas:

$$G_{g,out} = \mu_{out} \cdot A(x) \cdot p_g \cdot \sqrt{\frac{2}{R \cdot T_g}} \cdot \frac{k}{k-1} \left[ \left( \frac{p_a}{p_g} \right)^{\frac{k}{k-1}} - \left( \frac{p_a}{p_g} \right)^{\frac{k+1}{k}} \right],$$  \hspace{1cm} (2)

with $\frac{p_a}{p_g} > \beta_{critical}$ and

$$G_{g,out} = \mu_{out} \cdot A(x) \cdot p_g \cdot \sqrt{\frac{k}{R \cdot T_g}} \left( \frac{2}{k+1} \right)^{\frac{k+1}{2(k-1)}},$$  \hspace{1cm} (3)

with $\frac{p_a}{p_g} \leq \beta_{critical},$

where $\mu_{out}$ - flow rate coefficient in the gap between the shutoff and control element and the guide 10; $A(x)$ - variable flow area of adjustable automatic gas throttle; $p_a$ - atmospheric pressure; $\beta_{critical}$ - critical pressure ratio.

Flow area $A(x)$ is defined on the basis of design parameters of the adjustable automatic gas throttle. For this research we accept a linear law of area change $A(x)$ according to the position of shutoff and control element.

Mass gas flow rate at the gas cavity inlet is determined by the constant throttle 9 with a supercritical pressure drop and is written similar to the expression (3):

$$G_{in} = \mu_{tr} \cdot A_{tr} \cdot p_{g,in} \cdot \sqrt{\frac{k}{R \cdot T_g}} \left( \frac{2}{k+1} \right)^{\frac{k+1}{2(k-1)}},$$  \hspace{1cm} (4)

where $\mu_{tr}$ - throttle 9 flow rate coefficient; $A_{tr}$ - throttle 9 flow passage area.

Mass gas flow rate caused by the motion of the bellows butt end is defined as follows:

$$G_{g,x} = \frac{p_g}{R \cdot T_g} \cdot A_b \cdot \frac{dx}{dt},$$  \hspace{1cm} (5)

where $x$ - needle movement; $t$ - time, $A_b$ - equivalent area of the bellows butt end.
Equation of the needle equilibrium as a dynamic element with lumped parameters $m, J, D$, we present as [10]:

$$m \frac{d^2x}{dt^2} + D \frac{dx}{dt} + F_{\text{friction}} \cdot \text{sign}(\frac{dx}{dt}) + J \cdot \dot{x} + p_3 \cdot A_b - p_g \cdot A_b + m \cdot g \cdot (1 \pm n_x) = 0,$$

(6)

where $m$ - reduced mass of the needle with bellows; $F_{\text{friction}}$ - dry friction force; $p_3$ - liquid cavity pressure; $g$ - acceleration of gravity; $n_x$ - axial acceleration.

Hereafter, we write main formulas to define work processes in the liquid cavity.

Pressure change in the liquid cavity is connected with the flow rate change of the liquid entering the cavity [10]:

$$\frac{V_3}{\rho_{fl}} \cdot c \cdot \frac{dp_3}{dt} = Q_{fl,x} + Q_{p,3},$$

(7)

where $V_3$ - liquid cavity volume; $\rho_{fl}$ - fluid oxygen density; $c$ – velocity of sound in the medium; $Q_{fl,x}$ - volumetric flow rate of fluid oxygen connected with the motion of the bellows butt end; $Q_{p,3}$ - volumetric flow rate of the liquid in the inlet pipe connection.

Equation of fluid oxygen motion in the supply pipe connection with the account of hydraulic losses takes the form of [10]:

$$p_2 - p_3 - R_{p,con} \cdot \rho_{fl} \cdot Q_{p,3} = \frac{l_{p,con} \cdot \rho_{fl}}{A_{p,con}} \cdot \frac{dQ_{p,3}}{dt},$$

(8)

where $p_2$ - fluid oxygen pressure at the pipe connection inlet; $p_3$ - liquid pressure at the pipe connection outlet (before the bellows butt end); $A_{p,con}$ - pipe connection area; $l_{p,con}$ - pipe connection length; $Q_{p,3}$ - volumetric flow rate in the pipe connection; $R_{p,con} = \frac{128 \cdot \nu \cdot l_{p,con}}{\pi \cdot d_{p,con}^4}$ - pipe connection resistance for laminar flow mode; $\nu$ - fluid oxygen kinematic viscosity; $d_{p,con}$ - pipe connection diameter.

Volumetric liquid flow rate caused by the motion of the bellows butt end is determined similar to the formula (5):

$$Q_{fl,x} = A_b \cdot \frac{dx}{dt}.$$

(9)

The mathematical model for unsteady turbulent flow within distributed parameters has been used for calculation of mainline characteristics [12].

3. RESEARCH ON STATIC AND DYNAMIC CHARACTERISTICS OF POGO SUPPRESSION DEVICE IN MATLAB/SIMULINK SOFTWARE

The obtained system of equations (1) - (9) describes functioning of the pogo suppression device depending on different disturbances (pressure $p_2$ at the pipe connection inlet, pressure $p_{g,in}$ at the constant throttle inlet 9, $n_x$ - axial acceleration). The study of transient characteristics of the dampener has been performed with the numerical technique in the MATLAB/SIMULINK software environment. Figure 2 shows transient processes in the pogo suppression device at the pressure step change at the pipe connection inlet $p_2$ from 0.7 MPa to 0.75 MPa. The pressure increase leads to the movement of the shutoff and control element to the decrease of the adjustable automatic gas throttle. At the same time the gas cavity pressure increases.
Steady-state pressure in the gas cavity is higher than the pressure in the liquid cavity because during the dampener work there is a purging of gas cavity from the constant pressure source.

Figure 2. Transient processes of the shutoff and control element stroke \( x \) and pressure in liquid \( p_3 \) and gas \( p_g \) cavities at the step change of pressure \( p_2 \) at the pipe connection inlet

The pogo suppression device frequency response necessary for analysis of the influence of fuel lines on the LV longitudinal stability has been obtained with the use of the Simulink – Control Design embedded software tool.

The nonlinear small deflection linearization entering into the system of equations in an automatic mode has been performed for calculation of the pogo suppression device frequency response. Amplitude-frequency and phase-frequency responses of the pogo suppression device have been calculated in the form of ratio of complex flow oscillation amplitude at the pogo suppression device inlet to the complex pressure oscillation amplitude. Such ratio of pogo suppression device parameters is called its input acoustic admittance.

Figure 3 presents the modulus and the argument of acoustic admittance or amplitude-frequency and phase-frequency responses of the pogo suppression device. As can be seen from the figure, the dampener resonance frequency is equal to 15.5 Hz and ratio \( |\Delta p_3 / \Delta p_2| \) on this frequency makes \( 6.16 \frac{mL}{s \cdot Pa} \).

Considering the phase response we see that passing through the resonance frequency the ratio argument \( \text{Arg}(\Delta p_3 / \Delta p_2) \) rotates by 180°.
Figure 3. Input acoustic admittance of pogo suppression device

Figure 4 shows dependences of volume change of the liquid, displaced by the bellows in its movement with harmonic oscillations on sub-resonance, resonance and above-resonance frequency.

From the above responses we can see that they represent the hysteresis dependences as for a mechanical oscillation dampener in the form of dependence of the applied force on the deformation. In this case the closed curve area achieves maximum value on the resonance frequency of 15.5 Hz. In this work mode the pogo suppression device possesses maximum property to “absorb” liquid pressure oscillations at the pipe connection inlet. On the frequency of 4 Hz the closed curve area is less than on frequency of 20 Hz. It occurs due to the difference of amplification coefficient values for different frequencies.

Figure 4. Dependence of volume change of the liquid displaced by the bellows in its movement on the pipe connection inlet pressure with harmonic oscillations on sub-resonance 4 Hz, resonance 15.5 Hz and above-resonance 20 Hz frequency
4. INFLUENCE OF POGO SUPPRESSION DEVICE ON FREQUENCY RESPONSE OF FUEL LINE

While modeling the pneumohydraulic system consisting of the fuel line and the pogo suppression device the pressure \( p_2 \) at the fuel line inlet and the pressure \( p_4 \) at the opposite end of the line with the connected engine have been specified as boundary disturbance effects. Impact of the installed pogo suppression device on the values of the line amplification factor and natural frequency has been considered for calculation of frequency response of the fuel line consisting of two segments with different lengths.

The modeling results show that the installation of the pogo suppression device allows reducing the amplification factor more than 3 times and lowering the resonance frequency from 13.6 Hz to 3.1 Hz.

![Figure 5. Amplitude-frequency and phase-frequency response of the fuel line in frequency range up to 30 Hz](image)

Thus, introduction of local flexibility allows to avoid the risk of resonance frequency coalescence of fuel line and of elastic body, and so to increase the LV longitudinal stability margin.

5. CONCLUSION

The developed mathematical model of the pogo suppression device can be used to predict deviations in its characteristics caused by technological spread of the unit element parameters or by the defect appearing in its construction, as well as to research its influence on frequency response of the fuel line in order to provide the LV longitudinal stability. The developed method of dynamic characteristics calculation and selection of pogo suppression device parameters is an integral constituent of the dynamic characteristics research process of the vehicle longitudinal oscillation loop.

REFERENCES


ABSTRACT

Care for clean and healthy environment should be increasing on daily basis. Different kind of hydraulic fluids are used nowadays. Majority of them are harmful. Use of tap water as hydraulic pressure medium is one of possible solution. The work is based on development and researches of check valve for water hydraulic. The first part of the paper includes background and overview of standard check valves on market. The second part is based on the design and development of a new check valve. On presented check valve we carried out numerical calculations and measurements of pressure drop for two types of hydraulic fluid – tap water and mineral hydraulic oil. There were also performed strength calculations for critical parts of the check valve. In the last part of the work, the comparison and analysis of experimental results of pressure drop between the two different fluids in the check valve are presented.

KEYWORDS: Water hydraulics, Check valve, Hydraulic oil, Computational Fluid Dynamics (CFD), Oil hydraulics, Pressure drop

1. INTRODUCTION

Unexpected outflows of harmful hydraulic liquids, i.e., mineral oils, into the ground and even into underground drinking-water supplies are a frequent occurrence. One of today’s major challenges is to use alternative, natural based hydraulic fluids to protect our environment. In power-control hydraulics (PCH) there are two ways in which we can protect the environment. The first solution is to use biodegradable oil [1-6] instead of a mineral oil. But this is only a partial solution because biodegradable oil has to contain the necessary additives, which are sometimes detrimental to the environment. The second – and better – solution is to use tap water instead of mineral oil. This solution is harmless to the environment, but is very difficult to realise [7-9]. For water hydraulics a relatively simple conventional control valve already exists on the market; however, the check valves are needed for almost every hydraulic machine. Despite many years of water-hydraulics research there is still insufficient understanding of the mechanisms and performance.

In this work we present some numerical and experimental results on the new design of check valve for use with water comparable to mineral oil.
2. DESIGN AND DEVELOPMENT OF A NEW CHECK VALVE

2.1. Working principle and state of the art

Check valves are one of less complex hydraulic control valves. The two main goals of check valves are to assure flow just in one direction with low pressure drop and to tight flow in the opposite direction. It should prevent against non-controlled flow direction, protect hydraulic pumps and motors against pressure surge effect.

There are different types of check valves on the market. They are used in hydraulics, water supply networks, pharmacy, food industry, power stations [10, 14, 15], etc. Check valve implementation must be done on the basis of material, size, robustness and general hydraulic system requirements. Check valves for food industry and pharmacy are often made from different non-corrosion materials such as stainless steels. Check valves should have possibility to be simply disassembled into pieces [11]. For usage in hydraulic systems with small flow rate mentioned valves are small, less robust, often made from plastic materials, as they offer better corrosion and chemical resistance than metals [12].

There are also such check valves, which are made of only one piece [13]. The check valve can be closed by gravity and therefore without the spring. In the case of application with used spring is necessary to ensure a regular cleaning of the components, since the spring offers the possibility of accumulation and to the proliferation of bacteria within the valve [11].

For the closing elements are used beads or various variants of the conical closing elements.

On the market there are several versions of hydraulic non-return valves, from the smallest (the size of a few mm, which are intended to extremely low flow) to the major cases that are designed to a high flow rate (over 100 l / min) and are used in the industry, construction machinery, forest, etc.

There is also some L-shaped [16] or any other shapes. Most of them are designed for hydraulic systems, today is the desire to widen the scope on the water hydraulics as it is in case of a spill, unlike hydraulic oil, harmless to the environment.

Axial check valve is most commonly used check valve in hydraulic industry. A schematic illustration of this kind of valve is shown on Figure 1. A schematic view of check valve for radial flow redirection is shown on Fig. 2.

![Figure 1. Check valve with in-line - axial flow direction](image-url)
2.2. Design of new check valve

In order to understand the function and to examine the details of check valve, we have designed a 3D model. Figure 3 presents the tested valve in a dedicated housing for the measurement purpose. We have installed two diaphragm pressure sensors and two temperature sensors on the measuring unit. One pressure and one temperature sensor are connected before and another two after tested specimen.
3. NUMERICAL CALCULATIONS

3.1. Numerical background

**Finite volume method**

For the purpose of computational fluid dynamics, finite volume method is used. Partial differential equations, such as the Navier-Stokes equation and the continuity equation must be solved. The form of the Navier-Stokes equations for incompressible fluid with constant viscosity can be written as vector of transport equation (Eq. 1).

\[
\frac{\partial (\rho \mathbf{v})}{\partial t} + \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = -\nabla p + \nabla \cdot [\eta (\nabla \mathbf{v} + (\nabla \mathbf{v})^T)] + f
\]  

(1)

For each component of the coordinate system (e.g., X component) can be written in the form of general transport equation, where we are interested in the size of the selected physical quantities in the region of interest or domain. For conservative systems transport equation (Eq. 2) can be used.

\[
\rho \left[ \frac{\partial \phi}{\partial t} + (\mathbf{v} \cdot \nabla \phi) \right] = \nabla \cdot (\Gamma^\phi \nabla \phi) + S^\phi
\]  

(2)

Since we were looking for place and time, depending on the function \( \phi(x, t) \), the solution of transport equations and corresponding initial and boundary conditions had to be taken into consideration. \( \phi \) represents one of the quantities of interest (e.g., speed in the x-direction), usually expressed per unit mass.

The transport equation taken into account must be rewritten in a conservative design and integrate the control volume (Eq. 3).

\[
\int_V \frac{\partial (\rho \phi)}{\partial t} dV + \oint_S (\rho \mathbf{v} \phi) \cdot dS = \oint_S (\Gamma^\phi \nabla \phi) \cdot dS + \int_V S^\phi dV
\]  

(3)

By using Gaussian sentence and an additional adjustment, we can derive the final form of equation (Eq. 4).

\[
\int_V \frac{\partial (\rho \phi)}{\partial t} dV + \sum_f \rho_f \phi_f \cdot \mathbf{n}_f S_f = \sum_f \Gamma^\phi_f (\nabla \phi)_f \cdot \mathbf{n}_f S_f + S^\phi p V_p
\]  

(4)
3.2. Numerical model with parameters

Due to the turbulent flow within the valve it is necessary to create mesh based on the boundary layers. Since our focus was on local conditions inside valve, we choose the element size 0.5 mm around the bead, which is also a key element of the check valve.

![Figure 4. Modified mesh for CFD analysis with element size 0.5 mm (60mm x 180 mm)](image)

3.3. Results of numerical calculations

Numerical fluid analysis showed the layout streamlines within the valve. As shown in Figure 6 we obtain the maximum pressure drop through the ball, which is also expected. The pressure drop through the valve with mineral hydraulic oil is 0.59 MPa at a nominal flow rate of 100 lpm.

![Figure 5. Flow simulation at spherical lock element and oil flow rate 100 lpm (60 mm x 180 mm)](image)
Comparison of the pressure drop through the same valve with two different liquids can be seen in Figure 7. In the first case is used water and the other mineral oil. Mineral oil has at maximum nominal flow of the valve 100 lpm pressure drop 0.59 MPa, while water has 0.5 MPa. The main impact has the viscosity of the oil, which is noticeable in the differential pressure.

![Graph showing pressure drop comparison between water and mineral oil](image)

*Figure 6. Results of numerical simulations of flow through check valve for water and mineral oil*

4. MEASUREMENTS

4.1. Specimen – check valve

Check valve was designed in the way so it can be simply and fast disassembled. Check valves are consisted of housing from two pieces, seat, closing element, guidance element and spring. Design of the valve allows researchers to experiment with different closing elements (ball, different conical elements, etc.) and different number of flow channels (from 1 to 6). In this paper are presented results for the fully open slot.

Physically constructed prototype, which was used for the measurements is shown in Figure 8. Detailed view of tested valve with labeled components can be seen on Fig. 9.

![Prototype of water and oil check valve](image)

*Figure 7. Specimen – prototype of water and oil check valve*
4.2. Water hydraulic test rig

For the measurement purpose we designed dedicated water hydraulic test rig, which is made from elements presented in Figure 9.

Legend:
1 – Pump with electromotor
2 – Pressure relief valve
3 – Check valve
4 – Hydraulic accumulator – piston type
5 – Ball valve no. 1
6 – Ball valve no. 2
7 – Flow meter
8 – Pressure sensor no. 1
9 – Temperature sensor no. 1
10 – Temperature sensor no. 2
11 – Pressure sensor no. 1
12 – Specimen – tested check valve
13 – Reservoir of water
Oil hydraulic test rig was designed from similar elements as water hydraulic test rig (Fig. 10).

![Image of Specimen – check valve on water hydraulic test rig](image)

**Figure 10. Figure of Specimen – check valve on water hydraulic test rig**

4.3. Test parameters

Firstly it was necessary to assemble the check valve with a desired combination of the closing elements. Then we added sensors (two pressure and two temperature). When the specimen was prepared, it was inserted into the hydraulic test rig assembly and through electrical cables connected to the control/measurement structures. Using LabVIEW, we monitored desired parameters and saved measurement results for post-processing and analysing.

4.4. Testing procedure

Measurements with tap water were held in the second test rig with hydraulic accumulator. When the present pressure was reached, the pump turned off and the main valve was opened. Later we started the program on the computer for data acquisition and opened ball valve. Water has been released with a maximum flow rate of 60 l/min. The whole measurement took 5 seconds, data were acquired with a frequency of 100 Hz. Each measurement was repeated three times.

We gathered the following data:

- Temperature 1 [°C] - liquid temperature before entering the non-return valve,
- Pressure 1 [MPa] - the pressure of the liquid before entering the non-return valve,
- Temperature 2 [°C] - temperature of the liquid outflow from the non-return valve
- Pressure 2 [MPa] - the pressure of the liquid on the outlet of the check valve,
- Flow rate [l/min].

4.5. Measurement results

Measurements were made to validate numerically determined results. Since the water hydraulic test rig was not able to produce larger flows then 60 l/min, values were extrapolated. To our surprise, the measurement results have shown the opposite of simulation results, the pressure drop was higher in water than in oil. In the graph (Fig. 11) we can see the pressure drop in the water for 0.05 MPa higher than in oil.
5. DISCUSSION AND ANALYSIS

If we focus only on the water, we can conclude from the measurement results (Fig. 12), that the steepness of the curve of the pressure drop in dependence on the movement is greater and more linear comparing to the numerically calculated data. Numerical results have shown smaller pressure drop and less continuously rising curve.
6. CONCLUSION

A new design of check valve for water hydraulic research has been done. It allows researchers to easily replace components for testing purposes. In the paper are presented numerical and measurement results for pressure drop depending on the flow rate of two different liquids – tap water and mineral oil.

Numerical simulations has shown that mineral oil has higher pressure drop (0,6 MPa) then water (0,5 MPa). But experimental tests proved the opposite, water had higher pressure drop. From comparing numerical and experimental data we can conclude that experimentally obtained results rise more linearly then numerically derived.

In the future we will make measurements for different number of axial throttles inside the valve to verify the influence of the number of slots on the pressure drop through the check valve. Also we plan to test different shapes of closing elements. We have found water check valve, which is used in water high pressure washer. Currently we are making measurements to determine pressure drop through mentioned valve.

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REFERENCES


INVESTIGATION OF THE MAIN IMPACTS ON ELECTROSTATIC CHARGING IN FILTERS

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ABSTRACT

Currently hydraulic fluids with a reduced ash and zinc concentration are deployed to a greater extent. Unfortunately these fluids exhibit a low electric conductivity. When these fluids pass through filters they become electrostatically charged. This in turn can lead to discharges, which are capable of destroying the filter and facilitate the oxidation of the fluid. These effects pose a threat to the functionality of the entire system. In order to face these potential problems it is mandatory to gain a better understanding of the underlying mechanisms and the contributing factors. In this paper an initial model for electrostatic charging is proposed. Subsequently the existing experimental setup and some required modifications are presented before the experimental results of a selection of filter material samples are given. This investigation includes the impact of the pressure, temperature, flow rate as well as the housing’s ground as influencing factors. Using these results, general observations regarding the main contributing factors of electrostatic charging are derived. In addition to these observations a qualitative explanation of some of the identified tendencies is made.

KEYWORDS: electrostatic charging, filters, experimental results, model, fleece, electrostatic density

1. INTRODUCTION

Just about everyone knows that rubbing a balloon against a sweater will cause the balloon to get electrostatically charged. In a way this form of electrostatic charging is akin to the electrostatic charging in a hydraulic system. This is particularly the case when a poorly conductive hydraulic fluid is passing through a filter. In this instance the two poorly conductive materials come into close contact and a charge transfer can occur [1]. When these charges are allowed to amass, the potential for electrostatic discharges significantly increases. In a number of specialised fields the problem of electrostatic charging has been investigated. Klinkenberg et al. investigated the charging of petrol as early as 1958 [2]. Power transformers are another application which has a well-documented history with electrostatic discharges [3, 4]. In all presented examples the charge is generated by an electro-chemical double layer, which is then separated by the flow of the fluid [5, 6]. A consequence is the accumulation of charge in the hydraulic fluid, particularly in the event of a poorly conductive fluid.

The underlying intention of using hydraulic fluids without the additives soot and zinc is to reduce the environmental impact of the waste water processing. As a direct consequence the conductivity decreases significantly. An ordinary hydraulic fluid exhibits a conductivity of well above 500 pS/m whereas some
hydraulic fluids can go as low as 10 pS/m. The aim of the presented research is to identify additional key factors responsible for the electrostatic charging of the hydraulic fluid. The key factors under investigation are the flow rate, the system temperature, the system pressure and the housing’s grounding. The experiments are conducted with two filter fleeces, which differ only in pore size. A total of three fleece combinations will be under investigation. For the experiments the pre-existing test bench at the Institute for Fluid Power Drives and Controls (IFAS) of RWTH Aachen University was modified.

The paper is structured in five sections. The initial section will briefly summarise the applied measurement system and how it differs from already available methods. Another part of this section will be dedicated to the improved integration of the measurement system and the motivation for this modification. Kühnlein et al. published a paper focusing on the measurement system in greater detail [7]. The second section takes a closer look at a simplified model, which is capable of explaining qualitative effects with regard to the electrostatic charging of the hydraulic fluid when passing through a filter. In succession to the model the experimental results are presented. This is followed by the derivation of the tendencies regarding the key factors for electrostatic charging. This second to last section establishes the connection between the observed charges in combination with the simplified model. The last section of this paper summarises the results of the experiment and closes with an outlook.

2. TEST BENCH

2.1. Measurement System

Previous research has shown that there is no readily available method for determining the charge of a hydraulic fluid depending on its operating conditions. Therefore Kühnlein et al. developed a system, that can be incorporated into a test bench, which is a crucial aspect for the experiments. Previous methods always demanded that a small sample is extracted from the system through a valve or a bypass. Under any circumstances the measurement methods are based on a discontinuous method. For greater detail with regard to the measurement system please refer to [7].

The goal of the measurement system is to be able to measure continuously within the system, thus making it possible to observe the fluid’s charge as the fluid flows at various temperatures and pressures through the test bench. All measurement methods exploit the electric field emitted by the fluid. The measurement system developed by Kühnlein et al. bases on this principle as well [7]. Figure 1 shows a cross-section of the developed measurement system. The non-conductive hose, which contains the charged fluid, is in the centre of the experimental setup. In this instance the fluid carries a positive charge. Since non-conductive materials do not obstruct the effect of an electric field, the inner electrode of the cylindrical capacitor experiences the electric field emitted by the fluid.
As a consequence the inner electrode is subjected to the effect called induction. When a conductive and neutral body enters an electric field, as seen in Figure 2, the body will experience a separation of charges. This effect is utilised to couple in a charge into the cylindrical capacitor, thus creating a potential differential in the capacitor.

In the designed experimental setup the capacity ($C$) of the cylindrical capacitor is known and the induced potential differential ($U$) can be measured by using a voltmeter with a particularly high inner resistance. Therefore the charge in the capacitor ($q$) can be calculated with Equation (1). Under ideal circumstances the inner electrode will remain neutral throughout the entire process.

$$q = C \cdot U$$ (1)

The key difference to alternative methods, such as a mini-static tester [8] or an absolute-charge-sensor [9], is the continuous measurement capabilities. Another advantage of this measurement method is that, at least in theory, a full integration into the main stream of the hydraulic flow is possible.

2.2. Test bench modifications

The original hydraulic system design by Kühnlein et al. used a total of two charge measurement chambers. The first chamber was placed in a bypass directly before the filter module, as seen in Figure 3. During a measurement cycle the flow was diverted by a valve through the charge measurement chamber. After one minute of flow the initial charge was determined. Following this measurement of the initial charge in the fluid the first valve directed the flow through the filter module.
Once the hydraulic fluid passed the filter module the flow was sent through the second measurement chamber. The charge measurement cycle was repeated once more before the second valve allowed the flow to pass straight to the reservoir, without any further measurement. This was also the initial state. This state was used during the adjustment of the temperature, pressure and flow rate.

This particular setup relies on a very brief run-in period, once the flow is directed into the filter module. Unfortunately preliminary experiments have shown that the run-in period cannot be neglected. This made a redesign of the measurement setup necessary. The resulting test bench schematic is depicted in Figure 4. The crucial advantage of this revised layout is the reduced dependence on quickly reaching a stationary state. With this configuration the stationary state is no longer interrupted by the redirection of the flow. This means that the necessary repetition of a measurement can be in much quicker succession. Another advantage of this configuration is the bypass around the filter module. Due to the bypass it is possible to adjust the system parameters without exposing the module to undefined or undesired system parameter configurations.

3. MODEL FOR ELECTROSTATIC CHARGING IN A FILTER

A typical filter consists of a number of layered filter fleeces which are folded together with a supporting material, see Figure 5. This package is then assembled around a supporting structure, which is made out of
a perforated metal plate. At either end there is a cap which holds everything together. Before looking at an entire filter element, single components should be investigated. This also extends to the two different types of flow present within a filter, based on the quantity of surface exposed to the fluid.

When the fluid flows into and out of the filter there is relatively little contact between the filter material and the hydraulic fluid. The second situation in a filter occurs when the fluid passes through filter fleeces. In this particular situation a significantly larger exposure of the fluid to the fleece occurs. Because of this higher degree of exposure, the second described situation will form the basis of the following model.

![Sectional view of a filter element](image)

It can be assumed that the charging due to flowing through the filter is greater, since the surface area within the filter fleece is larger than the outside area. The later presented results focus on this situation, which is reflected by reducing the test object to flat filter fleeces instead of whole filter elements.

A typical filter fleece consists of two major components. The first component is typically a fibre. The filter fleeces under investigation in this paper always consist of fibreglass. The second component is the binder in the form of a resin. It is by adjusting the ratio resin to fibre and the manufacturing process, that the pore size is determined.

The simplified model assumes that a filter fleece can be modelled as a bundle of tiny tubes, as illustrated in Figure 6. In order to reflect characteristic features of a fleece certain descriptive values have to be obtained, such as the inner surface and the average pore size. For the average pore size the Mean Flow Pore Size (MFP) is used, which is provided by the fleece manufacturer. The inner surface is determined by using the BET-method in accordance with DIN ISO 9277. In addition to these microscopic features the macroscopic features, such as the geometric dimensions, like the thickness of the fleece and the diameter, have to be documented. The diameter of the fleece is due to the filter module’s size, whereas the fleece thickness is also provided by the fleece manufacturer.

The first step in parameterising the model is by determining the number of required tubes to achieve the necessary inner surface of the fleece. Table 1 shows the characteristic values of one of the used filter fleeces.
Figure 6. Simplified model of a filter fleece

Table 1. Characteristic values of test filter fleece

<table>
<thead>
<tr>
<th>Filter fleece</th>
<th>Diameter [mm]</th>
<th>Thickness [mm]</th>
<th>Inner surface [m²]</th>
<th>d_{MFP} [µm]</th>
<th>Number of tubes</th>
<th>Open surface [mm²]</th>
<th>Average rate of flow [mm/s] @ 8.3 l/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>fine fleece</td>
<td>140</td>
<td>0.48</td>
<td>1.524</td>
<td>&gt;144·10⁶</td>
<td>7</td>
<td>5,542</td>
<td>25</td>
</tr>
<tr>
<td>rough fleece</td>
<td>140</td>
<td>0.6</td>
<td>1</td>
<td>&gt;36·10⁶</td>
<td>1</td>
<td>5,863</td>
<td>23.6</td>
</tr>
</tbody>
</table>

The number of tubes can be determined by calculating the lateral surface of one of these pipes and dividing the inner surface by this number. In order to check for the feasibility of this number it is prudent to calculate the area required for the tubes’ end faces and compare it with the available surface area. Coincidently the sum of the end faces also represents the area through which the flow has to pass. With this information it is possible to make an estimate for the average flow velocity in one of the tubes. In the example in Table 1 the assumed flow rate through the filter is 8.3 l/min, which results in an average velocity of approximately 25 mm/s in a tube. Crucial for later comparisons is the observation that the average velocities within the fleece are almost the same, although the MFP of the two fleeces differs significantly.

For the model the work of Klinkenberg et al. [2] forms the basis. They researched the electrostatic charging of fuel as it passes through tubes. For this purpose they developed two equations (2) and (3) [2]. Equation (2) describes the charging process for laminar flow. The condition for laminar flow is a smooth surface of the pipe alongside a certain transition distance [10]. Considering that the real system, the filter fleece, has frequently intersecting flow trajectories it is not reasonable to assume laminar behaviour in a classical sense. Therefore equation (3) for turbulent flow will serve as the basis for this model.

\[ q_{\text{laminar}} = - \int_{t=0}^{T} 8\bar{v}\pi\varepsilon_0\zeta dt \]  
\[ q_{\text{turbulent}} = - \int_{t=0}^{T} 0.04Re^{0.75}\bar{v}\varepsilon_0\zeta dt \]
As a consequence the model is dependent on the previously estimated average velocity ($\bar{v}$), the permittivity of the fluid ($\varepsilon\varepsilon_0$) and the kinematic viscosity ($\nu$) as part of the Reynolds number. The Reynolds number further introduces the hydraulic diameter to the model, which is approximated by the MFP.

The last remaining variable is the zeta-potential ($\zeta$), which is also referred to as the electrokinetic potential of a fluid. According to the International Union of Pure and Applied Chemistry (IUPAC) the calculation or determination of this potential is under discussion. This is due to the dispute regarding the viscosity and the permittivity of the fluid in the double layer [11]. Figure 7 illustrates a simple model for the zeta-potential, which was first formulated by von Smoluchowski in 1921 [12]. Summarized, the zeta-potential is the electrostatic potential observed at the shear plane separating the immobile layer of a fluid at an interface from the flowing sections. For a more detailed explanation of the experimental setup please refer to [13].

![Diagram of electrostatic potential](image)

**Figure 7.** Distribution of electrostatic potential near a charged surface and the resulting electro-osmotic velocity under an applied field [13]

4. EXPERIMENTAL RESULTS

Preliminary experiments on hydraulics of different levels of contamination revealed that the contamination in itself can reduce the electrostatic charging. The presented experiments were conducted with oil having a contamination level of 14/11/9 according to ISO 4406:1999. The conductivity of the used oil is 10 pS/m.

The results presented here are based on experiments which were conducted with the modified test bench shown in Figure 4. A measurement cycle always starts with the flow being directed through the bypass of the filter module. In this state the flow rate, the temperature and the system pressure are adjusted. Once the correct system parameters are reached, the flow is directed through the filter module.

Figure 8 summarises the results of the three tested fleece configurations and is structured in four layers. Each layer represents one system parameter which is varied. At the bottom is the oil temperature, which is either 25°C or 30°C. The next layer stands for the flow rate, which is either 6 or 8.3 l/min. Besides the flow rate and the temperature layer a pressure layer also exists. In the layer of the system pressures two pressures, 12 and 50 bar, are represented. The last layer at the very top of the diagram indicates which cluster of measurements is generated with a grounded filter module.
As mentioned earlier, there are three filter fleece configurations under investigation. The first two configurations are the two individual filter fleeces tested separately. The third configuration combines the fine and the coarse fleece with another in the order one would find them in a filter. That means the first layer is a coarse fleece followed by a fine fleece.

5. OBSERVATIONS AND INTERPRETATION

When looking at the results in Figure 8 six observations can be made. The charge density:

1. decreases with an increase in temperature,
2. increases with an increase of the flow rate,
3. exhibits hardly any change due to grounding,
4. demonstrates only little pressure sensitivity,
5. increases with a decrease of the MFP,
6. of the fleece combination is not the sum of the charge density of the individual fleeces.

The current data suggests that the charge density is very sensitive towards changes in temperature. At the moment it is not clear what the underlying cause for this sensitivity is. It could be due to a strong correlation with the kinematic viscosity or an interdependency of the conductivity with the temperature. In fact Equations (2) and (3) suggest that a decrease in kinematic viscosity would cause an increase in charge density, due to the inverse correlation of the kinematic viscosity and the Reynolds number. Yet the detected charge densities at 30°C are significantly lower than their lower temperature counterparts. Preliminary findings also suggest that this decrease of charge density continues with a further increase in temperature until a negligible level is reached. Assuming the model to be capable of indicating trends of the fluid’s charge, this would mean that other characteristics of the fluid, such as the zeta-potential, change in correlation with the temperature. This is a plausible assumption considering that the zeta-potential is dependent on the dynamic viscosity [11, 13]. Unfortunately the zeta-potential could not be determined for this particular parameter combination in an experiment thus far. In order to parameterise the model, further investigations regarding the zeta-potential are required.

Figure 8 indicates a strong relationship between the charge density and the flow rate. A close examination of Equation (3) already suggests such a relationship.

The last two observations regarding the MFP and the fleece combination are of particular interest. Intuitively the smaller pores of the fine fleece would cause more friction, thus causing a higher charge density. As the results in Figure 8 show, this is not the case. Again a look at Equation (3) gives a clue why the charge density responds the way it does. The Reynolds number depends on the average velocity as well as the hydraulic diameter. This means the charge density might increase if the average velocities remain about the
same whilst the hydraulic diameter increases. Table 1 contains the average velocity as well as the MFP, which is the hydraulic diameter in this model. Comparing the average velocities for the two different fleeces shows that the velocity remains about the same for both fleeces. The hydraulic diameters, that are the MFPs, on the other hand differ by a factor of two.

Another observation is that the charge density of the fleece combination is not equal to the sum of the fleeces’ individual charge densities. Actually, the values for the combination’s charge density seem to correspond to a weighted average.

6. SUMMARY AND OUTLOOK

This paper is structured in five sections and begins with a brief review of the developed measurement method before reviewing the test bench. This also includes a look at the recent modifications of the test bench in order to accommodate a quicker reaching of a quasi-stationary state. For this purpose the measurement chambers were integrated into the main stream of the hydraulic system. The second section of this paper proposes a simplified model for charging of the hydraulic fluid when passing through a filter fleece. This model is based on the work of Klinkenberg [2] and adopts the equation for turbulent flow to a filter fleece. For the simplification of the fleece it is assumed that a fleece can be modelled as a bundle of very rough tubes. The resulting bundle has the same inner surface area as the original fleece. The introduction of the model is followed by the presentation of the experimental results, which indicate a strong correlation of charging with the temperature and the flow rate.

At the moment the research with and the optimisation of the test bench continues. The aim of the future modifications is to enable the test bench to test full size filter elements. Also an investigation of oils with different kinematic viscosities is still pending. With these results it will be possible to differentiate between a temperature and a viscosity effect. In addition to the use of an other oil there will be a detailed investigation of the fleece combinations. Part of these experiments will also include the reversal of the fleece combination.

7. ACKNOWLEDGEMENT

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NOMENCLATURE

<table>
<thead>
<tr>
<th>Designation</th>
<th>Denotation</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>C</td>
<td>Capacity</td>
<td>[F]</td>
</tr>
<tr>
<td>(d_{MFP})</td>
<td>Mean Flow Pore size used as hydraulic diameter</td>
<td>[µm]</td>
</tr>
<tr>
<td>p</td>
<td>System pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>q</td>
<td>Charge</td>
<td>[C]</td>
</tr>
<tr>
<td>Q</td>
<td>Flow rate</td>
<td>[l/min]</td>
</tr>
</tbody>
</table>


<table>
<thead>
<tr>
<th>$t_{\text{fleece}}$</th>
<th>Thickness of fleece</th>
<th>[mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T$</td>
<td>System temperature</td>
<td>[$^\circ$C]</td>
</tr>
<tr>
<td>$U$</td>
<td>Potential differential with the capacitor</td>
<td>[U]</td>
</tr>
<tr>
<td>$v$</td>
<td>Liquid velocity</td>
<td>[m/s]</td>
</tr>
<tr>
<td>$\bar{v}$</td>
<td>Average velocity through a tube representing a fleece’s pore</td>
<td>[mm/s]</td>
</tr>
<tr>
<td>$x$</td>
<td>Distance from shear plane</td>
<td>[mm]</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>Relative permittivity</td>
<td>[-]</td>
</tr>
<tr>
<td>$\varepsilon_0$</td>
<td>Permittivity of vacuum</td>
<td>[F/m]</td>
</tr>
<tr>
<td>$\zeta$</td>
<td>Zeta-potential, electrokinetic potential</td>
<td>[mV]</td>
</tr>
</tbody>
</table>

REFERENCES


FOCUS ON TECHNICAL CLEANLINESS IN HYDRAULIC MANIFOLD SYSTEM MANUFACTURING

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ABSTRACT

ISO 16232 (2007) standard for technical cleanliness was created based on needs from automotive industry in Europe. It has not been used so far widely in hydraulics manufacturing where various cleanliness standards are used to analyze and monitor hydraulic fluid cleanliness. Parker Hannifin is global leader in motion and control technologies. Parker Hannifin’s Hydraulic Controls Division Europe is developing and manufacturing hydraulic manifold systems and special valves for focused Marine, Oil&Gas and Heavy Mobile applications in Business Unit Lokomec (later PH Lokomec) located in Tampere, Finland. Products are mainly customer specific and developed in close co-operation with customers.

PH Lokomec joined a multi-company project named KompuNW (1.1.2011 – 30.4.2013) focusing on technical cleanliness. Tampere University of Technology’s Department of Intelligent Hydraulics and Automation (later TUT/IHA) was the research party in the project. During this project PH Lokomec developed knowledge and understanding about technical cleanliness and started to shape a vision for itself about technical cleanliness in future. Several process changes were introduced during the project and PH Lokomec decided to include technical cleanliness as one of the key areas to focus in future operational developments. Lean philosophy is a key part of Parker Hannifin’s strategy to continuously pursue operational excellence. PH Lokomec has identified a very strong link between several key Lean tools and technical cleanliness. PH Lokomec is continuously developing manufacturing processes to improve technical cleanliness of products.

KEYWORDS: ISO 16232, technical cleanliness, hydraulic manifold systems, Lean

1. INTRODUCTION

In Finland the first remarkable component cleanliness research project was done by VTT/Production technology in 1994 [1]. The title of research project was “The identification and decreasing of contaminants in mobile hydraulics during manufacturing”. The laboratory measurements of oil cleanliness were done in connection of elementary functional test bench constructed by VTT. E.g. hydraulic hoses, and cylinders
were widely studied with this test bench using 4-channel UCC CM20 particle counter according to NAS 1638:1964 standard.

The second serious research project in this survey area in Finland was done by TUT/IHA in 1999 – 2002 with title “Improving the cleanliness of hydraulic components and systems by way of developing work phases in manufacture, transport and assembly work” (PUHKO projects) [2]. The influence of production improvements in industry were measured by means of the oil cleanliness measurements using PAMAS 3316 FM particle counter according to standards NAS 1638:1964 and ISO 4406:1987.

More and more companies, involved in precision engineering, are embracing methods and procedures for extracting and analyzing particulate contamination from production and the environment. For instance, in the automotive industry, (Verband der automobilindustrie, VDA) volume 19 (2004), “Technical cleanliness testing – Particulate contamination of function-relevant automobile components,” and its international counterpart, namely ISO 16232 (2007), volumes 1 through 10, “Road vehicles – Cleanliness of components of fluid circuits,” are now referred to in supply contracts with regard to particulate cleanliness for the respective components. As a result, the technical cleanliness of parts is rendered objectively assessable and comparable. [3]

The German car industry has done multi-company co-operation with Fraunhofer Institute, Stuttgart and e.g. some other partners concerning several fluid components in car industry during the first decade of 2000. The title of project was InnoNET project “TecSa-line”, Automatic measure system with sensor for macro particles in fluid. This project was presented in Parts2clean fair in Stuttgart in 2009. For power hydraulics the example of maximum allowed particle size is given to be 600 µm [4].

VDA published the second publication (Part 2) concerning this theme in 2010. The title is “Technical cleanliness in assembly - Environment, Logistics, Personnel and Assembly Equipment” [5]. An overview about the component cleanliness in ten years period, mainly in automotive industry, is presented in reference [6].

The Component cleanliness Network (KompuNW project, mainly 2011 – 2012) is the third serious research project to improve the initial cleanliness of manufactured and assembled hydraulic components in Finland [7], [8]. Now, the equipment used, are designed for component cleanliness measurements with concerning standards. Pressure rinsing with suitable solvent is the most common particle extraction method in commercial test cabinets. The functional test bench, designed and constructed in IHA, uses hydraulic oil for extraction of contaminants from assembled fluid power components. Both of these need to have their own extraction procedure for each studied part or component.

No matter, how the particles are extracted from the studied item on the membrane, they all must be photographed, measured, classified, and calculated to formulate the certain component cleanliness code (CCC) that is the result of the cleanliness analysis. This hard routine work is mainly done by automatic microscope with proper software in a very effective PC. However, a qualified researcher is needed to take care of this close examination to ensure that the whole analysis is executed according to the standardized rules.

2. CLEANLINESS STANDARDS APPLIED IN HYDRAULICS

It is very common that the current cleanliness standards are not very well known in fluid power industry. So, they must be clarified here to give better understanding to their differences in use.

2.1 Oil cleanliness standards

Oil cleanliness of fluid power systems have been measured with online equipment since early 1990s. The most common way to describe the cleanliness of oil in the fluid power system is the use of ISO 4406:1999 [9]
Another current possibility is to use SAE AS4059:2005E [10] standard that has certain profit in use as it is pointed in following description. These standards are partly compared in Table 1.

<table>
<thead>
<tr>
<th>Standard</th>
<th>Type of size range</th>
<th>Particle size ranges [µm(c)]</th>
<th>Cleanliness classes</th>
</tr>
</thead>
<tbody>
<tr>
<td>ISO 4406:1999</td>
<td>cumulative</td>
<td>≥4, ≥6, ≥14</td>
<td>0 – 28</td>
</tr>
<tr>
<td>SAE AS4059:2005E</td>
<td>cumulative</td>
<td>&gt;4, &gt;6, &gt;14, &gt;21, &gt;38, &gt;70</td>
<td>000 – 12</td>
</tr>
<tr>
<td></td>
<td>differential</td>
<td>6–14, 14–21, 21–38, 38–70, and &gt;70</td>
<td>00 – 12</td>
</tr>
</tbody>
</table>

The ISO 4406:1999 standard shows the cleanliness classes of oil with three cumulative particle sizes: ≥4 µm(c), ≥6 µm(c), ≥14 µm(c). The less particles are measured the lower the cleanliness class number is and the better is the cleanliness of the oil. The measured cleanliness class is compared to the cleanliness requirements of the fluid power system. Those three limit values are chosen from the small cumulative particle size area because they describe partly the size of critical clearances between moving parts of the common hydraulic components.

The SAE AS4059:2005E shows the cleanliness of oil with six cumulative particle size ranges or using the corresponding five differential ranges. This standard gives better opinion about the presence of big particles in oil because they are not calculated in a one big sum as they are in ISO 4406 (≥14 µm). This subject is discussed more in L. Elo et al. [11]. Compared to oil cleanliness, component cleanliness measurements concentrate mainly on considerable big particle sizes. The origin of those is in manufacturing process. The most critical sizes are from 100 µm up to >1000 µm.

2.2 Component cleanliness standards

The ISO 18413:2002 [12] was prepared for fluid power components and systems and ISO 16232:2007, Part 1-10 for automobile components. These standards have same kinds of particle extraction and analysis methods. Of course some differences exist, too. The main difference is that ISO 18413 does not include as clear code for component cleanliness reporting as it is in ISO 16232-10 [13]. This is the reason why the method described in ISO 16232 was chosen to use in TUT/IHA.

The component cleanliness code (CCC code - ISO 16232:2007) is described with number of particles with respect to the wetted surface area (A), or the wetted volume (V) or straight with respect to the component itself (N). After the letter (N, A, or V) the different particle size ranges are presented in parenthesis separated with “/”.

Appendix 1 presents Table A for used particle size ranges (size classes B – K) and Table B for number of particles (cleanliness levels 00 – 24). If the result is announced with respect to component itself the number of measured particles of each size range is written after the size range letter.

The component cleanliness code (CCC code) may e.g. be some of the following:

- CCC = N (B585600 /C58200 / D18500/ E3600/ F2800/ G700/ H190/ I4 / J0 / K00)
- CCC = A (C16 / D18/ E12/ F8/ J0)
- CCC = V (B20 / C16/D18 / E12/ F12/ G12/ H8 /I1 / J1 /K00)

The cleanliness of components is allowed to compare between each other only when they are weighted over the same kind of wetted area (A) or wetted volume (V) of component. The critical particle size ranges are suited between the component manufacturer and customer though nearly all size ranges are often calculated (see CCC examples above).

When comparing the Tables 1 and 2, it is easy to understand that the ISO 16232 standard gives better understanding to the presence of big particles in the studied component.
Table 2. Particle size ranges in component cleanliness standards

<table>
<thead>
<tr>
<th>Standard</th>
<th>Type of size range</th>
<th>Particle size ranges [μm]</th>
<th>Cleanliness classes</th>
</tr>
</thead>
</table>

These measurements are normally done using suitable solvent or low viscosity oil as extraction liquid. The extracted particles are possible to calculate with inline-method. The choices are optical particle counter or automated microscope using proper computer program.

3. SINCE KOMPUNW PROJECT

The basic research in component cleanliness of fluid power in Finland was started with large industrial based KompuNW-project (Component Cleanliness NetWork). The KompuNW project was financed by TEKES – the Finnish Funding Agency for Technology and Innovation and 12 Finnish industrial companies. Two of them were advanced Finnish manufacturer of mobile machines and 10 of their local subcontractors. Department of Intelligent Hydraulics and Automation (later IHA) was the research party in the project. The project was carried out from the beginning of 2011 up to April 30th, 2013.

Primary target in this project was to find out how the cleanliness of new fluid power components is measured in practice, and secondly, what is the current range for cleanliness code in each component group. The cleanliness results of fluid power components are published in references [7], [8], [11], [14], and [15]. The problem is that there are no other published measurements done by other research teams, for the present, to be able to compare the current cleanliness levels of components.

Before the project the general view within PH Lokomec was that the level of knowledge and processes related to product cleanliness are on a good level. Term technical cleanliness was not commonly used in hydraulics industry and neither in PH Lokomec.

PH Lokomec’s key processes were engineering, manifold block machining, manifold assembly and hydraulic testing. Before KompuNW project the main cleanliness measurement of products was done in hydraulic testing process. The cleanliness of oil in hydraulic test unit is continuously measured with Parker icountPD online particle detectors according to ISO 4406 standard. No less than 100% of assembled products go through the test process. There was no measurement of technical cleanliness of components before assembly. During the KompuNW process it was soon discovered that ISO 4406 oil cleanliness standard is not sufficient to measure technical cleanliness of components. Size range of particles that typically occur in hydraulic components was discovered to be larger than what can be properly expressed with ISO 4406 standard. ISO 16232 appeared to cover correct size range but the measurement process is very heavy and slow so it cannot be applied as a part of manufacturing process like oil cleanliness monitoring. It can however be used to measure the cleanliness capabilities of manufacturing processes.

After KompuNW project the component cleanliness services have been developed further in TUT/IHA with the funding of Centre for Economic Development Transport and Environment (Pirkanmaan ELY-keskus) since 2013. Several company–specific studies are carried out during this time. E.g. PH Lokomec has been continued co-operation with IHA also during this ELY-Kompu project phase.

4. PH LOKOMEC’S JOURNEY SINCE JOINING KOMPUNW PROJECT

PH Lokomec is developing and manufacturing hydraulic manifold systems and special valves and the key processes are engineering, manifold block machining, manifold system assembly and full hydraulic testing of
manifold systems. Full hydraulic testing is carried out immediately after final assembly and testing includes at least following steps

- Static pressure test
- Verifying all system functions according to specification
- Adjusting all adjustable valves according to specification
- Flushing the flow channels with the test unit oil (VG 46).

Hydraulic oil used in test units is monitored on-line from the return line of the test station upstream the return line filters. This way it is possible to detect possible particles flushed out from the tested product but only particle size range based on ISO 4406 standard (or alternatively NAS 1638 standard). PH Lokomec has had no quality problems or other major concerns in terms of cleanliness of end products. This was mainly because of the fact that no less than 100% of assembled products undergo full hydraulic testing with clean on-line monitored hydraulic oil. Flushing during the test procedure as the final cleaning step was known to be very efficient. However it was understood that even full hydraulic testing of the product does not mean that all possible particles are always removed from products. There are such features in manifold systems where even comprehensive flushing does not create constant flow through each drilling and finally out from the product. One example of these type of features are pilot channels controlling pilot stages of valves. Therefore main motivation to join KompuNW project was to learn how to increase technical cleanliness before the hydraulic testing to minimize the possibility of particles remaining in the product.

In the beginning of KompuNW project PH Lokomec determined the range of components and processes to focus on. Although some different types of components were also included the main focus was put to machined manifolds. The focused process begins in the cleaning process which is done after machining and de-burring the manifold blocks. Process ends before hydraulic testing of the products which means the final assembly of the manifold systems.

Due to the complexity of measurement process to achieve sufficient results according to ISO 16232 the task was split into two different steps

- Capabilities of different washing processes
- Cleanliness levels of different locations in manufacturing process.

4.1. Capabilities of different washing processes

After machining and de-burring there are two different washing processes to be monitored. In-house process consists of manual jet washing after in-house manual de-burring. Automatic ultrasonic washing using basket dunking technology is done as a combined outsourced operation together with thermal de-burring.

Part of the machined volume was processed through in-house process and part thought outsourced process. To analyze the performance of these processes it was decided to define certain process stages to focus on. Table 3 shows which process stages were identified and selected.

<table>
<thead>
<tr>
<th>Process measured</th>
<th>Cleanliness CCC-code</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. After outsourced thermal de-burring and ultrasonic washing</td>
<td>“CCC=N(C14/D13/E11/F7/G5/H4/I0/J0/K00)”</td>
</tr>
<tr>
<td>2. After outsourced thermal de-burring and ultrasonic washing + transportation</td>
<td>“CCC=N(C14/D13/E13/F9/G8/H8/I5/J4/K2)”</td>
</tr>
<tr>
<td>3. In-house washing with manual jet washing machine</td>
<td>“CCC=N(C13/D12/E10/F7/G6/H5/I1/J00/K00)”</td>
</tr>
</tbody>
</table>

CCC code in inverted commas (“ ”) is presented in cleanliness level code format although official way would be to demonstrate the number of particles in each size class.
As can be seen from first measurements (Table 3), there is not a great difference in cleaning result when comparing the two cleaning processes used for the main production volumes. These processes are demonstrated as processes 1 and 3. At the same time the results indicate that during the transportation of cleaned products back from outside operation, cleanliness suffers significantly. For reference it was also examined how a simple automated industrial jet washing machine performs and results indicated low capability of this process compared to process 1 and 3 as expected. Based on these results a variety of improvement ideas were collected, analyzed and many of them implemented. Following are some examples of process improvements started or executed.

- Better protected transport solution was implemented
- Technical investigation and investment planning was started in order to in-source all de-burring and washing operations to eliminate transportation and maximize cleaning result.

4.2. Cleanliness level of different locations in manufacturing process

Measuring and analyzing cleanliness levels in different locations in manufacturing process proved to be a very challenging task. To reach reliable and comparable results it would have been necessary to conduct a large amount of measurements for identical products stored in different locations. It would have also been necessary to mount exactly similar product in exactly same position for equally long time period. Furthermore the initial cleanliness level of the analyzed products would have had to be exactly the same.

During a brainstorming a more simple method was invented to more easily collect at least as reliable information as would have been possible by conducting a complex product exercise. Clean petri dishes including a clean oiled membrane sheet would be placed to selected locations for equally long time period and afterwards membrane sheets could directly be analyzed with automatic microscope. This way it was possible to collect comparable data of how the clean membrane is contaminated by ambient particles in different locations.

Figure 1 demonstrates an example of results where contamination particles have been divided into two different size categories; greater than 25 µm and greater than 100 µm. Reference location is a modern open office equipped with filtered air-conditioning.

As can be seen there are significant differences between different locations in manufacturing. Closed storage shelf is proven to be almost ideal protection against contamination to ambient particles. For making this finding a prototype of transparent cover hatch was developed to cover storage shelves. On the other
hand products stored unprotected in machining area are proved to get contaminated most certainly as was expected. PH Lokomec had this suspicion in mind already when creating current manufacturing strategy and it has proven to a crucially successful decision to fully isolate machining into separate ambient space from component inventories, manifold assembly and hydraulic testing. This can be listed as the first major learning point in KompuNW project.

Another important verification of earlier assumptions was that even an open storage shelf protects cleanliness better than open assembly stations. Areas near outdoors proved to be poor for cleanliness as expected. Based on these results a variety of improvement ideas were collected, analyzed and many implemented. Following are some examples of process improvements started or executed.

- Final inspection process of machined manifold blocks was changed in order to make cleaning as the last process step in machining facility before moving parts to assembly facility
- Protection against particles immediately after cleaning
- All pallets in stock shelf are covered with plastic hoods
- New kitting process was created in order to pre-kit assembly work orders by dedicated kitting team just in time before the need in assembly line to minimize storing goods unprotected in assembly lines.

In addition to all process improvements also education and training were provided to all related PH Lokomec employees by TUT/IHA personnel. Purpose of this training was to raise awareness of technical cleanliness and to share the results discovered so far how technical cleanliness can be adhered or destroyed. It is worth mentioning that the training sessions received a true interest from the audience and many valuable discussions followed.

5. CLOSE LINK TO LEAN PHILOSOPHY

Lean philosophy has been a part of Parker Hannifin corporate strategy for years. Therefore also PH Lokomec has been applying Lean methodology for a decade in manufacturing and some years in office processes. As mentioned in previous chapter one key finding for PH Lokomec during KompuNW project was that in addition to cleaning process it is very important to maintain the achieved cleanliness level as long as product is assembled, tested and protected with plugs and flanged to the final stage before installation to customer application. During KomnuNW project it was discovered that several Lean tools and methods can be directly applied to improve processes to maintain better technical cleanliness level of products through these processes. Lean principles originating from Toyota Motor Company and first introduced by Womack et.al. [16] has been found to be applicable also in component cleanliness improvement.

One of the best known summaries of Lean thinking have been presented as five step cyclical process [17], [18]. Key findings from KompuNW related to Lean are presented in this framework

- Identify value from customer perspective
- Define the value stream
- Flow
- Pull
- Pursue perfection

As a basis for all process improvements it is essential to look for what is the value from customer perspective. Looking at the first step component cleanliness as a critical factor of reliability of a hydraulic system has been further addressed as part of customer value in PH Lokomec manufacturing processes. Although the link between reliability and component cleanliness has been evident for long time it is becoming even more critical.

In the second step, “defining the value stream”, overall high-level picture of the steps needed for creation of the defined value is created. Commonly used tool helping in this process step is value stream mapping [19].
With value stream mapping all activities within a factory are mapped to a visual presentation and all value-adding process steps are identified and separated from non-value-added steps. Non-value-added steps, which are also called waste in Lean, are such as overproduction, waiting and delays, excess transportation, over-processing, excess inventory, movement and quality defects [17, pp. 27–29]. With value stream mapping PH Lokomec has identified several process steps in its manufacturing which are not only non-value-adding but also damaging the value from cleanliness point of view. Considering cleanliness as a part of customer value there are actually only few value-added steps in value stream adding this value.

As an example a manifold block is manufactured in a process which consists of raw material cutting, machining, de-burring and washing. The cleanliness is created mainly in de-burring and washing processes. Still, with value stream mapping a lot of transportation, waiting and movements have been identified in this process. During KompuNW these steps were found to be critical in terms of cleanliness because, without proper protection, they tend to damage cleanliness, i.e. the value from customer point of view. These steps typically represent most of the total process lead time. Therefore a lot of emphasis has been put to eliminate these non-value-added steps. Although reduction of waste and decreasing lead time is mainly seen as one of the main drivers of productivity and inventory measures it has found not to be in conflict with good component cleanliness: the less non-value-added time the less potential losses in component cleanliness. Even though some simple and economical ways to protect the manifold during moving, transportation and stocking was found, protection of manifold is also a non-value-added step and therefore should be eliminated if possible. Keeping in mind the fact that no protection is perfect the only way to ensure sufficient component cleanliness is to dramatically cut process lead time by eliminating non-value-added process steps.

“Flow” and “pull”, the third and fourth steps of the above-mentioned Lean process, are means to decrease non-value-added steps and overproduction. Creating continuous flow at PH Lokomec has meant process changes from separated process steps to continuous processes. For example, separated de-burring, washing and inspection processes has been put to single cells to avoid queue and waiting time as well as excess moving of manifolds. “Pull principle” has been for years a natural part of PH Lokomec manufacturing process. This means no product is manufactured without direct link from actual customer demand. By working all the time with direct link to customer demand long time in inventory is avoided and component cleanliness is more easily maintained during the process.

Pursuing perfection, the last step of the process, focuses on continuously improving the way all work is done in each process. There are several Lean tools in use in PH Lokomec to support this step. The most important tool in terms of component cleanliness is 5S. It is a systematic way to improve and maintain clean and well-organized workstations. 5S consists of five Japanese words which are translated to English as [17, pp. 150–152]:

- Sort
- Straighten
- Shine
- Standardize
- Sustain

Shortly put this means sorting really necessary items from unnecessary (Sort), straightening necessary items by defining a place for everything (Straighten) and cleaning a work cell (Shine). The first three steps are put to a continuous cycle by defining documented routines for everyone (Standardize) and creating an auditing process to sustain the process as part of daily work (Sustain).

PH Lokomec has been running 5S for nearly a decade and seen its effectiveness in terms of productivity, safety and overall working conditions. During KompuNW its importance was further increased and related audit process is currently covered also by location top management. Maintaining achieved cleanliness of a component later being part of final assembly is not all about shortening waiting times and protecting them during moving and transportation. Once brought to assembly and test workstation and unpacked from protective package there is highly increased risk of damaging the cleanliness level of a component. Without standardized ways of handling components and routines for cleaning work station regularly the component
cleanliness is very likely damaged. Cleaning procedures are also found critical for all boxes, shelves and transportation units.

As a summary Lean and component cleanliness have been found to support each other. As mentioned earlier in this chapter Lean is a cyclical process. This means it is repeating continuously from first step of value definition and continuing through all five steps just to be started from beginning. This concept of continuous improvement has been sustained as part of PH Lokomec process development. Having started a journey to better component cleanliness has been just a first step of endless process of waste elimination. The next steps to be taken are further described in the next chapter.

6. CONTINUOUS IMPROVEMENT AND VISION OF THE FUTURE

PH Lokomec has only begun the journey in the continuous improvement of technical cleanliness. Although this is not yet a common trend in industry, there is a clear dedication to focus in technical cleanliness. It has been discovered that due to current complexity and time consumption of technical cleanliness measurement process, such measurement cannot be implemented as part of manufacturing process. Despite of this it can still be used efficiently in analyzing capability of a process to create or maintain technical cleanliness. So far there are no common agreed target levels of technical cleanliness used between hydraulic component manufacturers or machine builders.

A vision has been created in PH Lokomec to describe desired future state for internal technical cleanliness. This vision is considered when planning and reviewing internal strategies and when planning improvement activities. It includes several key learning points from KompuNW project as well as things identified after the project but heavily based on the re-shaped thinking around technical cleanliness.

Key parts in PH Lokomec’s vision of technical cleanliness are

1. Continue to keep machining in separate ambient environment to clean hydraulic parts
2. Efficient and repeatable de-burring and cleaning process only in-house
3. Simple but efficient protection of clean components against ambient particles with minimum waste
4. Minimal manufacturing lead times and inventories
5. Simplified production flow with minimum amount of process steps
6. No transportation for clean products outside from manufacturing facilities
7. Regularly audited processes – especially 5S
8. Best possible control of ambient conditions where clean parts stored and used
9. Only clean products from suppliers (equal to in-house CCC requirement)
10. Oil monitoring in hydraulic test bench for particle sizes according to ISO 16232
11. Maximum allowed particle size for components in assembly process below 400µm and only up to one particle of size 200-400µm is allowed which means CCC code “CCC=N(H0/I00/J00/K00)”

Above list is only a summary of main points of the vision and for some points there is not yet a clear strategy on how to achieve it. One of these is point number nine - supplier cleanliness. It is so far impossible to expect that every supplier would be capable to measure according to ISO16232 and assure consistency to certain level. Some already can but not all. This is anyhow a part of the future vision and it will be achieved one day.

Most of the points in the vision have already a corresponding strategy on how to achieve them and improvements are being planned and executed. As an example there is an investment project ongoing for achieving the points number 2 and partially 4, 5, 6, and 11. PH Lokomec is currently investing over one million US Dollars to build a new automated thermal de-burring and cleaning cell for machined manifold blocks. This cell will enable insourcing all de-burring and cleaning to an efficient automated in-house process. Cleaning solution consists of automatic ultrasonic washing line based on basket dunking technology with underwater jets and air bubble agitation. There will be an optional process step also for Hot...
Black Oxide surface treatment in the washing line. This cleaning solution is based on already proven technology. Combined with thermal de-burring it is considered to be the best available solution for de-burring and cleaning steel manifold blocks in low volumes and wide products mix [20]. This cell will improve the consistency of the product quality and cleaning result but also reduce manufacturing lead time dramatically enabling reduction on inventory levels which is expected to further improve cleanliness of assembled parts.

Technical cleanliness is believed in PH Lokomec to be one of the future focus improvement areas of machine manufacturers applying hydraulics. General assumption is that functional problems caused by particles in a hydraulic system are one of the main reasons for problems during startups of new hydraulic applications. Hydraulics is known to be under heavy competition against competing technologies, especially electromechanical. Improved technical cleanliness is expected to decrease problems in hydraulic system startups and therefore increase productivity and decrease cost as well as manufacturing lead time. It will most likely also decrease component wear and therefore increase life time and robustness of hydraulic system.

CONCLUSIONS

An overview was shown how component cleanliness research was developed over twenty years period. The main differences of most important standards in this research field were explained. The KompuNW project and the published results were discussed shortly. PH Lokomec’s achievements and findings were told detailed. The introduced component cleanliness tasks were connected to lean philosophy that has been in use for a decade in manufacturing. Several key parts in PH Lokomec’s vision of technical cleanliness were listed and discussed open-minded.

REFERENCES


# Appendix 1:

**Component cleanliness code tables – ISO 16232:2007 – Part 10 [13]**

**Table A. Size classes for particle counting**

<table>
<thead>
<tr>
<th>Size class</th>
<th>Size $x$ (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>$5 \leq x &lt; 15$</td>
</tr>
<tr>
<td>C</td>
<td>$15 \leq x &lt; 25$</td>
</tr>
<tr>
<td>D</td>
<td>$25 \leq x &lt; 50$</td>
</tr>
<tr>
<td>E</td>
<td>$50 \leq x &lt; 100$</td>
</tr>
<tr>
<td>F</td>
<td>$100 \leq x &lt; 150$</td>
</tr>
<tr>
<td>G</td>
<td>$150 \leq x &lt; 200$</td>
</tr>
<tr>
<td>H</td>
<td>$200 \leq x &lt; 400$</td>
</tr>
<tr>
<td>I</td>
<td>$400 \leq x &lt; 600$</td>
</tr>
<tr>
<td>J</td>
<td>$600 \leq x &lt; 1000$</td>
</tr>
<tr>
<td>K</td>
<td>$1000 \leq x$</td>
</tr>
</tbody>
</table>

**Table B. Definition of the cleanliness level of a component**

<table>
<thead>
<tr>
<th>Number of particles per 1000 cm² per 100 cm³</th>
<th>Cleanliness level</th>
</tr>
</thead>
<tbody>
<tr>
<td>More than</td>
<td>Up to and including</td>
</tr>
<tr>
<td>0</td>
<td>0</td>
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<td>0</td>
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<td>1</td>
<td>2</td>
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<td>2</td>
<td>4</td>
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<td>8</td>
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<tr>
<td>$8\times10^6$</td>
<td>$16\times10^6$</td>
</tr>
</tbody>
</table>
DESIGN OF A HIGH SPEED SINGLE PISTON PUMP FOR PISTON/CYLINDER PAIR AND SLIPPER/SWASH-PLATE PAIR OIL FILM INVESTIGATION

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ABSTRACT

Measurement of oil film characteristics in friction pairs is an effective method to understand the friction behaviour and energy behaviour of the hydraulic piston pump. In this paper, a single piston pump with high rotary speed for piston/cylinder pair and slipper/swash-plate pair oil film investigation is presented. The design problem of flow distributor and flow pulsation attenuation is discussed. A separate flow distributor is designed to make piston chamber connect either suction port or discharge port synchronously. The effect of oil elasticity on distributor’s leakage is analyzed and a lag angle is applied to distributor for less leakage. The pulsation attenuation effect with accumulator is analyzed using linearization method. Finally the whole single piston pump system is modeled in AMESim software. The simulation results shows the single piston pump achieve pulsation amplitude less than 9.4% of average flow-rate and volume efficiency higher than 56.3%.

KEYWORDS: Single piston pump, oil film, flow distributor, flow pulsation attenuation

1. INTRODUCTION

The axial piston pump is the key hydraulic power component to transmit mechanical power to hydraulic power due to its high-pressure, high power capacity and compact design. The friction pairs in piston pump including piston/cylinder pair, slipper/swash-plate pair, and cylinder-block/valve-plate pair are decisive to the pump’s energy efficiency, noise level and durability. To achieve good performance of the piston pump, many studies about friction pairs have been carried out focusing on wear characteristics [1] [2] and energy dissipation [3] both numerically and experimentally. This is more critical for aviation piston pumps, since they always run at an extremely higher rotational speed than industrial pumps, which will cause wear failure if lubrication is inefficient.

The research group at RWTH Aachen studied the friction behaviour of piston/cylinder pair aiming at ensuring the lifelong performance of the PVD-coated piston [2] [4]. They optimized the geometry of the piston-cylinder contact to reduce the friction forces between piston and bushing, which is an indicator for the contact pressure. Simulation models were built as a tool to determine good contours of piston and useful combinations of gap width and cylinder length [5]. An experimental test rig was built up to measure the friction force and verify the numerical model. The measurement was carried on a wobble designed single piston pump which runs at low number of revolutions \(n_{\text{max}} = 5 \text{ rpm}\). The valve plate of the original hydrostatic displacement unit is replaced by a servo valve, so that pressure changing functions for different valve-plate designs can be imitated.
Research at Technical University of Hamburg-Harburg is conducted focusing on improving unit efficiency in axial piston machines. The calculation of the gap flow as well as the pressure and temperature distribution in the gap between piston and cylinder in swash plate type axial piston machines is deduced [6]. Special pumps to measure the pressure and temperature distribution are designed to verify calculation results. Figure 2 presents a test pump modified from standard axial piston pump. The temperature distribution in the cylinder block and the dynamic pressure inside the displacement chamber under normal operating conditions are measured, which consists with the real working conditions. A similar test pump with sensors mounted on the rotary cylinder block was designed by Monika I. to measure the friction force of the piston/cylinder pair under conditions and operating parameters comparable with series swash plate axial piston machines [7] [8].

Figure 3 is another test pump presented by Monika I. to verify the thermal module of pump simulation software called CASPAR [9] [10] [11]. The pump is the single piston type with a rotating swash plate and can be used to measure the temperature field in the gap of piston and cylinder. To ensure separate suction and discharge process, two check valves are integrated into the test pump.

For a piston pump with rotary speed up to 12000RPM, the oil film field measurement through directly mounting the sensors on the cylinder block will be extremely difficult. Since the high speed piston pump has
a much more compact body than industrial piston pump, it is hard to mount enough pressure sensors and temperature sensors on the cylinder block even use miniaturized sensors. Another negative consideration is to transmit the sensor signal via wireless telemetry at an unconventional high rotary speed. So in this paper, a single piston type test pump is designed, which has a rotary swash-plate and a static cylinder with sensor mounted on it motionlessly.

There exist three main problems to design such a high speed single piston pump. The first one is the rotor dynamic balance, since the asymmetric swash plate rotates at a high speed. The second one is the distributor design due to high running speed that commonly used check valve control as distributor does not function anymore. The third is that the sinusoidal output flow pulsation caused by single piston mechanism. In this paper, the rotor dynamic balance is not discussed although it occupies a lot effort. A separated axial control journal type distributor is present at this paper to ensure the high speed single piston pump run functionally. The fluid elasticity related pressure build-up lag is discussed and an optimization of lag angle is applied to the distributor. The volume of accumulator needed to smooth output flow pulsation is discussed through fluid resistor and capacity analysis. Finally the whole high speed single piston pump is emulational investigated in AMESim software to verify its working performance.

2. THE SPECIFIED SINGLE PISTON PUMP

The purpose of the test pump is to measure the surface temperature distribution and dynamic pressure field in the gap between piston and cylinder as well as the oil film thickness between slipper and swash plate. The measurement of two friction pairs is integrated into one piston machine and can be conducted synchronously. The design parameters of the test pump are listed in table 1.

<table>
<thead>
<tr>
<th>Table 1. Design parameters of the test pump</th>
</tr>
</thead>
<tbody>
<tr>
<td>maximum working pressure (MPa)</td>
</tr>
<tr>
<td>piston diameter (mm)</td>
</tr>
<tr>
<td>maximum rotary speed (RPM)</td>
</tr>
<tr>
<td>piston stroke (mm)</td>
</tr>
<tr>
<td>Displacement (mL/rev)</td>
</tr>
</tbody>
</table>

There are two major differences from former designed pumps for the same measurement purpose. One is the high drive speed of the pump, which is up to 12000rpm. The high rotational speed makes the pump output more power at the same dimension than commonly engine driven pumps, which makes more compact and higher power density hydraulic systems possible, but at the same time makes suction inefficiency to be a problem. The other one is the integrated design that oil film parameters of piston/cylinder pairs and slipper/swash-plate pairs can be measured synchronously.

For those specific measurement requirements, an axial single piston pump as depicted in figure 4 with a separate distributor(7) at shaft end and rotary swash plate(8) in middle of the shaft(1) is designed. Three miniaturized pressure sensors and three thermocouples are mounted on the cylinder to measure the gap pressure and gap temperature. The cylinder block can rotate every 45 degrees to be mounted, so totally 24 points of gap pressure and 24 points of gap temperature can be measured. The output pulse flow caused by single piston movement of piston/cylinder assembly(5) can be guided to the chamber of a steady piston/cylinder assembly(3) with slipper inclined on the swash-plate through a tube. The shaft is supported by housing assembled by the front end cover(2), rear end cover(6) and the housing(4).
3. DESIGN OF FLOW DISTRIBUTOR

A separate flow distributor is designed to connect the piston chamber to either suction port or pressure port synchronously according to the piston movement direction. Because of the high rotational speed of the single piston pump, the valve control as flow distributor is not possible to achieve that task. So a control journal like flow distributor is proposed. The difference to the control journal is that the piston chamber port, suction port and pressure port are all steady whereas the piston chamber which is distributed by control journal rotates with pump shaft.

The structure of the presented distributor is depicted in figure 5. It contains a body(d) which has a pressure port (a), a chamber port(b) and a suction port(c). A rotary shaft(e) slides in the hole of distributor’s body, which switch chamber port(b) to either pressure port(a) or suction port(c). The left part of figure 5 shows the discharge state of the flow distributor when chamber port(b) connects pressure port(a). The right part of figure 5 shows the suction state as chamber port(b) connects suction port(c). So as the shaft(e) rotates one circle, the piston chamber experiences one period of periodic connection.

Figure 5. The principle structure of flow distributor

Figure 6. Dimension of the flow distributor
As to compensate the unilateral load force, two ring grooves are milled in the shaft over an angle the same with the angle that connects either pressure port or suction port as depicted in figure 6. The total hydraulic force on the circular surface of shaft is considered to be zero, only the hydraulic force that applies on the grooves is to be considered. For a balanced hydraulic force state, \( c = 2a \). The dimension \( b \) is so chosen that sealing gap from compensation grooves to low pressure area is in a acceptable length.

Because the actual hydraulic oil is not the ideal fluid, the pressure built-up process in piston chamber should be taken into consideration when design flow distributor. The time for pressure in piston chamber to raise up form suction pressure to discharge pressure \( t_s \) can be expressed in equation (1) as

\[
\Delta p = \Delta p_{\text{out}} - \Delta p_{\text{in}} = \frac{E \cdot \pi r_p^2}{L_0} \int_0^{t_s} \sin \omega t dt
\]

where \( \Delta p_{\text{in}} \) is the suction pressure, \( \Delta p_{\text{out}} \) is the discharge pressure, \( E \) is the bulk modulus of hydraulic fluid, \( r_p \) is the radius of piston, \( \omega \) is the angular velocity of single piston pump, \( L_0 \) is the maximum piston chamber length.

A lag angle for flow distributor to connect the hydraulic port is defined as

\[
\beta = \omega t_s
\]

The flow distributor is so mechanically installed that the discharge port is connected to chamber port when the swash-plate has rotated \( \beta \) angle since piston begins its discharge stroke. The opening area via which the piston chamber port connected to discharge port or suction port is showed in figure 7.

**Figure 7. Suction and discharge opening of flow distributor**

4. FLOW PULSATION ATTENUATION

The output flow of the single piston pump has a large pulsation, which is caused by the linear movement of piston with sinusoidal velocity, as depicted in figure 8. The output flow varies from zero to 49L/min. To measure the properties of oil film, the test pump must work at a specified pressure set up by a relief valve. The relief valve has a minimum working flow-rate as to overcome the nonlinearity caused by spool friction and small valve opening. If the pulsed flow applied on the relief valve, noise and oscillation will occur in the
hydraulic system. So as to conduct the measurement successfully, the pulsed flow which varies from zero to a certain value should be attenuated.

The hydraulic accumulator is commonly used to attenuate the pressure fluctuation in hydraulic system, which in this paper acts as a flow pulsation attenuator. The simplified hydraulic circuit to analysis the flow pulsation attenuate effect is depicted in figure 9. The flow source with a sinusoidal pulsed flow as illustrated in figure 8 is connected to a T junction, whose other two junctions are connected to an accumulator and a relief valve respectively.

Assume $\Delta Q_1, \Delta Q_2, \Delta Q_3, \Delta p$ is small changes of steady input flow $Q$, steady flow of the relief valve $Q_1$, steady flow of the accumulator $Q_2$ and the junction pressure $p$ respectively. The relationship between junction pressure change $\Delta p$ and flow change $\Delta Q_1$ can be written as

$$\Delta Q_1 = C_r \cdot \Delta p \quad (1)$$

where $C_r$ is the relief valve flow rate pressure gradient. The change of the oil volume in accumulator can be expressed as

$$\Delta V_i = \int_{\Delta t} \Delta Q_2 dt \quad (2)$$

which is opposite to the change of gas volume in accumulator

$$\Delta V_g = -\Delta V_i \quad (3)$$

In a small time period $\Delta t$, the expression (2) can be rewritten as
\[ \Delta V_f = \Delta Q_2 \cdot \Delta t \quad (4) \]

The air pressure in the accumulator assumed to be equal with the junction pressure \( p \). Then the air state equation can be expressed as

\[ pV_g^\lambda = p_0 V_0^\lambda = K = \text{constant} \quad (5) \]

where \( V_g \) is the gas volume in accumulator, \( \lambda \) is gas polytropic index, \( p_0 \) is accumulator pre-charge pressure, \( V_0 \) is accumulator’s volume. Then the relationship between the small change of steady junction pressure \( p \) and steady air volume in accumulator can be written as

\[ \Delta p = -\frac{\lambda K}{V_g^{\lambda+1}} \Delta V_g \quad (6) \]

Combine the equation (1) (3) (4) (6) and flow relationship in T-junction, we can get a equation set (7) to describe the flow fluctuation characteristics of the hydraulic system in figure 9. The equation set is presented as

\[
\begin{align*}
\Delta Q_1 &= C_r \cdot \Delta p \\
\Delta V_f &= \Delta Q_2 \cdot \Delta t \\
\Delta V_g &= -\Delta V_f \\
\Delta p &= -\frac{\lambda K}{V_g^{\lambda+1}} \Delta V_g \\
\Delta Q &= \Delta Q_2 + \Delta Q_1
\end{align*}
\]

(7)

Then the small change \( \Delta Q_1 \) can be induced from set (7) as

\[ \Delta Q_1 = R_r \cdot \Delta p = \frac{\Delta Q}{V_g^{\lambda+1}} \quad (8) \]

A block diagram can be written to express equation set (7), as depicted in figure 7.

Figure 7. Block diagram of pulsation attenuation process

For an accumulator that works to attenuate flow pulsation, \( V_g \) keeps almost a constant that equals 70% of the accumulator’s volume. So equation (8) can be change to

\[ \Delta Q_1 = \frac{\Delta Q}{0.7 V_g^\lambda} \quad (9) \]
The $\Delta Q$ and $\Delta t$ of single piston pump can be infer from a 50% pulse ratio square pulsation flow curve that has the same average flow-rate as showed in figure 8. $\Delta Q$ equal the average flow-rate and $\Delta t$ equals half a period.

To get a smaller $\Delta Q$, at the same $\Delta Q$, a bigger $V_0$ and a smaller $C_r$ should be guaranteed, which means larger accumulator and relief valve with larger hydraulic resist effect will lead to a more smooth flow-rate at relief valve.

To verify the correctness of equation (8), the hydraulic circuit in figure 9 is modeled and simulated in AMESim software. The input flow is a sinusoidal pulsed flow as expressed in figure 8. The valve flow rate pressure gradient $C_r$ is set as 30L/min/MPa and $p_s$ is set as 10MPa. The simulation and calculation results of amplitude of $\Delta Q$ are listed in table 2.

### Table 2. Simulation and calculation amplitude of $\Delta Q$, [L/min]

<table>
<thead>
<tr>
<th></th>
<th>$V_0=0.2$L</th>
<th>$V_0=0.5$L</th>
<th>$V_0=1$L</th>
<th>$V_0=2$L</th>
</tr>
</thead>
<tbody>
<tr>
<td>simulation amplitude</td>
<td>7.41</td>
<td>2.98</td>
<td>1.49</td>
<td>0.74</td>
</tr>
<tr>
<td>calculation amplitude</td>
<td>9.10</td>
<td>4.41</td>
<td>2.37</td>
<td>1.23</td>
</tr>
</tbody>
</table>

The simulation amplitude is about 60% of the calculation amplitude when $V_0$ gets bigger, which is cause by the linearization of non-linear gas state equation near a steady state and the approximation of Input flow change $\Delta Q$. For an engineering design process to compensate flow pulsation, the result of equation (9) is acceptable because it means a larger volume will be chosen for the aimed flow pulsation amplitude.

In this design case, a hydraulic accumulator with a volume of 1L is chosen to get a flow pulsation amplitude less than 1.5L/min.

## 5. SIMULATION AND DISCUSSION OF THE SINGLE PISTON PUMP

The whole single piston pump is tested in AMESim software. The simulation model is depicted in figure 11. The model consists of a piston/cylinder pair, two controllable orifices that simulate the oil port connection in flow distributor, and calculation blocks to decide orifice opening and piston linear velocity.

![Simulation model of the single piston pump](image-url)
The simulation parameters are listed in Table 3.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>working pressure (MPa)</td>
<td>15</td>
<td>V₀ (L)</td>
</tr>
<tr>
<td>tank pressure (MPa)</td>
<td>1.5</td>
<td>p₀ (MPa)</td>
</tr>
<tr>
<td>rotary speed (RPM)</td>
<td>12000</td>
<td>Cᵢ (L/min/MPa)</td>
</tr>
</tbody>
</table>

Table 3. Simulation parameters of single piston pump

Figure 12 shows that when the piston begin its suction stroke, the opening area of both suction window and discharge window keep zero, so the piston chamber is a dead volume and its pressure goes down as it expands. When swash-plate has rotated a lag angle $\beta$ since stroke begins, the suction window begins to open, so the suction flow begins as showed in figure 12. A reverse flow occurs when the suction window just open because the pressure in piston chamber is higher than tank pressure, which is the main reason for leakage of distributor. The lag angle $\beta$ is so designed that in single piston pump’s whole working pressure range, the leakage is acceptable. For higher pressure, the volume need to release or raise pressure is larger. So the lag angle $\beta$ is designed according to its medium working pressure. The chamber pressure during suction reaches tank pressure after approximately 75% of the piston stroke, before which the chamber pressure keeps zero. It means cavitation happens, which is evitable for the extreme high speed of single piston pump. The effect of cavitation is reduced when fluid contains less air. During discharge process, the chamber pressure firstly raise up as the dead volume decreases. A reverse flow also happens just when the discharge window opens.

![Figure 12. Piston chamber volume, pressure and flow-rate](image)

The output flow rate of piston pump considering the accumulator as a part of pump at different working pressure is displayed in figure 13. The volume efficiency decreases as the working pressure is loaded. The output flow rate at 20MPa working pressure is about 7.63L/min, from which can be inferred that the volume efficiency is about 56.3%. The usage of the single piston pump is to measure the oil film characteristics, which means the volume efficiency is acceptable in this design. The flow pulsation raises at higher working pressure, which is in accordance with the equation (8). The flow pulsation amplitude at working pressure of 20MPa is less than 9.4% of average flow-rate and acceptable for relief valve load application.
6. CONCLUSION

A high speed single piston pump for oil film investigation is designed. The mechanical structure of the pump to achieve the measurement task is presented, which characteristics at testing piston/cylinder pair and slipper/swash-plate pair synchronously. A separate control journey like rotary flow distributor is designed and the lag angle considering elastic effect of oil is discussed. The hydraulic accumulator that needed to attenuate flow pulsation is mathematically analyzed. The whole pump model is setted up in AMESim software and the simulation result shows the pump has a volume efficiency of 56.3% at 20MPa, 12000 RPM, which mean the designed single piston pump can output enough flow and the oil film investigation can be conducted functionally.

REFERENCES


CONSTANT IMPROVEMENT IN BIO HYDRAULICS – A CHALLENGE, A DREAM OR AN IMPOSSIBILITY?

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ABSTRACT

Bio hydraulic fluids' modern development has been ongoing now over three decades. For traditional hydraulic fluids, based on mineral oils, it took over 40 years to reach their present quality level! So we are getting closer for that period also for biohydraulic fluids. Is there still something that can be done? Some properties, which still can be improved? Some technical aspects, which can be as brilliant as within traditional fluids?

Mother Nature gave us raw materials for bio lubricants, natural esters, which are the best and if we add there a little bit of knowledge and chemistry, where will we end up?

Oxidation stability, cold stability properties, life time, anticorrosion properties, low friction and wear, good water and air separation, hydrolytic stability- the list is pretty long, what we have been able to improve during the years and even in the last 2-3 years!

What is also essentially important: Now we have standardised methods for measuring all possible chemical, physical, technical, environmental, safety properties of bio hydraulic fluids. These reliable facts from the testing are available.

Bio hydraulics excel today with their technical performance with traditional fluids and as an "extra benefit" customers get the environmental and safety properties.

In this paper the latest test results in bio lubricants, the development steps in R&D during the years, comparison to the international bio hydraulic fluids and environmental standards.

KEYWORDS: Bio lubricants, Standards in bio fluids, raw materials in bio lubes, hydraulic stability

1. GENERAL

Where do we need a lubricant? It is needed for example to minimize friction and wear, to remove heat, to transfer power, to prevent corrosion, to help the motion of the solid particles, to remove wear particles, to provide a liquid seal.(1,2)

Bio lubricants “new-coming” have been ongoing for almost forty years. Same time was used to develop mineral oil based lubricants to their best performance.
Bio lubes aren’t any new invention- already 4000 years ago it was understood that by using natural vegetable oils, the friction could be lowered significantly. Well, of course that time the word friction was not familiar nor the reason why so happened! (1,3)

Now we know, that the friction is very low, the wear is minimised, the change of viscosity by temperature is small, the viscosity index is high, the flash point is high, their load carrying ability is great, they have very low volatility etc. We have also learned that the almost perfect molecule that Mother Nature has given to us has its weaknesses like oxidation or thermal stability. But there is always help near: Chemistry!

As we today all know very well the chemistry behind natural esters, we can easily modify these molecules for reducing their weaknesses and leave the excellent properties untouched. That is one big charm of the chemistry and the knowledge in it.

The improvement of bio lubricants’ raw materials have been ongoing step wisely. More You know, more doors are opened to develop the molecules. This will be explained by examples later in this paper.

The analytical methods, which we use in testing the bio lubricants technical and environmental properties has been very important, technical and environmental properties. Their development work during years have been as important as the development of the products itself. (4, 5, 6, 7)

Why it is important continuously discuss and inform about bio lubricants: It is said, that world widely at least 50% of all lubricants end up to the nature: Volatility, spills, total loss applications, via accidental spillage, non-recoverable usage, industrial and municipal waste, urban runoff, refidency process. It is estimated that even 70-80% of mobile hydraulic fluids are entered to the nature. (2) The costs of clean-up caused by mineral oil are very high. So called Holland List specifies the limits, when the soil needs to be purified after oil spill has happened:

- 500 mg/kg of soil, in residential areas and in protected water zones
- 1000 mg/kg for general area: ≤ 5000 mg/kg can be tolerated in areas with no ground water relevance.

It has been estimated that alone in Germany at least 300 000 tn of lubricants are spilled into the nature yearly.(1,8)

It is estimated, that more than 90% of all hydraulic fluids could be manufactured and technically fulfil their task from renewable sources. In Germany there is actually one whole factory, that is only using bio lubricants in all their applications The benefits are for example, that the over usage of fluids reduced significantly, their need for water reduced by 75%, they did not need any more any extra corrosion protectors, their portfolio of needed lubricants reduced from 40 to 5! This factory uses lubricants around 500 tn yearly.(9)

In table 1 there are estimations of the usage of lubricants worldwide in past decade. (10)
Table 1: World wide usage of lubricants and changes in it years 2000 – 2012

<table>
<thead>
<tr>
<th>YEAR</th>
<th>USAGE (mnt)</th>
<th>VARIANCE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2000</td>
<td>36.4</td>
<td>-0.3</td>
</tr>
<tr>
<td>2001</td>
<td>35.6</td>
<td>-2.2</td>
</tr>
<tr>
<td>2002</td>
<td>35.7</td>
<td>0.5</td>
</tr>
<tr>
<td>2003</td>
<td>35.4</td>
<td>-0.8</td>
</tr>
<tr>
<td>2004</td>
<td>36.1</td>
<td>1.9</td>
</tr>
<tr>
<td>2005</td>
<td>36.5</td>
<td>1.0</td>
</tr>
<tr>
<td>2006</td>
<td>36.9</td>
<td>1.2</td>
</tr>
<tr>
<td>2007</td>
<td>37.1</td>
<td>0.5</td>
</tr>
<tr>
<td>2008</td>
<td>36.0</td>
<td>-3.0</td>
</tr>
<tr>
<td>2009</td>
<td>32.2</td>
<td>-10.6</td>
</tr>
<tr>
<td>2010</td>
<td>34.5</td>
<td>7.0</td>
</tr>
<tr>
<td>2011</td>
<td>35.1</td>
<td>1.9</td>
</tr>
<tr>
<td>2012</td>
<td>35.0</td>
<td>-0.5</td>
</tr>
</tbody>
</table>

3. ANALYTICALLY- HOW DO WE TEST BIO LUBRICANTS?

There have been already quite a while several international standards and requirements for different type of biolubricants to guarantee especially their technical performance but as well the environmental and healthy aspects. In table 2 there is a summary of these standards and requirements (1,11)

Table 2: Bio lubricants standards and requirements

<table>
<thead>
<tr>
<th>Standard</th>
<th>Product groups</th>
<th>Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>SS 155434</td>
<td>Biohydraulic fluids</td>
<td>Both technical and</td>
</tr>
<tr>
<td></td>
<td></td>
<td>environmental</td>
</tr>
<tr>
<td>DIN 51524 teil3</td>
<td>Biohydraulics</td>
<td>Technical</td>
</tr>
<tr>
<td>ISO 15380</td>
<td>Bio lubricants, industrial oils and</td>
<td>Technical</td>
</tr>
<tr>
<td></td>
<td>related products</td>
<td></td>
</tr>
<tr>
<td>Volvo VCE 1286,1</td>
<td>Biohydraulics</td>
<td>Technical</td>
</tr>
<tr>
<td>Caterpillar BF-2</td>
<td>Biohydraulics</td>
<td>Technical and</td>
</tr>
<tr>
<td></td>
<td></td>
<td>environmental</td>
</tr>
<tr>
<td>VDMA 24568/24569</td>
<td>Biohydraulic fluids, bio total</td>
<td>Technical, environmental</td>
</tr>
<tr>
<td>EU Ecolabel</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>loss lubricants, bio greases</td>
<td>and healthy</td>
</tr>
<tr>
<td>German Blaue Engel</td>
<td>Biohydraulics, biogreases</td>
<td>Technical and</td>
</tr>
<tr>
<td></td>
<td></td>
<td>environmental</td>
</tr>
<tr>
<td>Nordic White Swan</td>
<td></td>
<td>Technical, environmental</td>
</tr>
<tr>
<td></td>
<td></td>
<td>and healthy</td>
</tr>
<tr>
<td>Dutch VAMIL</td>
<td>Biolubricants, biogreases</td>
<td>Environmental</td>
</tr>
<tr>
<td>Green Cross(USA)</td>
<td></td>
<td>Environmental</td>
</tr>
<tr>
<td>Ecomark (Japan)</td>
<td></td>
<td>Environmental</td>
</tr>
<tr>
<td>Maple Leaf(Canada)</td>
<td></td>
<td>Environmental</td>
</tr>
<tr>
<td>UK’s Environmental Agency</td>
<td>Chain saw oils, biohydraulics</td>
<td>Technical, environmental</td>
</tr>
</tbody>
</table>
Inside these standards there are many standardized methods and analysis with acceptance limits. These methods have been developed to show and prove, that bio lubricants’ technical performance is ok and has been improved significantly during the years.

For example hydrolytic stability – how stable the bio lubricants are against water or what happens to bio lubes in the presence of water. Why this important to test is clarified in Figure 1. Esterification reaction is simplified in this figure by showing, that when a fatty acid reacts with an alcohol in equilibrium reaction there form an ester and water. As this reaction is an equilibrium reaction, it means, it can go also back to reagents direction, meaning when water reacts with an ester, fatty acid and alcohol can be formed. Then the acid number of the product is higher and there form products, which can cause for example serious corrosion problems.

\[
R' - COOH + R'' - OH \rightarrow R' - COO - R'' + H_2O
\]

Figure 1: A simplified esterification reaction

How this unwanted reaction can then be avoided? Well the answer is once more: By chemistry. You need to know how to build Your ester molecule so that it is not that sensitive against water.

During the past decades there have been several hydrolytic stability tests, which reproducibility and reliability in general have not been good. For example of tests is Beverage Bottle Method (ASTM D 2619). (12)

We have finally developed and standardized a test, which we can rely on and which shows the hydrolytic stability of a bio fluid. This test was developed by Swedish authorities and in this developing process there were involved many research institutes, bio fluids’ producers and huge amount of ring tests were done. This test is now known as SS 5515181 standard. In the test to 250 ml of oil 25 ml (10%) of water is added in water lock bottle, which is shown in figure 2. The glassware is kept in 90 °C. The change of acid value (mgKOH/g) is measured after 120 hours and after 192 hours. The smaller the change, the better resistance the tested oil has to react with water. In Figure 2 is shown the glassware in which the test is done and in figure 3 are test results with different bio hydraulic fluids done by that test. (13)

Figure 2: The glassware used in SS 5151581 hydrolytic stability test
Table 4: A general example how a recipe of a bio hydraulic fluid can look like (13)

<table>
<thead>
<tr>
<th>Raw material/ Additives</th>
<th>Description</th>
<th>Weight percent</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ester</td>
<td>Either unsaturated or saturated</td>
<td>94-99.39</td>
</tr>
<tr>
<td>Antioxidant</td>
<td>Amine or/and Phenolic type</td>
<td>0.5-2.0</td>
</tr>
<tr>
<td>Anticorrosion</td>
<td>Sulphur/Nitrogen cont. compounds</td>
<td>0.05-1.0</td>
</tr>
<tr>
<td>Cold stability improvers</td>
<td>Acrylic polymers</td>
<td>0.05-2.0</td>
</tr>
<tr>
<td>Antifoam agents</td>
<td>Silicone type chemistry</td>
<td>0.01-1.0</td>
</tr>
</tbody>
</table>

5. LATEST DEVELOPMENT RESULTS FROM BIO LUBRICANTS

In total loss bio lubricants in conveyour oils a huge challenge has been hanging in air for years: How to improve their cold stability properties, so that they stay fluidly min up till -30°C for weeks. The pour point of normal rapeseed oil varies from quality and year being around -10°C to – 15°C. This challenge has been finally answered both by improving the bio lubes cold stability properties and also by developing a reliable test method to test these properties at labscale.

Table 5: Test results from cold stability testing in laboratory for different biolubes and for comparison a mineral oil based conveyour oil (13)

<table>
<thead>
<tr>
<th>Product</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Days in -30°C</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3 fluidy</td>
<td>crystals on</td>
<td>fluidly</td>
<td>fluidly</td>
<td></td>
</tr>
<tr>
<td>5 fluidy</td>
<td>thick crystal</td>
<td>fluidly</td>
<td>fluidly</td>
<td></td>
</tr>
<tr>
<td>7 fluidy</td>
<td>some crystals</td>
<td>some crystals</td>
<td>stuff fluidy</td>
<td></td>
</tr>
<tr>
<td>10 fluidy</td>
<td>a lot of</td>
<td>solid</td>
<td>almost solid</td>
<td></td>
</tr>
<tr>
<td>14 fluidy</td>
<td>solid</td>
<td>solid</td>
<td>almost solid</td>
<td></td>
</tr>
</tbody>
</table>

1 = Tenalube CVO, ISO VG 68
2 = Bio Conveyour oil, ISO VG 68
3 = Bio Conveyour oil, ISO VG 68
4 = Conveyour oil based on recycled mineral oil, ISO VG 46
It is very well known by theory and now as well proved in real saw mill, that vegetable oil based conveyour oils lubricates much better than conventional mineral oils. This means that the higher purchasing costs for bio lubricants can be compensated and as well money saved by using much better lubricating bio lubricant. The needed amount varies from application area being 60 - 70% less oil need with bio lubricant when compared to recycled mineral oil based product.

In hydraulic fluids like explained earlier the hydrolytic stability has been improved vastly, also the seal compatibility has been improved significantly. It has also claimed, that bio hydraulic fluids are more sensitive to soft, yellow metals. Therefore a new corrosion test has been developed together with additive company for zink corrosion measurement.

In Nordic countries outside mobile hydraulics the cold stability properties and their stability are very important. How fluidy the oil is in cold temperatures and how fluidy it will stay when the cold temperature last for several weeks?

We have to keep in mind, that when improving for example the above mentioned properties, this does not mean that average good or already excellent properties of bio hydraulics are changed! On the contrary, all of them are also at least a little bit enhanced, if possible. In table 6 there are the latest development work results in a bio hydraulic fluids, ISO VG 46. (17)
### Table 6: Latest development work results on Bio Hydraulic Fluid, ISO VG 46, Terralube HFO

<table>
<thead>
<tr>
<th></th>
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</tr>
</thead>
<tbody>
<tr>
<td>(mm³/s)</td>
<td>(mm³/s)</td>
<td>(°C)</td>
<td>(mm/s)</td>
<td>(min)</td>
<td>ΔAV(mgKOH/g)</td>
<td>(min)</td>
<td>ΔAV(mgKOH/g)</td>
<td>(h)</td>
<td>Δvisc.40°C (%)</td>
<td>(ppm)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>49,7</td>
<td>10,2</td>
<td>199</td>
<td>265</td>
<td>550</td>
<td>1275</td>
<td>3465</td>
<td>5350</td>
<td>-57</td>
<td>no</td>
<td>&lt;1</td>
<td>0,42</td>
<td>0,88</td>
<td>10</td>
<td>0,29;5,4</td>
<td>2570</td>
<td>&gt;12</td>
<td>0</td>
<td>1a</td>
<td>0</td>
<td>330</td>
</tr>
</tbody>
</table>
Even if we have very good standards to test our bio lubes at laboratory, we still do need tests in bigger scales. We have in our production facility a test pump running and working 24/7. These test results have been compared to real life field test and this pump test has been proved to be 7 to 10 times harder for the tested oil than real life testing.

Last but not least before a biohydraulic fluid is ready to be launched to the market, it goes to a field test. This field test is run in a real mobile hydraulics and after certain hours in use, a sample is fully analyzed at the laboratory.

Table 7: Some test results from a field test from ISO VG 46 bio hydraulic fluid. Change of viscosity at 40°C(%) and change of Total Acid Number (mgKOH/g) by hours in use.(13)

<table>
<thead>
<tr>
<th>A(MGKOH/G)</th>
<th>99 199 309 412 554 1160</th>
</tr>
</thead>
<tbody>
<tr>
<td>H</td>
<td>0 99 199 309 412 554 1160</td>
</tr>
</tbody>
</table>

As from the field test results can be seen, the tested hydraulic fluid, Terralube HFO, ISO 46, has been nearly unchangeable during one year test!

6. CONCLUSION

As a researcher’s point of view, we have made huge progress in bio lubes R&D work as well in their analytics in past years. Bio lubricants’ properties are at least as good as mineral oil based comparable products. In many cases already they are far better than comparable products. Now our main task is to pass this information through out the world. There have been bad, even miserable bio products in the market in the past years. The mistakes made, have the bad reputation of this whole bio lubricants’ branch last many years and it is very hard work for us, bio lubes producers to convince the potential customers, that we also have learnt from mistakes and we also have made giant steps in our development work and our knowledge has grown hugely during the learning years. The only way to show this and convince them is by facts, which some of them are shown in this paper. One step ahead is the slogan, which we have all these years obeyed!
REFERENCES:


Biodegradability is a process, in which organic compounds disperse by enzymes. These enzymes are produced by living organisms. The determination of ultimate degradation is mineralization of substances to CO₂, H₂O and biomass. It does not mean that a formation of CO₂ would be an indicator for complete degradation of the molecule. It is only an indicator for the complete degradation of one carbon molecule!

By ISO 9439 ultimate degradation is defined as "Breakdown of a chemical compound or organic matter by micro-organisms in the presence of oxygen to carbon dioxide, water and mineral salts of any other elements present (mineralization) and to produce of new biomass. (1,14)"

We should be careful, when talking about biodegradability and products, which are claimed to biodegrade. The actual term today is greenwashing. While many products claim to biodegradable, the question is how quickly they do biodegrade and to what? There are products that are inherently biodegradable or readily biodegradable. This means that a product which are claimed to be biodegradable can persist in the environment for years. (15)

In table 3 there are shown different biodegradation methods and their limits to acceptance. (16)
4. CHEMICALLY - WHAT DO BIO LUBRICANTS ACTUALLY CONTAIN?

The raw materials in Total Loss Biolubricants are mainly (generally far over 95%) different vegetable oils. The most common are rapeseed-, soybean- and sunflower oils. Depending on their technical application there are also additives - to improve cold stability behaviour, for better oxidation stability, anticorrosion, emulsifiers, tackifiers etc.

Raw material quantity is in bio hydraulics generally 96.5 - 98.9 percent, so it plays a huge role in what kind of properties the final product will have. The choice of raw material will have at least an affect to volatility, oxidation stability, thermal stability, anti-wear characteristics, seal material compatibility, hydrolytic stability, thermal stability, anti-wear characteristics, seal material compatibility, hydrolytic stability, friction behavior. Bio lubricants do as well transfer heat from the surfaces much quicker than traditional fluids. (1,2)

For hydraulic fluids mainstream in raw materials are synthetic esters, which have been produced from natural esters with different chemical reactions. The development of these esters has also been a stepwise process. The more we know - the easier to make right reactions. For example the length of the carbon chains reflects in viscosity, viscosity index, compatibility of the final product. Branching of the carbon chain affects to pour point, oxidation stability and biodegradability of the final product. The acid value i.e steric hindrance affect to thermal stability and hydrolytic stability. First good esters for bio lubricants were for example polyol esters. They can be produced either chemically or enzymatically. Polyol esters do have very good thermal and oxidation stability and hydrolytic resistance. Next development step in esters were so-called complex esters. They were produced by several chemical reactions. They have very good lubricity, excellent thermal and oxidation stability and low temperature properties. Another approach to ester production is the usage of high oleic acid contain fatty acids as raw material. This gives better oxidation stability. (1)

REACH and CLP have had a huge impact to bio lubricants’ additives. The rules what You can have in Your bio products without any labels on them are very strict. When You are calculating the total effect of Your products to environment and/or to healthy You have to use certain summary system where everything is affect to everything and You have to always take the worst scenario in account. This puts bio lubricants producers inside REACH area to very difficult position! And this brings quite many new challenges and cost for additive producers.
APPLICATION OF NANO-STRUCTURED COATINGS TO THE HEAT TRANSFER SURFACE OF HEAT EXCHANGERS

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ABSTRACT

Traditionally, the attempts to improve the heat exchangers performances focused on the increasing of heat transfer surface according to the need to keep overall dimensions and pressure drop restrained and heat transfer efficiency high. Now that the relationship between surface width, geometrical efficiency and dimensions is reaching its maximum limits, the attention begins to move on the increase of the heat transfer coefficient. Under this condition, the proposal of the usage of nano-structured coatings is to induce the property of super-hydrophobicity to the surfaces in contact with the fluids involved in the heat transfer. This property is able to produce an increasing of the heat transfer coefficient and a decrease of power losses caused by friction by intervening on thermal and dynamic boundary layer. The technology developed is based on the principle which in nature is known as “Lotus effect” to obtain a super-hydrophobic surface. Moreover, modifying a phase in the production process is possible to produce a hydrophilic behavior, depending on the customer’s needs. The application of this technology to real products with measurable targets will open the doors to an entire new generation of products to be applied in fluid power systems of mobile machinery where the efficiency and dimensions constraints are in many cases critical. The paper is aimed to determine the thermal and fluid-dynamic performances of heat exchangers with super-hydrophobic surfaces. A comparison between the nano-coated heat exchanger and the “traditional” one will be shown in order to evaluate the given improvement in terms of: heat transfer coefficient, pressure drop reduction, industrial feasibility. The solution proposed by the authors is patent pending.

KEYWORDS: nano-structured coatings, super-hydrophobic, heat exchangers

1. INTRODUCTION

Most of the surveys on the effects caused by slip flow have been based on flows in micro-channels and nano-channels [1], [2] because in scales larger than the millimetric one, these effects are not visible. This paper discusses these effects on millimetric scales ducts coated by a special class of materials called super-hydrophobic which are able to reduce wettability of fluids in contact to this kind of surfaces and so to switch the viscous friction behavior from sliding to rolling. This is much evident as fluid contact angle increases. The effects on heat transfer are the increase of the heat transfer coefficient because of an induced temperature gradient grow to the wall. The super-hydrophobic coating is achieved by synthesizing via sol-gel a nano-structured ceramic layer and functionalizing it with fluoroalkysilane (FAS) able to induce such a low wettability that corresponds to about 150° contact angle. Drag reduction in turbulent flows can be achieved through a number of different mechanisms. However, in pressure-driven laminar flows, the use of super-hydrophobic surfaces represents one of the first technologies capable of reducing drag in devices that are
larger than the molecular scale. The development of these surfaces (given their ability to produce significant drag reduction over a broad range of Reynolds numbers, both laminar and turbulent) could profoundly affect a variety of important existing technologies [3]. A simplified analytical model is described below; its results are just theoretical and must be verified experimentally especially because it does not take into account many thermo-fluid dynamic effects that occur in practice.

2. THEORETICAL BASIS

The impossibility to integrate Navier-Stokes equations to obtain an analytical solution for a general fluid motion is now universally acknowledged. This is the main reason why the numerical discretization models have been studied and refined for five decades of work. But when applying these equations to laminar motion, the problem is simplified so much that becomes possible to achieve an analytical solution from the system of differential equations that describe the fluid motion.

2.1. The slip boundary condition model

Fig. 1 shows a schematic of the 2-D velocity profiles of a fluid in relative motion between two plates. As documented by [4], in presence of slip flow the velocity value near the wall is not null but equals to a slip velocity \(u_s\). By lengthening the velocity profile in slip flow condition to the vertical axis, it derives an intersection which is distant from the wall of a quantity called slip length. The model made by [4] specifies a relationship between slip velocity and slip length as follows:

\[
\frac{u_s}{L} = \left| \frac{\partial u}{\partial y} \right|
\]  

(1)

where \(L\) is the slip length and \(\frac{\partial u}{\partial y}\) is the velocity gradient with respect to the vertical axis \(y\).

![Fig. 1. Schematic depiction of no-slip flow (on the left) and slip flow (on the right) boundary condition.](image)

2.2. Slip length vs. contact angle

A survey carried out by [5] provides a correlation between static contact angle and slip velocity. They used water as working fluid and both organic and inorganic hydrophobic surfaces. The results they found seem to be consistent with the following quasi-universal relationship:

\[
L \propto (\cos \theta_c + 1)^{-2}
\]  

(2)

where \(\theta_c\) is the static contact angle that a liquid droplet forms with the solid surface which is placed on, and \(L\) is the slip length.
2.3. Slip velocity vs. slip length

It is possible to obtain the velocity profile of the fluid in a circular cross-section duct by introducing the following hypotheses:

- Axisymmetric flow;
- Fully developed flow;
- Newtonian fluid;
- Independence on gravitational effects.

Applying these hypotheses to Navier-Stokes equation expressed in cylindrical coordinates, the following expression is obtained:

\[ u(r) = u_c \cdot \left(1 - \frac{r^2}{R^2}\right) \]  

(3)

where \( u_c \) is the duct centerline velocity, which is the maximum velocity, \( R \) is the cross section radius, \( r \) is the radial coordinate of the cross section. Eq. (3) is obtained considering null the velocity at the wall. Applying the slip boundary condition to eq. (3), it becomes:

\[ u(r) = u_c \cdot \left(1 - \frac{r^2}{R^2}\right) + u_s \cdot \frac{r^2}{R^2} \]  

(4)

where \( u_s \) is the slip velocity at the wall. Applying eq. (4) to eq. (1) is possible to obtain a relationship between slip velocity and slip length. For a parabolic velocity profile eq. (1) must be calculated for \( r = R + L/2 \) because the gradient of the line passing through two points is equal to the mean of the gradients of the lines tangent to the parabola in those two points. The expression obtained is the following:

\[ \frac{u_s}{u_c} = \frac{L}{R} \cdot \left(2 + \frac{L}{R}\right) \]  

(5)

2.4. Calculation of viscous friction factor

The law that relates viscous shear stress at the wall to velocity gradient is the Newton law, as follows:

\[ \tau_w = \mu \cdot \frac{\partial u}{\partial r} \cdot \frac{1}{R} \]  

(6)

where \( \tau_w \) is the shear stress at the wall, \( \mu \) is the fluid dynamic viscosity, \( \partial u/\partial r \) is the velocity gradient calculated at the wall. In relation to the velocity profile expressed in eq. (4), shear stress assumes the following formulation:

\[ \tau_w = 2 \cdot \mu \cdot \frac{u_c}{R} \]  

(7)

Applying a force balance between viscous shear stress (\( \tau_w \)) and pressure drop caused (\( \Delta p \)), the following relationship derives:

\[ \Delta p \cdot \pi \cdot R^2 = \tau_w \cdot 2 \cdot \pi \cdot R \cdot l \]  

(8)

where \( l \) is the duct length that experiences the pressure drop and \( R \) is the duct radius. The expression that relates the friction pressure drop in a straight duct to the friction factor (\( f \)) and the flow velocity is the following:

\[ \Delta p = f \cdot \frac{l}{D} \cdot \rho \cdot \frac{W^2}{2} \]  

(9)

where \( \rho \) is the density of the fluid, \( W \) is the mean velocity of the fluid, \( l \) is the duct length and \( D \) is the duct hydraulic diameter. The fluid mean velocity (\( W \)) is found calculating the mean function of eq. (4):
\[ W = \frac{1}{\pi \cdot R^2} \cdot 2 \cdot \pi \cdot \int_0^R \left[ u_c \cdot \left( 1 - \frac{r^2}{R^2} \right) + u_s \right] \cdot r \cdot dr = u_s + \frac{u_c}{2} \]  

(10)

Inserting eq. (9) and eq. (10) in eq. (8) the expression of friction factor in presence of a non-zero slip velocity is achieved:

\[ f = \frac{64}{Re} \cdot \left( \frac{1 - \frac{u_s}{u_c}}{1 + \frac{u_s}{u_c}} \right) \]  

(11)

where \( Re \) is the fluid Reynolds number in the duct. By eq. (11) it can be noticed that the new found friction factor would be equal to that corresponding to a no-slip boundary condition in laminar flow, that is \( f_{NS} = \frac{64}{Re} \). Thus the friction factor in a slip condition \( f \) is related to the one in a no-slip condition \( f_{NS} \) as follows:

\[ f = f_{NS} \cdot k_{slip} \]  

(12)

where:

\[ k_{slip} = \frac{1 - \frac{u_s}{u_c}}{1 + \frac{u_s}{u_c}} \]  

(13)

2.5. Calculation of Nusselt number

As well as in case of Navier-Stokes equation, the energy conservation equation in differential form is impossible to solve analytically for any kind of flow motion but, introducing the same hypotheses as in the case of fluid motion equation plus the uniformity of the wall heat flux, it becomes possible. To calculate Nusselt number in a circular cross section straight duct in fully developed laminar flow, it is necessary to solve analytically this differential equation. Applying the hypotheses above, the following simplified energy conservation equation is obtained:

\[ u \cdot \frac{\partial T}{\partial z} = \frac{\alpha}{r} \cdot \frac{\partial}{\partial r} \left( r \cdot \frac{\partial T}{\partial r} \right) \]  

(14)

where \( z \) is the axial coordinate of the duct, \( r \) is the radial coordinate, \( T \) is the temperature as a function of \( r \) and \( z \), \( \alpha \) is the thermal diffusivity of the fluid, \( u \) is the velocity as a function of \( r \) and \( z \). Applying the velocity function in eq. (4), the Fourier’s law of thermal conduction and the hypothesis of wall heat transfer uniformity to eq. (14), the temperature profile is obtained as follows:

\[ T(r) = T_w - (T_w - T_c) \cdot \left[ 1 - \left( \frac{1 + \frac{u_s}{u_c}}{3 + \frac{u_s}{u_c}} \right) \cdot \frac{r^2}{R^2} - \frac{r^4}{4 \cdot R^4} \right] \]  

(15)

where, \( T_w \) is the temperature at the wall, \( T_c \) is the temperature at the center-line of the duct, \( u_s \) and \( u_c \) are the slip velocity and the center-line velocity. Applying eq. (15) and the definition of bulk temperature to Newton’s law of cooling the Nusselt number as a function of the ratio of slip velocity and the maximum velocity \( (u_s/u_c) \) is obtained:

\[ Nu = \frac{48 \cdot \left( 1 + \frac{u_s}{u_c} \right)^2}{11 + 10 \cdot \left( \frac{u_s}{u_c} \right) + 3 \cdot \left( \frac{u_s}{u_c} \right)^2} \]  

(16)
2.6. Theoretical results

Fig. 2 shows the variation of friction factor related to the slip/max velocity ratio. It shows that in order that the friction factor varies significantly, slip velocity should be at least 0.01 times greater than the maximum velocity at the center-line of the duct.

![Fig. 2. Variation of the fiction factor vs. slip/max velocity ratio.](image)

Similarly as the variation of friction factor, Fig. 3 and Fig. 4 show that the Nusselt number variation becomes appreciable when \( u_s/u_c \) is greater than 0.1. In this case it is visible a maximum increase of the Nusselt number of 83%.

![Fig. 3. Nusselt number vs. slip/max velocity ratio.](image)

In Fig. 5 is drawn the function expressed in eq. (5). It shows that to reach a slip/max velocity ratio value equal to 0.1, slip length should be greater than 40% of the pipe radius.

The proportionality relationship expressed in eq. (2) can be transformed to an equation if a multiplier constant \( k_{θL} \) is introduced:

\[
L = k_{θL} \cdot (\cos θ_c + 1)^{-2} \quad (\text{nm})
\]  

(17)

Varying the value of \( k_{θL} \), the slip length versus contact angle curve can be traced. In Fig. 6 three curves corresponding to three different values of \( k_{θL} \) parameter are shown. Since the existence condition of eq. (17) is \( \cos(θ_c) \leq 1 \), the contact angle value must satisfy the following condition:

\[
k_{θL} \leq 4 \cdot L \quad (\text{nm})
\]  

(18)
Fig. 4. Nusselt number relative variation vs. slip/max velocity ratio.

Fig. 5. Variation of the slip/max velocity ratio vs. slip length/pipe radius ratio.

Fig. 6 shows the extremely high gradient of slip length curve at contact angle values near to perfect hydrophobicity. Thus, if the order of magnitude of the duct radius is 1 mm, to accomplish the slip length/pipe radius ratio equal to 0.4, the slip length order of magnitude should be equal to \(4 \times 10^5\) nm that corresponds to a contact angle greater than 170°.

Fig. 6. Variation of the slip length vs. contact angle.
By this theoretical model we can deduce that the advantages achievable with the use of a hydrophobic surface are remarkable only for very high contact angle values. The elevated gradient of slip length curve implies a great variation of slip velocity against a small change of contact angle. In fact the contact angle varies with respect to the temperature of the fluid.

3. EXPERIMENTS

The experiments are essentially based on the measurement of three physical quantities: volume flow rate, static pressure upstream and downstream the heat exchanger (hot fluid side) and static temperature upstream and downstream the heat exchanger, both hot and cold fluid side. Furthermore, knowing the thermo-physical properties of the process fluid adopted for the tests (hydraulic oil ISO VG46), it is possible to extrapolate the overall viscous friction factor ($f$) and the overall heat transfer coefficient ($U$), as follows:

$$f = \frac{2 \cdot S_f^2 \cdot \Delta p_m}{\rho(T) \cdot Q^2} \quad (19)$$

$$U = \frac{\rho(T) \cdot Q \cdot c_p(T) \cdot \Delta T_m}{S_h \cdot \Delta T_{ml} \cdot F(T)} \quad (20)$$

where $S_f$ is the flow through surface at the inlet of the finned part of the heat exchanger, $\Delta p_m$ is the static pressure difference measured upstream and downstream the heat exchanger, $\rho(T)$ is the fluid mass density of the as a function of the fluid instant temperature, $Q$ is the fluid volume flow rate, $c_p$ is the fluid specific heat capacity at constant pressure, $\Delta T_m$ is the measured temperature difference between upstream and downstream the heat exchanger, $S_h$ is the heat transfer surface area of the heat exchanger finned part, $\Delta T_{ml}$ is the logarithmic mean temperature difference, $F$ is a correction factor [6] depending on the heat exchanger shape and the way the fluids interact thermally and dynamically each other. Eq. (20) is based on the hypothesis of perfect heat transfer between the fluids, without considering the natural convective and radiative heat losses with the surrounding ambient. This means that the whole thermal energy owned by the hot fluid (hydraulic oil) is considered completely transferred to the cold fluid (ambient air).

3.1. The test bench

The bench utilized for the tests is powered by two volumetric pumps, one gear type, one rotary vane type. The rotary vane one supplies hydraulic oil to the system (hydraulic diagram in Fig. 7) at the service of the heat exchangers to be tested. The gear pump services other components of the bench like solenoid valves pilot circuits. The sensors are connected to a data acquisition unit controlled by a notebook through a control software.

![Fig. 7. The test bench.](image)
3.2. The heat exchangers

Two heat exchangers have been tested for the experiments aimed to determine the entity and the
differences of heat transfer coefficient and pressure drops caused by friction losses:
  • a standard crossflow heat exchanger (Fig. 9a);
  • a functionalized one with nano-structured coating (Fig. 9b).
Both heat exchangers are basically constituted by a core part (finned) which is the one where takes place
most of the heat transfer and an attachment part which is the one that connects the core to the hydraulic
pipes. The standard heat exchanger has been wholly assembled by brazing, on the other hand the core part
of the functionalized one, after having been assembled by brazing, has been immersed in the sol-gel by
using a technique called “dip-coating”, then it has been connected to the attachments by bonding with an
epoxy resin to avoid any damage to the coating due to the intensive heat produced in the brazing phase of
the process. Both of them are made in 3003 aluminum alloy.

![Diagram](image1)

Fig. 8. Hydraulic diagram of the supply system to the heat exchanger.

![Images](image2)

Fig. 9. Crossflow standard (a) heat exchanger and functionalized one (b).
3.3. Working conditions

The test has been performed in stationary regime referring to the volume flow rate as control variable. Volume flow rate had to be varied in the range of 1 to 10 liters per minute; above 10 l/min excessive fluid-dynamic dissipation phenomena, that would prevent to see the effects of the nano-structured coating, would occur. The cold fluid (ambient air) side of the heat exchanger works in suction condition so that air flow is the most uniform as possible. The flow is put in motion by a hydraulically powered dynamic fan. To control air volume flow rate, the oil flow rate in the hydraulic motor (the fan driver) has been controlled, by using a globe valve, to be equal for both heat exchangers. In fact they have been tested singularly so the highest attention has been paid to repeat the same working conditions for each test.

4. RESULTS

It is reported below a summary of the results achieved from the test operated at 70 °C inlet oil temperature. The correction factor (F) can be approximated to 1 because of the lower value assumed by the air heat capacity rate in comparison to the oil one. As for the calculation of the measurement uncertainty a confidence level, related to a Gaussian distribution of probability, of 95% has been adopted in relation to an uncertainty equal to 1.96 by the standard deviation.

Table 1 and Fig. 10 show three different pressure drop-flow rate variation behaviors:
1. Standard heat exchanger: parabolic function up to 4 l/min;
2. Standard heat exchanger: linear function over 4 l/min;
3. Functionalized heat exchanger: parabolic function along the whole flow rate range.

The first behavior reveals the presence of turbulent flow that is thought to be a fraction of the whole flow (transitional flow), also by virtue of what happens to the heat transfer coefficient for a flow rate up to 4 l/min (Fig. 11), in fact in that region heat transfer seems to be very unstable, in particular for what concerns the standard heat exchanger.

The second behavior is relative to a direct proportionality relationship between pressure drop and flow rate, so laminar flow occurs. Looking at the standard heat exchanger curve in Fig. 11, after 6 l/min heat transfer coefficient stabilizes to a constant value not depending on flow velocity. This is a typical laminar flow characteristic.

Table 1. Results of the test at 70 °C inlet oil temperature.

<table>
<thead>
<tr>
<th>Volume flow rate (l/min)</th>
<th>Standard heat exchanger</th>
<th>Functionalized heat exchanger</th>
<th>Percentage gain</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Pressure drop (Pa)</td>
<td>Heat transfer coefficient (W/m² K)</td>
<td>Pressure drop (Pa)</td>
</tr>
<tr>
<td>1</td>
<td>1683</td>
<td>99.41</td>
<td>3797</td>
</tr>
<tr>
<td>2</td>
<td>4584</td>
<td>11.33</td>
<td>5542</td>
</tr>
<tr>
<td>3</td>
<td>8214</td>
<td>67.89</td>
<td>7463</td>
</tr>
<tr>
<td>4</td>
<td>13474</td>
<td>125.67</td>
<td>10772</td>
</tr>
<tr>
<td>5</td>
<td>18897</td>
<td>93.71</td>
<td>13598</td>
</tr>
<tr>
<td>6</td>
<td>24879</td>
<td>62.93</td>
<td>17444</td>
</tr>
<tr>
<td>7</td>
<td>31045</td>
<td>58.96</td>
<td>20821</td>
</tr>
<tr>
<td>8</td>
<td>38186</td>
<td>57.14</td>
<td>24233</td>
</tr>
<tr>
<td>9</td>
<td>43818</td>
<td>58.50</td>
<td>28769</td>
</tr>
<tr>
<td>10</td>
<td>50810</td>
<td>59.57</td>
<td>33663</td>
</tr>
</tbody>
</table>

This anomaly could be explained considering laminar-turbulence transition not in the context of a constant cross section duct but where turbulence is geometrically induced by the heat exchanger fins that are just aimed to interrupt thermal boundary layer and so to increase heat transfer coefficient. Fluid-dynamically speaking, fins are eddy promoters.
The third behavior shows, in the functionalized heat exchanger, the presence of turbulence for each value of volume flow rate as if the Superhydrophobic coating promotes turbulent flow in spite of laminar flow but also in this case heat transfer coefficient seems to reach a constant value. Between 4 and 6 l/min a leap from transitional turbulence to laminar seems to occur.

The variability of results, especially visible in Fig. 11, is due to laminar-turbulent transitions caused by the highly irregular geometry into which oil must flow. In fact turbulence is often caused by obstacles fluid encounters in its flowing that occurs even at very low Reynolds number values: 100 - 400 instead of 2300 – 4000 in constant cross section pipes.

![Fig. 10. Pressure drops vs. volume flow rate of the standard and functionalized heat exchangers at 70 °C inlet oil temperature.](image)

![Fig. 11. Heat transfer coefficient vs. volume flow rate of the standard and functionalized heat exchangers at 70 °C inlet oil temperature.](image)

5. CONCLUSIONS

It has been derived analytically a fully developed laminar Nusselt number and viscous friction factor that take into account the effects of a slip velocity boundary condition. It has been shown experimentally that heat transfer coefficient increases non-negligibly relatively to that corresponding to non-slip flow boundary conditions, and that pressure drop due to viscous dissipation in a nano-coated heat exchanger are firstly greater but then lower than that occurring in the standard one. It must be highlighted that in § 2 it has not been taken into account the effects of temperature in the formulation of friction factor and the Nusselt number. More important, the theoretical results do not refer to a transition turbulent flow as it is likely that occurs in the real case. Thus this issue has to be further analyzed with numerical discretization models able
to solve the problem that includes the turbulent transition flow. It is not fully understood so far why laminar flow is promoted by higher flow rate values in spite of turbulent flow. Further experiments are in progress to amplify the effects of this patent pending solution by using bigger heat exchangers so to take advantage of the scale factor. Also to characterize slip flow effects on heat transfer at higher temperatures, to improve the bench setup and to make measurements better revealing by testing several more samples of heat exchangers.

REFERENCES


CONTROL OF A SEMI-BINARY HYDRAULIC FOUR-CHAMBER CYLINDER

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ABSTRACT

In the past years there have been several publications about digital fluid power including multi-chamber cylinders. Multi-chamber cylinders are typically controlled by switching valves that connect the cylinder chambers to one of the supply pressures (often revered to as digital cylinder). This configuration allows for significant reduction in energy consumption and has the potential to recover energy from the load. However the number of discrete force levels that can be produced by this concept is limited. The target of this study was to improve the force resolution to achieve accurate control of the cylinder. For this, one of the chambers is no longer connected to the supply pressures by a switching valve but with a proportional valve. Control strategies for pressure, force and position control were developed and successfully implemented on a test bench at Bosch Rexroth. The laboratory measurements showed that compared to the digital cylinder with the new approach the controllability of the load/cylinder (speed and force resolution) can be improved without losing much of its energy efficiency.

KEYWORDS: Digital fluid power, four-chamber cylinder, fast switching valves, secondary control

1. INTRODUCTION

Digital fluid power is a technology under development that has seen a growing number of contributions and publications by several research groups from all over the world in the last decade [1]. This technology encompasses valves, pumps, actuators and transformers. Fault tolerance and energy efficiency are becoming more and more important for modern machines, especially in the area of mobile hydraulics. In this area the multi-chamber cylinder offers interesting possibilities, as its structure enables energy recovery. Big (or many small) fast switching valves furthermore allow a reduction in energy losses (due to reduced throttling losses). Research has shown that a system involving a four chamber cylinder with two supply pressures connected by fast switching valves may cause a reduction of up to 60% in energy consumption compared to a traditional load sensing system [2].

However, there is a downside to this approach. Since the pressure in each chamber can only be the same as one of the supply pressures (due to the direct connection with the switching valves), the system can produce only a limited amount of force values. The amount of force states is \(N^M\), where \(N\) is the number of supply pressures and \(M\) the number of cylinder chambers. For a configuration consisting of a four-chamber cylinder and two supply pressures (as shown in figure 1) the maximum number of force states would be 16. For many applications where a high force resolution is required this resolution is not enough for an accurate control.
One possibility to overcome this limitation is to replace one of the switching valves with a proportional valve [3]. Figure 2 shows such a configuration for a four-chamber cylinder. In this system, the combination of states of the three switching valves corresponds to 8 different force levels. New in this approach is that all the force values between these discrete levels can be achieved by adjustment of the proportional valve, which enables the pressure in chamber 3 to adopt any arbitrary value in the range between the two supply pressure levels. As this system combines discrete force levels with a continuous force component, we call this system a semi-binary four-chamber cylinder.

Analysis has shown, that it is beneficial to operate the proportional valve on the A3 area, assuming $A_1 > A_2 > A_3 > A_4$. This allows for the best compromise between switching hysteresis and loss in energy efficiency.

Assuming that there is no pressure loss over the switching valves the theoretical efficiency for a given semi-binary multi-chamber cylinder (with an area ratio of $A_1/A_2/A_3/A_4 = 3.9/2.0/1.2/1$) connected to two different supply pressure levels (in this case being HP = 100 bar and LP = 20 bar) is shown in figure 3.
The diagram clearly shows in which area the proportional valve is active and where the connections to the cylinder chambers are switched. Also the efficiency is best when handling high forces (negative or positive).

2. SYSTEM DESCRIPTION

To prove the theory described in the introduction a demonstrator was set up in the laboratory in Lohr am Main. The system was used to move a slide block in the horizontal plain (see figure 5). Due to the complexity of a multi-chamber cylinder and the fact that the here presented study does not target to optimize the design of such a cylinder, we chose to substitute the multi-chamber cylinder by a setup consisting of three cylinders. As shown in figure 4, two identical cylinders were used to represent the chambers 1 and 2. The dove tail guide of the slide block guarantees that the displacement and the pressures of these cylinders are identical. A smaller third cylinder in between the other two represents the chambers 3 and 4.
The following chapters describe the chosen components and their main characteristics.

2.1. Valves

Initially three Rexroth M-3 SED 10 valves were used for the switching valves. These 3/2 way valves connected the chambers 1, 2 and 4 to the supply pressures and have a switching time of ca. 40 ms to open and ca. 25 ms to close. These switching times are pressure, flow, frequency and temperature depended and this has a noticeable influence on the quality of the control (as will be shown in chapter 3). It was therefore decided to replace two of the M-3 SED 10 valves with HPV Gen2 2/2 way switching valves. This is a new Rexroth valve that is currently available as an A-sample. The big advantage of this valve is that it opens and closes symmetrically in 3.5 ms in both flow directions. A picture of the valve and an example of a measurement of this valve is shown in figure 6. The measurement was taken with a pressure difference of 300 bar over the valve. The second benefit of the HPV Gen2 valve is that, considering the size of the valve, it only has a pressure drop of 10 bar at a flow of 150 l/min. This is achieved by a flow optimised hydraulic design and an adapted coil, that uses a current of 13 ampere during the boosting of the coil. This coil runs under oil to achieve the required cooling when operated at switching frequencies of up to 60 Hz.
As shown in figure 4 the final system setup consists of four fast switching valves HPV Gen2, two each for the chambers 2 and 4 and one M-3 SED 10 3/2 way valve for chamber 1. It was chosen to keep one “slower” switching valve, to see the influence of the switching time on the system performance. For chamber 3 the Rexroth 4WRDE 10 V 25 proportional valve was used, which needs about 15 ms to fully open. Performance details can be found in the data sheet RE 29093 [5].

2.2. Control

Goal of the control system is to allow both position/speed control and force control of the cylinder. Due to the design of the system, where three of the four chambers are either connected to high or low pressure, a cascaded controller structure was chosen with a force controller in an inner loop and a position and speed controller in the outer loop. The force controller sets the desired pressure levels in each chamber. This includes the command values for the five switching valves, which connect chambers 1, 2 and 4 to either the high or the low pressure supply and a desired continuous pressure for chamber 3, which is controlled with a separate pressure controller for this chamber. The simplified structure is shown in Figure 7. The pressure controller for chamber 3 is a proportional-controller with a disturbance observer and the flow-characteristics of the valve to consider the relation between pressure over the valve, spool position and desired flow. Additionally a feed forward is used to compensate for cylinder movement.

The controller was implemented in Simulink and ran on a dSPACE 1104 rapid prototype electronic in the Bosch Rexroth test laboratories in Lohr am Main, Germany.

![Figure 7. Simplified cascade control structure for the semi-binary four-chamber cylinder [4]](image)

2.2.1. Force Controller

The force controller decides how to switch the high performance switching valves and how to select the desired pressure for chamber 3. Statically this is an easy task to do. The three switching valves with two different pressure levels create 8 possible force steps. Taking into account that the pressure in chamber 3 can be given any value between high and low pressure, each step span a range of possible force values. Thus the controller selects an appropriate step, switches the valves accordingly and calculates the desired pressure in chamber 3 with

\[
p_{3,d} = \frac{F_d - (p_1A_1 - p_2A_2 - p_4A_4)}{A_3}.
\]
The relation between steps and desired cylinder force can be seen in Figure 8.

![Figure 8. Static selection of the valve state step as a function of the desired cylinder force [4]](image)

Dynamically the right selection of the step and the desired pressure in chamber 3 is much harder. This is due to the fact, that a change in a force step involves a transition of at least 2 chambers, in worst case even of all 4 chambers. A switching in chamber 1 affects about 75% of the maximum cylinder pushing force, resulting in major peaks in the sum force, if the counter acting pressure is not changed in an absolutely synchronized way.

Therefore a model-based valve synchronization was implemented, which predicts the switching delays for every valve action and chooses the optimal signal timing for the valve actuation. When using the new HPV Gen2 valves, the modelling turns out to be a simple task and synchronizing measures are very effective, since the switching times are very short, symmetric and reproducible for all of these valves. In the shown test configuration a time synchronisation is still necessary to compensate for the much slower dynamic of the chamber 1 and chamber 3 pressure controls.

2.2.2. Position Control

The position control is kept quite simple. It consists of a P-controller with an underlying PI-control-loop for velocity, thus realizing a typical control cascade. The controller output is a desired acceleration, which can be converted into a desired cylinder force by multiplying with the mass. The controller structure is shown in Figure 9.

![Figure 9. Simplified scheme of the position controller without integral part [4]](image)

The integral part of the velocity controller had to be added to achieve accuracy. To avoid controller windup a complex logic was designed that controls the activity of the integral controller part. For example during step transitions the system is not in closed loop control any more for a short period of time and therefore the integral part has to be stopped.
3. RESULTS

3.1. Force Control

The presented drive concept is based on an effective control of the pressure forces. Good accuracy in the static and dynamic performance of pressure and force control is prerequisite for an acceptable motion performance.

With the possibility of controlling one pressure in a continuous way (using a proportional valve and pressure sensors to derive actual pressure forces from every chamber), the static and dynamic behaviour is remarkable good while no switching action occurs, as can be seen in figure 10.

![Figure 10. Ramp response of the force controller.](image)

Obviously major errors in force occur during switching actions. Clearly the error amplitude depends on the number of valves that are switched at a time. As the total force is the sum of pressure forces, the following effects lead to errors:

- Different timing of valves.
- Different dynamics of valves.
- Dynamic effects in the “constant” pressure supplies exceeding the dynamic capabilities of the proportional valve.
A closer look to a combined switching action of all valves in figure 11 shows the resulting demands for 
switching valves and the advantages of the HPV Gen2 valves.

The dynamic force error results from the difference in switching dynamics between the HPV Gen2 valves 
(controlling chambers 2 and 4) and the M-3 SED 10 valve (controlling chamber 1). The pressure signals 
show the higher pressure gradients in chambers 2 and 4 and the reproducibility of the switching timing. 
Switching signals to the HPV Gen 2 valves are delayed by the control concept to synchronize with the 
chamber 1 valve. The synchronization is related to the crossing of the mid-pressure level between high and 
low pressure. Therefore pressure 1 increase starts before and ends after pressure 2 and 4. While the 
synchronization works well in the shown measurement, an extended measurement campaign has shown the 
complex switching behaviour of standard switching valves effecting even higher dynamic errors.

Figure 11. Force dynamics with all valves switched.

It is also clear, that with the necessity of a well working synchronization the slowest valve dominates the 
force control dynamics and, therefore, limits the performance of the motion control. Clearly, the HPV Gen 2 
valves meet the demands of the concept by ensuring:

- high and
- reproducible performance.
Figure 12 shows a measurement in which only chamber 2 and 4 are switched while chamber 1 is unchanged and chamber 3 is regulated. The pressure values in the chambers 2 and 4 show significant oscillations after the change-over. This shows the hindrances of the drive concept:

- Due to the high performance of the HPV Gen2 valves the limited dynamics of the proportional valve causes force errors and dominates the reachable system performance. To eliminate this effect, the performances of proportional and switching valves would have to be synchronized.
- Fast switching actions excite the fluid system and cause pressure oscillations in pipes and chambers, which cannot be compensated by the proportional valve. To limit these effects, multiple design efforts have to be made.

In total, to make the drive concept work in terms of an accurate force control several high level requirements have to be met. The most important are fulfilled by using HPV Gen2 valves.

The second main part of design concerns an effective reduction of hydraulic inductances (i.e. pipe/house lengths) and an effective accumulator concept to stabilize supply pressures.

### 3.2. Motion Control

Figure 13 shows the ramp response of the position controlled system when using HPV Gen2 valves (see “position” in figure 13) or standard switching valves (see “no HPV” in figure 13), respectively. As there is no switching action during this movement, the difference in the two measurements only results from the higher controller gains that are reachable due to the HPV Gen2 performance.
The additional effects of better valve switching synchronicity and the resulting reduction of force errors show in figure 14.

The difference between behaviour with and without HPV Gen2 valves is caused by the higher frequency of force errors. Using “normal” valves (“no HPV” in figure 14) causes errors due to timing errors. Using the HPV Gen2 valves generates errors due to excitation of high frequency oscillations, which have a smaller impact on the accuracy of integral values like velocity and position. These oscillations can be minimized by appropriate measures in the design of future multi-chamber cylinder drives with HPV Gen 2 valves.
4. CONCLUSIONS

The presented study clearly shows that the semi-binary approach for a four-chamber cylinder increases the force resolution and the position control performance compared to a digital set-up. When choosing the area ratio wisely the loss in efficiency compared to a digital four-chamber cylinder is also acceptable.

The used A-sample HPV Gen2 valves met expectations and proofed suitable for this concept. Measurements showed that the dynamic of the proportional valve was the limiting factor.

Furthermore it is very clear, that only an optimised design of the piping, cylinder ports and accumulator will lead to a system behaviour that fulfils the requirements of todays needs concerning force and position control in industry and mobile applications.
REFERENCES


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[5] Rexroth data sheet 2012, 4/3 directional high-response valves, pilot operated, with electrical position feedback and integrated electronics (OBE), RE29093
ABSTRACT

Current pumps and motors have already passed their expiration date. Especially the manufacturing costs and the energy efficiency don't meet up to today's market demands. New pumps and motors should already have replaced the current inadequate designs, if only the industry would have developed such alternative solutions. This paper not only calls for such an innovation, it also outlines the guidelines for efficient, heavy-duty hydrostatic principles:

- Avoid lateral loads in sliding interfaces;
- Avoid high bearing loads;
- Avoid piston rings;
- Avoid wide seal lands;
- Avoid high velocities in sliding interfaces;
- Avoid the risk for cavitation;
- Avoid large dead volumes;
- Avoid complicated tolerance chains and kinematic conflicts;
- Reduce the barrel spring force;
- Reduce the losses of the displacement control in variable displacement pumps and motors.

The company who succeeds in such an innovation has the opportunity to strengthen its own market position, but more importantly, will also be able to open entirely new markets for hydraulic applications.

KEYWORDS: Design rules, efficiency, heavy duty, pump, motor

1. THE IMPORTANCE OF EFFICIENCY

For many years, efficiency has not been a major topic in the hydraulic industry. Productivity was the prime requirement: hydraulic systems were favoured because they offered robustness in conditions that no other technology could endure. But this has changed. Without sacrificing the durability, reliability, power density and other productivity demands, the market today demands energy efficiency. Obviously, losses result in higher cost of ownership. Table 1 gives a simple example of a small application, in which an average power loss of only 20% results in an increased fuel consumption of several thousands litres of fuel per year. The annual cost of this extra consumption is often higher than the initial investment costs of the pump itself.
Reduce the barrel spring force;
Reduce the losses of the displacement control

5.1. Avoid lateral loads in sliding interfaces.

In slipper type pumps and motors, the full hydrostatic power is transferred via the sliding contacts between the pistons and the cylinders. In addition, the centrifugal forces often create an additional side load between the piston and the cylinder. These lateral forces result in high friction losses and wear. Large friction forces also increase the tipping torque of the cylinder barrel. In order to prevent the barrel from tipping, a relatively strong axial force is needed in the contact between the barrel and the valve plate. This further reduces the efficiency.

![Figure 3. Slipper type (swash plate) principle (left) and bent axis principle (right). The red arrows show the principle hydrostatic piston force. The blue arrows show the resulting lateral reaction forces.](image)

In other designs, the lateral loads in sliding interfaces are avoided or strongly reduced. In bent axis machines, the pistons directly drive the shaft. The slightly tilted position of the pistons and the piston rings still create some friction in the contact between the piston and the cylinder, but this is smaller than the friction caused by the slipper type principle. Other examples, in which lateral forces in the sliding contact areas are (in principle) avoided, are the Digital Displacement principle from Artemis IP [9], the RAC-principle from Dr. Berbuer [8], and the Floating Cup principle [10] from Innas.

5.2. Avoid high bearing loads

The friction of rolling bearings is almost negligible if the bearings are small and only have to carry a minor load. The bent axis design is a clear example where this guideline has not been followed. The complete hydrostatic force of all pistons is acting on the main bearing, thereby creating a strong axial and radial bearing load. This again results in large bearings having a large diameter. The heavy load combined with the large bearing diameter results in relatively high losses and cooling issues [20].

The slipper type machine avoids strong axial loads on the roller bearings of the main drive shaft. The hydrostatic force from each piston is counteracted by the slipper and by the contact between the piston and the cylinder (thereby creating another friction loss; see the previous paragraph). The balancing principle is also applied in double, mirrored construction of the Floating Cup principle.
5.3. Avoid piston rings

In many piston machines, the piston has a line contact with the cylinder. In these machines, piston rings are generally used as a sealing element.

Piston rings are needed as a flexible sealing element, if the deformation of the cylinders is not rotational symmetric. Figure 5 shows, as an example, the result of a FEM analysis of the barrel of a 45 cc bent axis pump at a pressure level of 350 bar. The four plots show the (magnified) radial deformation of the cylinder at four different positions of the pistons (i.e. the sealing line of the piston rings). The analysis indicates that, in such a construction, the deformation is not rotational symmetrical. This is for a part due to the construction of the barrel itself. But also the pressurized neighbouring cylinders have a strong effect on the deformation. Furthermore, the deformation varies, depending on the piston position. In these constructions, only piston rings can be used to follow the unpredictable and variable deformation of the cylinders.

Figure 5. FEA of the cylinder barrel of a 45cc bent axis pump at 350 bar. The figures show the radial deformation at the sealing line of the piston relative to the original perfect circles, of 4 cylinders at different piston positions x (defined relative to the bottom dead centre).

Piston rings do not have a complete hydrostatic balance. The unbalance helps to pressurise the piston ring against the cylinder wall, thereby closing the gap between the circumference of the ring and the cylinder. But
the unbalance also results in friction losses. In addition, the expansion of the piston ring results in a gap of
the piston ring slot being opened and leakage occurs [21]. The leakage often increases due to wear.

Piston rings can be avoided if both the piston and the cylinder have the same predictable expansion due to
pressure loads and thermal expansion. In order to match the expansion of the piston and the cylinder, the
piston head must have a cavity. The pressure forces can then expand the piston head to match the
expansion of the cylinder. Since the expansion of the piston will be equal in all, radial directions, the cylinder
must have a rotational symmetrical expansion as well. Because of this, the cylinders must be isolated from
the barrel: each piston must have its own cylindrically shaped cylinder, completely isolated from the pressure
load of the neighbouring cylinders. This concept is applied in the Floating Cup principle.

5.4. Avoid wide seal lands

Due to the uncertainties related to the pressure profile in the sealing gaps, it is advantageous to reduce the
sealing areas to a minimum. Figure 6 shows two situations, one with wide seal lands, and one with narrow
seal lands. In both cases a hydrostatic balanced construction can be calculated assuming a linear pressure
drop in the gap. For the construction with wide seal lands, the pressure in the sealing areas has a much
larger contribution than the sealing (and pressure balancing) areas with narrow seal lands. Due to the factors
mentioned before in paragraph 2, the pressure profile in the gap will not be linear. Any deviation from the
assumed profile will cause a mismatch. This is true for both situations show in Figure 5. The consequences
are however much smaller for the construction having narrow seal lands.

![Figure 6. Pressure profiles for two hydrostatically balanced constructions, one with wide seal lands (left) and one with narrow seal lands (right).](image)

It could be argued, that a wide seal land offers a better sealing function and, thus, reduces the volumetric
losses. But the gap height has a much stronger influence on the leakage than the width of the seal land, and
the leakage will predominantly be determined by the part of the seal land, which has the smallest gap height.

5.5. Avoid high velocities in sliding interfaces

The viscous friction represents another group of energy dissipation. Viscous friction losses are related to the
size of the contact areas, the (non-uniform) gap heights, the oil viscosity and the sliding velocities. Slipper
type machines are characterised by large contact areas, being operated at high sliding velocities: between
the pistons and the cylinders, between the barrel and the port plate and between the slippers and the swash
plate. Due to the relatively large radius of the piston pitch circle of these machines, the sliding velocities are
often high.
The Floating Cup principle has a similar disadvantage. Due to the large number of pistons and the small tilt angle of the barrels, the pistons in the Floating Cup principle run on a relatively large pitch circle. As a result, the barrel ports are also running at a relatively high velocity. The situation is even worsened by the double, mirrored design, in which two barrels are running on two port plates. A solution has been found in the application of a special design of the seal lands of the barrels [22]. In addition, the Floating Cup principle hardly has any losses between the pistons and the (cuplike) cylinders.

The viscous losses are strongly related to the gap height: a narrow gap results in high viscous losses. A large gap height, on the other hand, results in high leakage. In reality, gaps do not have a constant gap height. Due to deformation and tilting, the gap heights vary strongly across the sealing areas. In most cases there will be spots where the gap height is very small, and high viscous losses occur, whereas on other places of the same gap, the gap height is relatively large, having a high leakage as a consequence. It is the task of the designer to find a bearing and sealing construction, which creates a more or less constant gap height, and which offers a good resistance against torque and force loads acting on the moving parts.

5.6. Avoid the risk for cavitation

The flow resistance of the low-pressure lines, up to the cylinders of the rotating barrel, causes a throttle loss, which directly influences the efficiency. But the flow resistance also increases the risk for cavitation. In order to avoid cavitation, it could be necessary to pre-charge the supply side. The charge pump will then create additional losses, especially since its (constant) displacement cannot be matched with the variable displacement of the main pump.

To avoid or to at least reduce the need for pre-charging, the resistance of the supply lines needs to be reduced to a minimum. Aside from having large flow areas (which is not possible if the supply side is not predefined), it is also advantageous to increase the number of pistons and reduce the piston stroke. A short piston stroke increases the piston area and often also the port opening area of the cylinder barrel. The short stroke also reduces the piston acceleration, and thus the required pressure for accelerating the oil column to
feed the cylinder during the suction stroke. The application of more pistons furthermore increases the total opening area of the barrel ports.

A bent axis pump performs worst in this respect. The number of pistons is smaller, and, due to the large swash angle, the stroke length is rather large compared to the piston diameter. The floating cup principle is on the other end of the design spectrum. It features a large number of pistons, which results in a large opening area of the barrel ports (see Figure 7).

5.7. Avoid large dead volumes

In positive displacement machines, the displacement chambers need to switch between the high and low-pressure lines. Every time, this switching or commutation occurs, the oil volume in the displacement chamber needs to be compressed and expanded, in an alternating way. In theory, the oil volume can be pressurised and depressurised without any losses, by using the movement of a piston (creating a certain volume change \( \Delta V \)) to create the required pressure change (\( \Delta p \)). In reality, the bulk modulus of the oil varies too much and is too uncertain to allow for such a construction. Moreover, the required volume change (\( \Delta V \)) varies with the operating conditions, i.e. the operating speed, pressure, displacement and even temperature.

In most hydrostatic pumps and motors, pressure relief or silencing grooves are used to soften the commutation. The throttle losses, which are a result of this commutation, are dependent on the volume of the displacement chamber during commutation. Part of this volume (the dead volume) does not contribute to the actual displacement, but nevertheless needs to be compressed and expanded as well for each commutation event. Depending on the design of the hydrostatic machine, this dead volume can be large or small. Large dead volumes, however, result in significant losses.

Slipper type pumps often have large dead volumes (Figure 7). The pistons have a large axial bore, which helps to reduce the centrifugal forces of the pistons. This results in lower friction forces between the pistons and the cylinders, and in a reduction of the tipping torque load on the cylinder barrel (paragraph 5.9), but it also increases the dead volume, and the related compression and expansion losses.

![Figure 8. Dead volume in a slipper type pump](image-url)
5.8. Avoid complicated tolerance chains and kinematic conflicts

Complicated tolerance chains and narrow tolerances not only increase the manufacturing cost; they also increase the risk for kinematic conflicts due to deformation of the components because of pressure loads and thermal expansion. Once a kinematic conflict occurs, it results in a strong increase of both friction and wear. Wear can further deteriorate the gap profile of sliding contacts, thereby destroying the hydrostatic balance, and thus, further increasing wear and friction. Friction will also cause local heat generation and thermal deformation, which can additionally worsen the situation.

It is strongly recommended to avoid constructions were the components are interlocked and cannot position themselves. A clear example of a problematic design is the gerotor principle in which a multi-lobbed inner gear is positioned inside an outer gear. On each tooth pair, a sealing line needs to be established, which makes the construction undefined. Better design examples are the axial piston machines, in which the barrel is free to find its position on the valve plate.

5.9. Reduce the barrel spring force

In axial piston designs, the barrel spring needs to be strong enough to avoid tipping of the barrel due to the combined action of centrifugal forces and piston friction. The barrel spring however increases the friction between the barrel and the port plate, and it is therefore important to reduce this force by means of reducing the piston friction (see paragraph 5.1), and by reducing the centrifugal forces of the pistons. Bent axis motors are better fitted for high rotational speeds than slipper type motors, largely due to the lower centrifugal forces of the pistons acting on the barrel. In the floating cup principle, the pistons are press-fitted into the rotor and are therefore unable to create a tipping torque. Instead, the centrifugal forces of cups will create a torque load on the barrel. However, this effect is rather small, due to the low weight of the cups and the short 'cup' stroke.

5.10. Reduce the losses of the displacement control

For variable displacement pumps and motors it is imperative to reduce the losses off the system, which varies and controls the effective displacement. There is hardly any research being published about this topic. Nevertheless, there are sufficient indications that pump and motor control systems are the cause of significant losses, often more than all the previous losses together. More research in this area is urgently needed.

6. THE DEMANDING MARKET

The market favours improvements. It is willing to buy a new technology, if the new technology offers a crucial advancement. The efficiency is certainly such a key performance parameter. But the market is demanding: it wants improvements without sacrificing any of the other advantages.

For the designer, this means that the new principle needs to have an improved efficiency, without sacrificing the durability, the reliability, the power density, the noise and pulsations levels, the dynamic control performance, manufacturing costs, or any of the other characteristics.

It is tempting to incrementally improve the design of current hydrostatic principles, since this reduces the development risks. But this has been tried in the past decades. A large effort has been spent on new materials and coatings, on micro finishing of pistons and piston skirts, and on reducing the churning losses, with only incremental progress. It is fair to conclude that the current axial and radial piston principles have come a maturity level, which cannot be improved any further, and that new principles need to be designed and developed.
REFERENCES


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Table 1. Example of the effect of pump losses on the fuel consumption

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average pump input power and efficiency:</td>
<td>50 kW at 80% average efficiency</td>
</tr>
<tr>
<td>Average loss:</td>
<td>20% i.e. 10 kW</td>
</tr>
<tr>
<td>Annual operation time:</td>
<td>1000 hours/year</td>
</tr>
<tr>
<td>Annual pump losses:</td>
<td>10 000 kWh/year</td>
</tr>
<tr>
<td>Average engine efficiency:</td>
<td>33%</td>
</tr>
<tr>
<td>Fuel consumption related to pump losses:</td>
<td>30 000 kWh/year = 108 GJ/year</td>
</tr>
<tr>
<td>Diesel fuel:</td>
<td>28 liter/GJ</td>
</tr>
<tr>
<td>Fuel consumption related to pump losses:</td>
<td>3024 liter/ year</td>
</tr>
</tbody>
</table>

In addition, losses increase the oil temperature and therefore the cost and the size of coolers. Friction losses are also often related to wear: severe friction in sliding interfaces generally reduces the durability and reliability. Furthermore, friction losses limit the operation at low operational speeds. To overcome the torque losses during start-up, the displacement of the motor needs to be increased, which results in strong part-load operation at normal operating speeds. Stick-slip also results in a strong non-linear behaviour of pumps when being operated at close to zero operating speeds, thereby resulting in restrained controllability. Volumetric losses, on the other hand, result in creeping movements with hanging loads, which is often undesirable. The reduction of losses in hydrostatic machines is therefore, without doubt, one of the most important design criteria for modern machines.

This paper describes a number of guidelines for the design of efficient hydrostatic pumps and motors. The paper is limited to heavy-duty machines, which can be operated at high-pressure levels of 350 bar and above.

Before discussing these guidelines, it should be noted that current definitions and standards for efficiency are confusing and often even misleading. In the literature, the efficiency is mostly split into hydraulic-mechanical and volumetric efficiencies [1]. But these definitions do not give a correct representation of the real volumetric losses (i.e. leakage) and the real friction losses, which makes it difficult to get a proper understanding of the losses. It is, for instance, nearly impossible to decide whether the losses, which occur in the silencing grooves during commutation, should be referred to as volumetric losses or flow resistance losses i.e. friction related losses. In this paper, these definitions are therefore not used.

Furthermore, the current standards for efficiency measurements of hydrostatic machines, like the ISO4409 [2], do not take the losses of the control system of variable displacement machines into account. There is almost no literature about the effect of the control system on the overall pump or motor efficiency. This is most curious, since variable displacement pumps and motors always need to be controlled in some way or the other, and the losses related to the control valves, control pistons and others should be included in the efficiency measurements in order to get a correct understanding of the actual losses of variable displacement pumps and motors in real applications.

2. EVERYTHING CHANGES AT HIGH PRESSURES

According to Dias [4], only 2% of the most common sources of hydraulic pump failures are related to the pump design itself. Most failures (80%) are due to mistakes in operation and maintenance (Table 2).
Table 2. Most common sources of hydraulic pump failures [4]

<table>
<thead>
<tr>
<th>Source</th>
<th>Failure frequency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design</td>
<td>2%</td>
</tr>
<tr>
<td>Manufacture</td>
<td>6%</td>
</tr>
<tr>
<td>Installation</td>
<td>12%</td>
</tr>
<tr>
<td>Operation and maintenance</td>
<td>80%</td>
</tr>
</tbody>
</table>

However, for heavy-duty machines, which operate at high pressure levels, the risk for design failures is much higher:

- Higher pressure loads result in stronger deformations of components, both overall and local;
- The throttle losses are higher at high pressures, which influences the oil temperature, and thus the viscosity of oil in sealing and bearing gaps;
- The viscosity is pressure dependent, which has a strong effect on the pressure profile in the sealing and bearing gaps;
- These previous three effects make it more difficult to achieve a correct hydrostatic balance of the sliding interfaces. Any unbalance will result in higher contact forces, due to the higher pressure levels, and thus in increased friction and wear;
- Smaller gap heights are required in order to maintain a low leakage at high pressures. This makes the construction more vulnerable to abrasion due to debris carried by the oil flow;
- The amplitude and variation of the load acting on the roller bearings and the swash plate bearings of variable displacement pumps is larger, which increases the risk for spalling [3], brinelling (impact deformation) and scuffing or galling [4];
- The load on the drive shaft creates a deformation, which can increase the inclination of the inner rings of the roller bearings. This reduces the lifetime of the bearings.

At high pressure levels, many design aspects that could be neglected before, suddenly become key parameters. The first parameter to consider is deformation. All hydrostatic principles have sliding contact areas, which can only function if the construction is hydrostatically or hydrodynamically compensated. The oil film in between the sealing and bearing areas, has a gap height which is mostly below 10 μm. Any deformation of the components facing these oil films, results in a change of the pressure profile in the gaps, and thus in the bearing and sealing capacity of the contact area. Even without any general bending or distortion of the components, the local compression of the material itself will already cause a different gap profile. As an example, a steel component, having a bulk modulus of 160 GPa, compresses 0.03% at a pressure of 500 bar. This may not seem much, but for a component with a thickness of 1 cm, a compression of 0.03% equals a deformation of 3 μm. This is in the same order of magnitude as the gap height itself. In most pump designs, the deformation will even be larger, due to general distortion of the components. Figure 1 shows, as an example, the axial deformation of a valve plate at a pressure load of 350 bar [5]. The figure shows the combined deformation of the valve plate and the underlying housing, which carries and supports the valve plate. Combined with the additional deformation of the cylinder barrel, the gap height variation due to deformation can amount to 12 μm at 350 bar.
Another key parameter for designing heavy-duty pumps and motors is the dynamic viscosity of the oil, which can change to a large extent while passing the sealing gap. The viscosity is both temperature and pressure dependent. At 500 bar, the dynamic oil viscosity is 2 to 3 times as viscous, compared to oil at atmospheric conditions. In addition, the oil heats up while passing the gap due to throttling effects (the gap acts as a resistance). In case of a relative movement between the contact areas, for instance between the rotating barrel of an axial piston pump and the stationary valve plate, the viscous and (potential) coulomb friction will further heat up the oil in the gap, thereby reducing viscosity of the oil even more while the oil passes the gap. A pressure drop of 500 bar, combined with a temperature increase of the oil in the gap of 20°C, result in a viscosity increase by a factor of five during the passage of the oil through the gap [6].

The change of viscosity has a strong effect on the pressure profile in the gap, and therefore also on the bearing and sealing capacity of the sliding contact areas in hydrostatic pumps. All these effects are pressure dependent and become more important (and difficult to handle) at high oil pressures. In addition there is the thermal expansion, which can especially be large in areas where poor lubrication conditions cause more friction, and therefore create a (local) temperature increase [7].

The complex relationship between the design parameters, the operating conditions, and the elastohydrodynamic (EHD) lubrication is still not completely understood. A pump has to work in a large range of operating conditions (pressures, rotational speeds, temperatures, swash angles): a specific design needs to fulfill all demands in all of the relevant and required operating pressures, speeds, displacements and temperatures. If, for instance also at low operating speeds, high torque losses due to friction need to be avoided, then it is no longer possible to rely on hydrodynamic lubrication. In that case, a hydrostatic solution needs to be found or the lateral forces need to be strongly reduced. The designer has the task to find solutions, which functions at all required operating conditions. Furthermore, the solution needs to be robust, despite inevitable production tolerances and component wear.

3. CONTROL OF VARIABLE DISPLACEMENT MACHINES

Variable displacement, heavy-duty pumps and motors are, without exception, axial piston or radial piston designs. The displacement is varied by means of changing the eccentricity in radial piston machines, or the swash angle in axial piston designs. An exception is the Digital Displacement principle [9] in which the displacement is changed and controlled by means of solenoid-activated valves for the commutation.
In most variable displacement pumps and motors, the displacement is controlled by means of a hydraulic system, often consisting of an actuator piston and a valve system to set and control the pressure in the actuator cylinder. There are many types of control valves, but, in a simplified representation, the control can be regarded as a simple pressure divider (see Figure 2) in which the resistance $R_1$ represents the control valve itself.

![Figure 2. Control of a variable displacement slipper-type pump [11]](image)

Due to the limited numbers of pistons, the pistons create a varying torque load on the swash plate, which results in an oscillation of the swash plate [11], [12]. The oscillation is counteracted by the pressure variation in the control piston, i.e. by the damping action of resistances $R_1$ and $R_2$. The dissipation in the resistances, and the leakage flow of the pressure divider, both have a detrimental effect on the efficiency. Other optional control valves, like the flow control, create substantial, additional losses.

These losses have a significant effect on the total efficiency. The leakage of the control system can be several litres per minute [13]. Bosch Rexroth mentions a fluid consumption of up to 4.5 l/min, also for relatively small variable displacement pumps [14]. Especially at low rotational speeds and small displacements, the influence of the control system on the efficiency can be dominant.

More research on the effects of the control system is urgently needed. For the designer, it is the task to design a control system, which avoids or, at least, minimizes the losses of the control system.

4. THE INFLUENCE OF THE ROTATIONAL SPEED

Many losses in hydrostatic machines are related to the rotational speed:

- At higher speeds, the risk for cavitation increases. A charge pump can be used and mounted to the main pump to overcome cavitation, but the charge pump will reduce the overall efficiency of the combination of main pump and charge pump. The charge pump is also used in closed circuits for compensating for the fluid compressibility, for replenishing hydraulic fluid that is lost due to leakage and for auxiliary functions and cooling [13]. Charge pumps are constant displacement pumps, which are dimensioned for worst cases, especially for maximum cooling demands. A low pump and motor
efficiency requires a larger cooling system (having more flow resistance) and a larger charge pump. This will further deteriorate the efficiency of the total system, but the basis is the often, poor efficiency of the pumps and motors.

- Higher rotational speeds result in increased flow losses, churning losses and flow impulse losses.
- High rotational speeds also increase the risk for tipping of the cylinder barrel of axial piston pumps and motors. In axial piston machines, the centrifugal forces of the pistons create a tipping torque on the cylinder barrel [15] (Manring). The friction between the pistons and the cylinders further increases the tipping torque. To counteract the total tipping torque, a spring is mounted to prevent the barrel from tipping. The spring force creates an additional load on the sliding contact area between the cylinder barrel and the valve plate. This additional load results in extra friction. Once a certain spring is chosen, the spring force and the related friction are always there, also at low operating speeds and low pressure levels, when a high spring force is not needed.

All of these factors are important at high rotational speeds. But efficiency is also a concern at low operating speeds. Hydraulic motors often suffer from stick-slip effects at low rotational speeds when mixed-lubrication and coulomb friction become dominant. As a result, the torque capacity is reduced at start-up and breakaway conditions, sometimes as much as 40% [16]. In addition, the torque is further reduced as a result of the limited number of pistons, which causes a variation of the drive torque. The reduction of the torque forces designers of hydraulic transmissions to compensate for these losses by oversizing the motors. As a consequence, the motors are often driven in part load conditions, and therefore reduced efficiency, as soon as the stick-slip effects have overcome.

Stick-slip is also detrimental for electro-hydraulic actuators, in which a speed-controlled electric motor is driving a constant displacement pump. The Strubeck-effects cause a strong non-linear behaviour of the control of these actuators [17]. Furthermore, these units are often driven at low rotational speeds. The increased friction and the often relatively high leakage losses strongly reduce the operating efficiency at these conditions.

Leakage is also a concern for holding a load at near zero speed conditions or when a complete standstill is required. Volumetric losses in hydraulic motors will then result in creeping movement, which is extremely undesirable in many applications.

5. WHAT TO DO?

Hydrostatic machines are not restricted in their efficiency by thermodynamic limitations, unlike for instance thermal engines, for which the efficiency is restrained by Carnot’s theorem. In principle, the efficiency can approach 100%. Hydrostatic machines are comparable to gear transmissions, in which one form of mechanical power is transformed to another (hydraulic power can be considered as a form of mechanical power). Yet, the peak efficiency of hydraulic pumps and motors is around 90% for most axial piston pumps and motors [18]. This is the ‘overall’ efficiency in the best point, without the losses of the control system. In reality, the average real world efficiency is much lower.

This paragraph describes a number of design guidelines for the design of efficient hydrostatic machines. The guidelines are meant for designers who need to develop a new (heavy duty) hydrostatic principle:

- Avoid lateral loads in sliding interfaces;
- Avoid high bearing loads;
- Avoid piston rings;
- Avoid wide seal lands;
- Avoid high velocities in sliding interfaces;
- Avoid the risk for cavitation;
- Avoid large dead volumes;
- Avoid complicated tolerance chains and kinematic conflicts;
LOAD INDEPENDENT VELOCITY CONTROL ON BOOM MOTION USING PRESSURE CONTROL VALVE

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ABSTRACT

This paper presents a novel scheme for closed loop velocity control of a pressure control valve with attention to the load dependency of both the dead band and the metering-in flow. The control strategy is designed with the use of a minimum of sensors. The only required sensors are position sensors on the cylinders. The performance of the proposed controller is being evaluated both with and without the additional use of pressure transducers. The control scheme is implemented experimentally on a cylinder actuated knuckle boom of a commercial vehicle loader crane.

KEYWORDS: Pressure control valve, velocity control, dead band compensation, valve mapping, mobile loader crane, feed forward control

1. INTRODUCTION

It is well known that directional control valves with pressure compensated metering-in flow in series with counterbalance valves may introduce instability in a system, see [1-3]. This is especially pronounced when the controlled actuator is subjected to a negative load, i.e., a load that tends to drive the actuator as a pump, because this will require the counterbalance valve to throttle the return flow. The required throttling of the return flow may be handled by the return orifice of the directional control valve as described in [4] thereby eliminating the oscillations. However, this is not a viable solution if the minimum load is 60% or less of the maximum load. In that case, the use of so-called pressure controlled metering-in flow is an alternative that may be put in series with a throttling counterbalance valve without any oscillations. This represents a cost effective alternative to high bandwidth servo valves with the main drawback being the lack of load independent flow, see also [5]. For systems with predominantly positive load on the cylinder during extraction and predominantly negative load during retraction it is possible to combine, the pressure controlled metering in during retraction with pressure compensated metering-in during extraction. That way the flow control offered by the compensator is kept when extracting, however there is a lack of load independent flow, hence velocity of the cylinder, during retraction. In that case, also the dead band of the valve is load dependent. This paper investigates these drawbacks experimentally and the use of position and pressure feedback are introduced with a view to copy the flow control performance of pressure compensated valves except for the oscillatory nature.
2. CONSIDERED SYSTEM

The mobile valve group PVG32 manufactured by Danfoss A/S, Figure 1, is available with pressure controlled metering-in. The controlled pressure is the pressure, \( p_c \), between the compensator and the main spool.

![Figure 1](image)

**Figure 1. (a) PVG control section with pressure compensated metering-in PA and pressure controlled metering-in PB, (b) Variables associated with pressure controlled metering-in.**

The variation in behavior is embedded in the main spool, i.e., the choice of main spool for the 4/3-way directional control valve determines the combination of flow and/or pressure control. A simplified analysis of the PB metering-in flow, that assumes fully turbulent flow characteristics and disregards any saturation phenomena and variation in spring compression, yields:

\[
Q_{PB} = C_d \cdot A_d(x) \cdot \sqrt{\frac{2}{\rho}} \cdot (p_c(x) - p_B) \tag{1}
\]

Where \( x \) is the main spool travel, \( C_d \) is the discharge coefficient of the orifice and \( A_d(x) \) is the variable discharge area. Steady state equilibrium of the compensator and flow continuity in the bleed off line can be formulated as:

\[
p_c = p_B + p_{ct} \tag{2}
\]

The spool position dependency of the inlet pressure can be derived from the area ratio, \( \mu(x) \), between the fixed orifice, \( A_0 \), and the variable bleed off orifice, \( A_{bo}(x) \), see Figure 1 (b).

\[
\frac{\Delta p_{A0}}{\Delta p_{Ab0}} = \frac{p_c - p_D}{p_D} = \left[ \frac{A_{bo}(x)}{A_0} \right]^2 \equiv \mu(x)^2 \tag{3}
\]

By combining equation (2) and (3):

\[
p_c(x) = \left[ \frac{1 + \mu(x)^2}{\mu(x)^2} \right] \cdot p_{ct} \equiv \beta(x) \cdot p_{ct} \tag{4}
\]

where \( \beta(x) \) is the characteristic of the pilot part of the valve. Hence, the bleed off orifice area \( A_{bo}(x) \) should close as the main spool opens, however, to avoid saturation it should not close fully. By careful design the pressure control of a PVG32 pressure controlled metering-in is linear after a certain dead band as deduced from the catalogue [6]:

\[
p_c(x) = \begin{cases} 
0 & 0 \leq x [mm] \leq 0.8 \\
300[bar] \cdot \frac{x[mm] - 0.8}{4.7} & 0.8 \leq x [mm] \leq 5.5 \\
300[bar] & 5.5 \leq x [mm] \leq 7
\end{cases} \tag{5}
\]
The performance of the pressure controlled metering-in valve has been obtained experimentally using the setup shown in Figure 2 (a). The load pressure, $p_B$, was varied by means of the pilot pressure to the counterbalance valve. In Figure 3 (left) are shown three 2-dimensional traces of the flow vs. load pressure and main spool travel. The curves display the volume flow vs. spool travel for fixed values of the load pressure.

![Diagram of experimental setup and hydraulic circuit](image)

**Figure 2.**(a) Experimental setup for mapping of pressure control spool. (b) Hydraulic circuit of jib boom.

**Figure 3.**(left) Mapping of pressure control spool. (right) Vehicle loader crane.
A control section with metering-in pressure control was mounted on a commercial mobile loader crane that, basically, is a four-degree of freedom machine consisting of a slewing motion, a main boom, a jib boom and a telescopic arm system, see Figure 3 (right).

The pressure controlled metering-in was applied to the cylinder controlled rotation of the jib boom relative to the main boom. Some important data on the jib cylinder are listed in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum cylinder length</td>
<td>$L_{AB,\text{min}} = 1160\text{mm}$</td>
</tr>
<tr>
<td>Maximum cylinder length</td>
<td>$L_{AB,\text{max}} = 1980\text{mm}$</td>
</tr>
<tr>
<td>Cylinder stroke</td>
<td>$h_c = L_{AB,\text{max}} - L_{AB,\text{min}} = 820\text{mm}$</td>
</tr>
<tr>
<td>Cylinder piston diameter</td>
<td>$D_p = 150\text{mm}$</td>
</tr>
<tr>
<td>Cylinder rod diameter</td>
<td>$D_r = 100\text{mm}$</td>
</tr>
<tr>
<td>Cylinder area ratio</td>
<td>$\varphi = \frac{D_p^2}{D_p^2 - D_r^2} = 1.8$</td>
</tr>
</tbody>
</table>

The hydraulic circuit of the jib boom motion is shown in Figure 2 (b).

The main spool is electro-hydraulically actuated and an inner closed loop ensures that spool position is proportional to the input signal.

### 3. CONTROL STRATEGY

The proposed control strategy is shown in Figure 4. The pressure control valve introduces two challenges related to motion control: load dependent dead band and load dependent flow.

In the following, the dead band compensation (DC) strategy and the velocity control (VC) are presented.

#### 3.1. A – Dead band compensation

The idea of dead band compensation is to minimize the time from an input have been registered until the system starts moving. Ideally, this is achieved by moving the main spool to the position just before it would make the system move.
Figure 5. Dead band compensation controller realization. (a) Shows the variable dead band compensation (VDC). (b) Shows the much simpler constant dead band compensation (CDC).

In this paper three types of dead band compensation are evaluated:

- **Variable dead band compensation (VDC):** VDC is the proposed strategy for a load-independent dead band compensation, see (a) in Figure 5. It consists of a 3-step control strategy.
  1. A feed forward branch with a constant value, \( DB_{\text{Deenergizer}} \). The effect of the deenergizing part is to remove the backpressure \( p_k \), before increasing the \( p_B \) pressure. The deenergizer constant, \( DB_{\text{Deenergizer}} \) has been found experimentally to a spool position of \(-1.50\)mm. This step is given 100ms to move the main spool before activation of the second step.
  2. Step 2 involves a second feedforward branch. The size is determined by sending the calculated reference pressure, \( p_{B,\text{ref}} \), (see formula below) through a valve mapping to determine the corresponding command signal.
  3. The last step consists of a PI-controller working on the difference in pressure \( p_B \) between the measured value and the calculated reference. The PI-controller is being activated when the error in pressure is below 5bar, giving the feed forward time to move the spool.

**Requirement:** Pressure transducer \( p_B \).

- **Constant dead band compensation (CDC):** The simplest form of dead band compensation, which requires no sensors or knowledge in depth of the system. The controller consists of a feed forward branch moving the spool to the fixed dead band, \( DB_{\text{valve,PC}} \), described in the datasheet in reference [6], or see part (b) in Figure 5. It has the value for the spool position of \(-0.80\)mm, corresponding to a command signal of \(-11.42\%\). However, since it does not utilize any sensors, it will not address the issue of the load-dependent dead band. It has been included as a reference for comparison with the VDC.

**Requirement:** No sensors

- **No dead band compensation (NDC):** The spool is in rest.

**Requirement:** No sensors

The required pressure for opening the counter balance valve, \( p_{B,\text{open}} \), can be expressed as:

\[
p_{B,\text{open}} = \frac{p_{cr} - p_1 + (\alpha + 1) \cdot p_A}{\alpha}
\]

The crack pressure, \( p_{cr} \), is set at 350bar and the pilot ratio, \( \alpha \), is 4.25. Due to the use of a 3-port counter balance valve, the backpressure helps closing the counterbalance valve. It is not possible to predict at which pressure \( p_B \) it opens, without adding another pressure transducer measuring \( p_A \). However, by adding a deenergizing delay to the dead band compensation the backpressure \( p_A \) is drained to tank, hence being negligible. This reduces the above expression to:

\[
p_{B,\text{ref}} = \frac{p_{cr} - p_1}{\alpha} \leq p_{B,\text{open}}
\]

where \( p_{B,\text{ref}} \) is the reference pressure, at which the dead band compensation settles.
3.2. B – Velocity control

Compared to the previously developed control strategy in reference [5] the controller has been expanded to make full use of the valve mapping and utilize a velocity reference as input (joystick). This way the operator closes the control loop.

The proposed controller consists of a position feedback P-controller. The goal is to control the velocity. However, the velocity feedback term derived from the position sensor of the experiments is too noisy to be used in closed loop control. The problem is avoided by generating a position reference, $s_{\text{ref}}$, from integration of the reference velocity obtained from the joystick.

Three versions of the velocity controller were designed, as shown in Figure 6. Each step adds functionality, but increases the complexity at the same time:

- **Control strategy no mapping (NM):** The simplest controller consists only of the position controller. The advantage is that it is very simple to implement, hence it does not require any knowledge of the system in advance (no mapping). The cost is expected to be a slower system and lower performance.
  
  **Requirement:** Position transducer.

- **Control strategy constant mapping (CM):** CM is a further investigation of the previous presented controller in reference [5]. Adding a feedforward branch with a mapped valve characteristic is expected to increase the performance level. However, it is more troublesome to implement than NM, since the valve must be mapped separately. It still only utilizes the position sensor, why an investigation of the sensitivity for a different load situation is of interest.
  
  **Requirement:** Position transducer.

- **Control strategy variable mapping (VM):** A pressure transducer, $p_B$, is added to make the controller able to adjust to the current load. Performance wise this is expected to result in the best performance.
  
  **Requirement:** Position transducer and pressure transducer to measure $p_B$. 

---

**Figure 6. Velocity controller realization, (a) Controller with use of pressure $p_B$ in the feed forward mapping (VM) (b) Controllers without use of pressure $p_B$. The blue + red part still utilize the valve mapping in the feed forward, but with $p_B$ constant (CM), whereas the blue part describes a controller without the mapped feed forward branch (NM).**
4. EXPERIMENTAL WORK

The experimental work has been divided into two parts:

- A – Dead band compensation
- B – Velocity control

The proposed control strategies have been implemented on the mobile loader crane and have been evaluated.

4.1. Evaluation

The two different parts of the experimental work is evaluated differently.

- A – Dead band compensation: The focus in this part is to measure the time consumption for:
  1. Settling time, which describes the time from the DC is activated until the spool have settled.
     - Activate the controller and measure.
  2. Dead time, is defined as the time delay between a command signal is given and movement of the boom can be registered.
     - With DC activated and a settled spool, a step input of \( u_{joy} = -60\% \) is applied.
  3. The ability to adapt to changes of the load situation is being tested by increasing the experienced load by extending the telescope from the standard 0m used in all other experiments to 2m and compare the settling time.

- B – Velocity control: Here the focus is to compare the overall performance of the VC and test the robustness of it.
  - A velocity ramp, see Figure 7, is used as input. The ramp time = 10s is kept constant. The ramp is defined by the rise time, \( t_r = 0.1s, 0.5s, 1s, 2s \), and the desired travel distance, \( \text{dist} = 0.1m, 0.2m, 0.3m \).

![Figure 7. Velocity ramp used in the experimental work (left). At the right is shown the corresponding position reference.](image_url)

The root-mean-square deviation (RMSD) is being used as criterion for the relative evaluation of the two data sets, reference and measured. It is defined as:

\[
\text{RMSD}_x = \sqrt{\frac{\sum_{t=1}^{n}(x_{reference,t} - x_{measured,t})^2}{n}}
\]  

(8)

The lower the value of \( \text{RMSD}_x \) the better correspondence between the reference and what actually happens.

In the later, the velocity error is defined as: \( \dot{s}_{error} = \dot{s}_{reference} - \dot{s}_{measured} \).
4.2. A – Dead band compensation

Figure 8 shows results of the proposed VDC strategy applied to the experimental setup. The effect of the deenergizing part is clearly seen as the backpressure $p_A$ starts to decrease quickly after applying the controller.

![Figure 8. Results of the activation of the VDC. (Top) The pressures over time, (Bottom) The command signal originating from the different parts of the VDC dead band strategy.](image)

The combination of a 3-port counter balance valve and the Danfoss pressure control valve in this application causes the VDC to only work for pressure $p_2$ above approximately 50bar when pressure $p_A$ is drained to tank, see Figure 8 (Top). Alternatively, the counter balance valve could be changed to a 4-port version, where the backpressure does not influence the opening of the valve.

4.2.1. Settling time

Table 2 shows the settling times for both the CDC and the VDC. The different results for the VDC strategy are an outcome of the high variation in pressure levels in the system when the spool valve is in neutral. As seen, the difference is more than a factor 3 between the ramp up and the ramp down. The settling time for VDC (after moving up), seems a bit high, but it requires only around half the time getting to 90% of the settling pressure, VDC (up, 90%). For practical use the 90% pressure will be sufficient to avoid the operator experiencing the response of the system to be slower. This is because the spool travel required between 90% and 100% settling pressure is minimal.

The settling time for CDC is independent of previous movement since it is a fixed value dead band.
Table 2. Settling time before movement for different strategies

<table>
<thead>
<tr>
<th></th>
<th>CDC</th>
<th>VDC (up)</th>
<th>VDC (up, 90%)</th>
<th>VDC (down)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Settling time</td>
<td>110ms</td>
<td>599ms</td>
<td>349ms</td>
<td>196ms</td>
</tr>
</tbody>
</table>

4.2.2. Dead time

Results from the dead time experiments are shown in Figure 9 (left) and the numerical values are presented in Table 3. The effect of the dead band compensation is clearly seen here on the response time with CDC being a little faster than NDC and VDC being clearly faster than the two others.

![Figure 9](image)

Figure 9. (left) Dead time before movement for different strategies. (right) The pressure $p_B$ over time when extending the telescopic system.

Table 3. Dead time before movement for different strategies.

<table>
<thead>
<tr>
<th></th>
<th>NDC</th>
<th>CDC</th>
<th>VDC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dead time</td>
<td>162ms</td>
<td>138ms</td>
<td>89ms</td>
</tr>
</tbody>
</table>

4.2.3. Ability to adapt to load variations

The VDC controller has been tested for two different load cases to see how it reacts. Figure 9 (right) shows the pressure $p_B$ for the telescopic system extension at 0m and at 2m. In both cases the VDC lets the pressure $p_B$ settle nicely – ready for moving the crane.

4.3. B – Velocity control

This section shows some results of the VC with VM and VDC using the velocity ramp as input ($t_r = 1s$ and dist = 0.3m). In Figure 10 (left) the total command signal send to the valve is shown. Furthermore, the figure shows the signal from each part of the controller. Due to the correspondence between the signal from the feedforward branch (FF-term) and the total command signal, it is clear that the experimental mapping of the valve used in the feedforward is quite accurate. Figure 10 (right) shows how the pressures in the system develop over time when the velocity ramp is used as input.

The green dashed lines in Figure 10 - Figure 13 indicates when the velocity ramp is active.
4.3.1. Comparison of different control strategies

The different velocity control strategies – NM, CM, VM - are all tested with CDC, $t_r = 1s$ and dist = 0.3m. The constant pressure $p_b$ used in the constant mapping strategy have been chosen manually from Figure 10 (right) to $p_{b,\text{const}} = 156\text{bar}$.

The velocity responses are compared in Figure 11. As expected, does the VM exhibits the best performance, with the CM not far off and the NM as the slowest. Though the main difference is present when ramping up the velocity, all are capable of moving the cylinder with more or less the desired constant velocity. The peak on the CM curve while ramping up the speed up indicates that the chosen mapping is not very accurate for low velocities, which is confirmed by the results in Figure 13.
Table 4. RMSD-values for different control strategies with the same ramp.

<table>
<thead>
<tr>
<th></th>
<th>NM</th>
<th>CM</th>
<th>VM</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMSD Position</td>
<td>0.0176m</td>
<td>0.0012m</td>
<td>0.0009m</td>
</tr>
<tr>
<td>RMSD Velocity</td>
<td>0.0075m/s</td>
<td>0.0033m/s</td>
<td>0.0025m/s</td>
</tr>
</tbody>
</table>

Figure 12. Velocity error of the 3 different VC strategies; NM, CM and VM.

Monitoring the velocity errors compared to the reference velocity, see Figure 12, is another way of assessing the performance of the control system. A positive value for the velocity error is a result of the boom going too fast. As already mentioned, VM is able to handle the change of velocity better than the others. The RMSD values in Table 4, clearly show the same picture. CM and VM are much better at following the reference input than NM. However, VM also shows a clear improvement over CM, so the addition of the pressure transducer, $p_b$, pays off performance wise.

4.3.2. The effect of the constant mapping strategy, CM, at different speeds

It was shown in the previous section that the constant mapping strategy, CM, could provide usable performance under certain conditions. But since the pressure $p_{b,\text{const}}$ is found manually based on knowledge of the specific work cycle it is of interest to see the performance during different conditions. Figure 13 (left) shows how much of the total command signal origins from the feed forward running the velocity ramp using two different values of dist (0.1m and 0.3m) in order to alter the velocity obtained during the operation. A percentage higher than 100% means that the valve mapping returns too high flow demands and the P-controller needs to do more work to compensate. The red curve shows a good compliance of the valve mapping to what actually would be required by the system running steady state with a value around 103% opposite approximately 126% for the black. However, both show problems handling velocity variations at the beginning and at the end of the ramp. This implies that changing the velocity at which the crane operates, hence the pressure levels, the mapping becomes less accurate. Indicating CM would be less effective at dist = 0.1m.
The RMSD values for both position and velocity have been found, see Table 5. The position error (the RMSD value), has increased by more than a factor 3 as a result of a load case the controller is not designed for. The RMSD value for the velocity is higher for dist 0.3m than for 0.1m which is surprising. The explanation can be found by looking at the velocity response for dist 0.1m in Figure 13 (right) where a high peak can be observed when accelerating. The inaccurate mapping returns too high command signals. Due to this, the system will in reality experience a step input when the controller is activated. This creates the peak. The controller handles well getting the speed back at the reference which is why the RMSD velocity value is fine. However, there is no compensation position wise for the distance travelled during the peak, that can cause the RMSD value to be more than a factor 3 larger than for dist = 0.3m. The CM can be concluded not to be well suited for an application like this, where the load situation changes much during operation.

![Figure 13. (left) The effect of feedforward compared to the total command signal using the constant mapping strategy, CM, with $p_{\text{const}} = 156\text{bar}$, for dist = 0.1m and 0.3m. (right) Velocity response of CM CDC with $t_r = 1\text{s}$ and dist = 0.1m.](image)

4.3.3. VM at different speeds

In this section the VM strategy is investigated further. In Table 6 RMSD values for experiments with different values for dist (0.1m, 0.2m and 0.3m) are shown. These experiments were carried out using VDC instead of CDC as in section 4.3.1. The improvement in the overall performance for the VM strategy, when using VDC, can be seen when comparing the RMSD values for dist = 0.3m, with the corresponding results in Table 4. Both RMSD values are smaller but it is most significant for the RMSD Position. The RMSD position values for dist=0.1m, 0.2m and 0.3m are approximately equal and the RMSD velocity values gradually decrease from the already fine values at dist=0.3m. It can therefore be concluded that the controller adapts well to changes of the load scenario.

![Table 6. VM: different values of dist with $t_r = 1\text{s}$.](image)
To assess the quality of the mapping used in VM at different speeds the feed forward’s share of the total command signal is shown in Figure 14. The mapping is accurate for dist = 0.3m, accounting for approximately 98% of the control signal when running steady state. The results for dist = 0.1m and dist = 0.2m are not quite as impressive as the one for dist = 0.3m but with a feedforward branch accounting for around 94% of the total signal the mapping still is good.

Figure 14. The effect of the variable mapping strategy, VM, with 3 different speeds (different values of dist).

4.3.4. VM for different values of rise time, tr

The final investigating is to test the controller with control inputs with different rise times, tr. In Table 7 is listed RMSD values for rise times; 0.1s, 0.5s, 1s, and 2s. The values for the slower rise times of 1s and 2s are quite similar and gradually increasing in size the faster the rise time become. Rise time 0.1s have a RMSD velocity value nearly twice the size of the one for tr = 2s. This indicates that the system is not fast enough to handle these accelerations properly. A rate limiter could be considered to solve this issue.

Table 7. VM: different values of tr with dist = 0.3m.

<table>
<thead>
<tr>
<th></th>
<th>tr = 0.1s</th>
<th>tr = 0.5s</th>
<th>tr = 1s</th>
<th>tr = 2s</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMSD Position</td>
<td>0.0009m</td>
<td>0.0009m</td>
<td>0.0007m</td>
<td>0.0007m</td>
</tr>
<tr>
<td>RMSD Velocity</td>
<td>0.0047m/s</td>
<td>0.0035m/s</td>
<td>0.0025m/s</td>
<td>0.0021m/s</td>
</tr>
</tbody>
</table>

5. CONCLUSIONS

Two issues caused by the use of pressure control valves were addressed: the load-dependent dead band and the load-dependent metering-in flow.

A variable 3-step dead band compensation routine (VDC) was developed. Experiments showed that it is capable of reducing the time until movement can be detected from 162ms with no dead band compensation to 89ms with the controller activated. Furthermore the VDC was tested in different load situations and showed to be able to adjust to the varying load – until 50bar, which represents the lower limit of valve mapping. However in order to function the VDC requires that pressure \( p_A \) is measured. The VDC was also compared to a constant dead band compensation (CDC), based on the dead band specified in the datasheet...
for the valve. Experiments show a dead time of 139ms on this, which is some improvement compared to the result with no compensation and it is easy to implement. However since it is a fixed value in all cases it does not address the load dependency of the valve, and higher pressure levels will result in longer dead times. Of course a custom-made compensation which does not need any sensors could be designed for a specific work-cycle, but that was not in the scope of this paper.

To address the issue of the load-dependent metering-in flow a feedback control scheme was investigated. It used the integrated operator velocity (joystick) input as position reference and a feed forward based on the current metering-in pressure level and an a-priori experimental mapping of the pressure control valve. It is referred to as VM (Variable Mapping) because it computes the feedforward term based on the measured pressure. This compares to CM (Constant Mapping) that was also investigated where the feed forward term is constant. Finally, a pure feedback controller with no mapping (NM) was also subjected to the same experiments as VM and CM. The results were rather unanimously in the favor of VM with respect to velocity and position accuracy in general. Also, it showed to be well suited to handle variations in load and velocity reference.

REFERENCES


P-TYPE ITERATIVE LEARNING CONTROL FOR TWO COUPLED HYDRAULIC CYLINDERS

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ABSTRACT

In this contribution, a proportional ILC algorithm (P-ILC) in combination with an underlying cascaded backstepping control structure is applied to a system of two rigidly coupled hydraulic cylinders. The P-ILC represents a model-free algorithm, which adjusts the desired trajectory for the current iteration based on information about the tracking error in the previous iteration. In the given case, P-ILC provides the desired trajectory for the cylinder position. The outer loop of the backstepping control involves an accurate tracking of both the desired position and the desired force of the coupled cylinders. It determines the desired difference pressures in both hydraulic cylinders for the inner control loops. In these inner control loops, the difference pressures of the hydraulic cylinders are controlled with high bandwidth, where the volume flows through two servo valves into the cylinders serve as control inputs. The proposed control algorithm is finally implemented and validated at a dedicated test rig.

KEYWORDS: hydraulic cylinder, control-oriented modelling, iterative learning control, backstepping control

1. INTRODUCTION

In the case of repetitive trajectory tracking tasks, iterative learning control (ILC) approaches are the first choice and outperform classical control designs. They are widely used and well suited for all precise motion control applications, where the motion task is repeated infinite times. Already within a few number of iterations, the tracking error is reduced significantly and remains small in the sequel. An overview of available ILC algorithms is given in [1] and [2].

In this contribution, a P-type ILC for the cylinder position is presented based on an underlying tracking control structure. For this purpose, a nonlinear cascaded control approach is employed using backstepping techniques. A typical application is given by endurance tests for hydraulic steer-by-wire systems. Hydraulic steer-by-wire systems offer a higher precision as well as a superior performance, allow for a mass reduction as compared to classical hydraulic steering systems, and require less installation space. The steering axis is usually actuated by a single hydraulic cylinder. Depending on safety requirements, however, one or two hydraulic valves are used to provide the necessary inlet and outlet volume flows for the hydraulic cylinder. For long-terms test of all the hydraulic components involved, accurate and reproducible position and load profiles have to be guaranteed, where the proposed ILC approach could be applied.

Nonlinear control approaches for position-controlled hydraulic cylinders with one servo valve are presented
in [3] and [4]. In [5] and [6], sophisticated approaches for force-controlled hydraulic cylinders are described.

Further applications of the proposed backstepping control design to fluidic systems – especially pneumatic systems – can be found in [7] and [8].

A hardware-in-the-loop (HIL) test rig for component tests as well as the development and validation of sophisticated control structures for hydraulic cylinders is available at the Chair of Mechatronics at the University of Rostock, see Fig. 1. The test rig consists of two rigidly coupled hydraulic cylinders, which are actuated each by a separate servo valve. One cylinder represents the specimen and is usually employed for an accurate position control, the other one for the generation of an application-specific load force. To guarantee a high bandwidth at force generating, two hydraulic capacities are directly installed in front of the second servo valve to maintain a constant pump pressure. Furthermore, in each hydraulic chamber a pressure sensor is available, and the actual position of the first hydraulic cylinder is measured. Due to the rigid mechanical connection between the cylinders as well as a geometric adjustment of the middle positions of both pistons, a further sensor for the second servo cylinder is not necessary. In between both cylinders, an additional force sensor is integrated that allows for a force measurement. With this sensor combination, hence, an accurate identification of the system parameter as well as the volume flow maps becomes possible. In this paper, a highly accurate nonlinear control concept as well as corresponding experimental results from the HIL test rig are presented.

![Figure 1. HIL test rig for the hydraulic system with two coupled cylinders.](image)

The paper is structured as follows: First, a physically-oriented state-space model of the mechatronic system is derived and further simplified in a subsequent model-order reduction step. Second, a P-type ILC is designed for the cylinder position and combined with a cascaded underlying control structure based on backstepping techniques. Experimental results show the convergence of the tracking error for the cylinder position and point out the advantages of the proposed control approach with only small tracking errors during transient phases as well as a negligible steady-state control error.

2. MODELLING OF THE MECHATRONIC SYSTEM

The mechatronic system can be split into a mechanical and a hydraulic subsystem. The mechanical system part covers the joint motion of the rigidly connected piston rods. The hydraulic subsystem describes the pressure dynamics in the cylinder chambers.
2.1. Mechanical Subsystem

The considered operation range of the system is characterized by values $-l_{\text{max}} < z_e(t) < l_{\text{max}}$, cf. Fig. 2. To differentiate between the two hydraulic cylinders, the index 1 is used for the left cylinder, whereas 2 denotes the right cylinder. The pressures $p_{1,A}(t)$ and $p_{1,B}(t)$ represent the two absolute pressures in the right and left hydraulic chamber of the first cylinder. The corresponding force on the piston is given by $F_1(t) = A_1 \left(p_{1,A}(t) - p_{1,B}(t)\right)$. Here, the parameter $A_1$ stands for the piston area of the first cylinder. For the second cylinder, the driving force is given by $F_2(t) = A_2 \left(p_{2,A}(t) - p_{2,B}(t)\right)$. The absolute pressures are denoted by $p_{2,A}(t)$ and $p_{2,B}(t)$, and $A_2$ represents the piston area of the second cylinder. Furthermore, a velocity-proportional damping force $F_D(t) = z_e(t) b_D$ is considered in the model.

![Figure 2. Mechatronic model of the test rig.](image)

A balance of momentum yields the equation of motion in the form of a second-order differential equation

$$\ddot{z}_e(t) = \frac{1}{m} \left[-z_e(t) b_D + A_1 \left(p_{1,A}(t) - p_{1,B}(t)\right) + A_2 \left(p_{2,A}(t) - p_{2,B}(t)\right) - F_U(t)\right],$$

(1)

with $m$ as the reduced mass of all the moving components connected to the hydraulic cylinders. Model uncertainty and nonlinear friction could be taken into account by a lumped disturbance force $F_U(t)$ in the case of adaptive backstepping design. In this contribution, however, iterative learning control counteracts these disturbances.

2.2. Hydraulic Subsystem

A mass flow balance for one of the four cylinder chambers, $i \in \{1, 2\}$ and $j \in \{A, B\}$, directly leads to

$$\frac{dm_{i,j}(t)}{dt} = \dot{p}_{i,j}(t) \cdot V_{i,j}(z_e(t)) + \rho_{i,j}(t) \cdot \dot{V}_{i,j}(z_e(t)) = \rho_{i,j}(t) q_{i,j}(t).$$

(2)

Here, the density $\rho_{i,j}(t)$, the chamber volume $V_{i,j}(z_e(t))$ and the volume flow $q_{i,j}(t)$ into the corresponding cylinder chamber are introduced. The elastic behaviour of the hydraulic fluid is described by

$$dp_{i,j}(t) = E(p_{i,j}) \frac{dV_{i,j}}{V_{i,j}},$$

(3)

with the bulk modulus $E(p_{i,j})$ of the hydraulic fluid, which in general depends on the pressure. In this paper, however, $E$ can be assumed with high accuracy as constant in the given pressure range. Mass conservation leads to the relationship

$$- \frac{dV_{i,j}}{V_{i,j}} = \frac{dp_{i,j}}{\rho_{i,j}}.$$ 

(4)
This results in the pressure dynamics

\[ \dot{p}_{i,j}(t) = -\frac{E V_{i,j}(\dot{z}_c)}{V_{i,j}(z_c)} + \frac{E}{V_{i,j}(z_c)} q_{i,j}(t). \]  

(5)

The differential equations for the pressures in the chambers \( j \in \{A, B\} \), of cylinder \( i \in \{1, 2\} \) become

\[ \dot{p}_{i,A}(t) = -\frac{E A_i \dot{z}_c(t)}{V_{i,0} + A_i z_c(t)} + \frac{E}{V_{i,0} + A_i z_c(t)} q_{i,A}(t), \]

\[ \dot{p}_{i,B}(t) = -\frac{E A_i \dot{z}_c(t)}{V_{i,0} - A_i z_c(t)} + \frac{E}{V_{i,0} - A_i z_c(t)} q_{i,B}(t). \]

(6)

Here, the chamber volumes are characterized by \( V_{i,A} = V_{i,0} + A_i z_c(t) \) and \( V_{i,B} = V_{i,0} - A_i z_c(t) \). The volume flows \( q_{i,A}(t) \) and \( q_{i,B}(t) \) serve as control inputs.

2.3. Model-Order Reduction and Derivation of Decentralized Models

An overall nonlinear state-space model for the whole test rig in the form \( \dot{x} = f(x, u) \) can be stated as

\[
\dot{x} = \begin{bmatrix}
\dot{z}_c \\
\dot{z}_c \\
\dot{p}_{1,A} \\
\dot{p}_{1,B} \\
\dot{p}_{2,A} \\
\dot{p}_{2,B}
\end{bmatrix} = \begin{bmatrix}
\frac{1}{m} \left[-\dot{z}_c b_D + A_1 (p_{1,A} - p_{1,B}) + A_2 (p_{2,A} - p_{2,B}) - F_U\right] \\
- \frac{E A_1 \dot{z}_c}{V_{1,0} + A_1 z_c} + \frac{E}{V_{1,0} + A_1 z_c} q_{1,A} \\
- \frac{E A_1 \dot{z}_c}{V_{1,0} - A_1 z_c} + \frac{E}{V_{1,0} - A_1 z_c} q_{1,B} \\
- \frac{E A_2 \dot{z}_c}{V_{2,0} + A_2 z_c} + \frac{E}{V_{2,0} + A_2 z_c} q_{2,A} \\
- \frac{E A_2 \dot{z}_c}{V_{2,0} - A_2 z_c} + \frac{E}{V_{2,0} - A_2 z_c} q_{2,B}
\end{bmatrix},
\]

(7)

with the state vector \( x = [z_c \ \dot{z}_c \ p_{1,A} \ p_{1,B} \ p_{2,A} \ p_{2,B}]^T \), the input vector \( u = [q_{1,A} \ q_{1,B} \ q_{2,A} \ q_{2,B}]^T \) and the output vector \( y = [z_c \ ((p_{1,A} - p_{1,B}) \cdot A_1 - (p_{2,A} - p_{2,B}) \cdot A_2)/2]^T \). This nonlinear model, however, turns out to be not fully controllable.

In the following, a model-order reduction is performed. For this purpose, two new state variables in the form of the difference pressures \( \Delta p_i(t) = p_{i,A}(t) - p_{i,B}(t) \) are introduced, where \( i \in \{1, 2\} \) indicates the individual cylinder. The corresponding differential equation can be stated as

\[ \Delta \dot{p}_i(t) = \dot{p}_{i,A}(t) - \dot{p}_{i,B}(t) = -\frac{E A_i \dot{z}_c}{V_{i,A}(z_c)} + \frac{E}{V_{i,A}(z_c)} q_{i,A}(t) - \frac{E A_i \dot{z}_c}{V_{i,B}(z_c)} - \frac{E}{V_{i,B}(z_c)} q_{i,B}(t) . \]

(8)

Furthermore, the relationship between the volume flow into chamber \( A \) and out of chamber \( B \) is given by \( q_{i,A}(t) = -q_{i,B}(t) \). Therefore, the effective volume flow for the difference pressure is defined as \( q_{i,AB}(t) = q_{i,A}(t) - q_{i,B}(t) = 2q_{i,A}(t) \). The product of the volumes can be written as \( V_{i,A}(z_c) \cdot V_{i,B}(z_c) = V_{i,0}^2 - (A_i \cdot z_c)^2 \), and the sum becomes \( V_{i,A}(z_c) + V_{i,B}(z_c) = 2 \cdot V_{i,0} \), with \( V_{i,A}(z_c) = V_{i,0} + A_i z_c \) and \( V_{i,B}(z_c) = V_{i,0} - A_i z_c \). The resulting differential equation for the difference pressure dynamics is

\[ \Delta \dot{p}_i(t) = \frac{-2EA_i V_{i,0} \dot{z}_c}{V_{i,0}^2 - (A_i \cdot z_c)^2} + \frac{E V_{i,0}}{V_{i,0}^2 - (A_i \cdot z_c)^2} q_{i,AB}(t). \]

(9)

Note that this formulation for the difference pressure dynamics holds for both hydraulic cylinders. Thereby, the
dimension of the state vector is reduced from \( \dim(x) = 6 \) to \( \dim(x) = 4 \). The dynamics of the multi-variable system is governed by

\[
\dot{x} = \begin{bmatrix}
\dot{z}_c \\
\hat{z}_c \\
\Delta p_1 \\
\Delta p_2
\end{bmatrix} = \begin{bmatrix}
-b z_c + A_1 \Delta p_1(t) + A_2 \Delta p_2(t) - \frac{F_u}{m} \\
-2EA_1 V_{1,0} \hat{z}_c + \frac{EV_{1,0}}{m} \\
-2EA_2 V_{2,0} \hat{z}_c + \frac{EV_{2,0}}{m} \\
V_{1,0}^2 - (A_1 z_c)^2 + V_{2,0}^2 - (A_2 z_c)^2 q_{1,AB}(t)
\end{bmatrix}.
\]

The corresponding state vector is chosen as \( \bar{x}(t) = [z_c(t) \; \dot{z}_c(t) \; \Delta p_1(t) \; \Delta p_2(t)] \), the input vector is given by \( u(t) = [q_{1,AB}(t) \; q_{2,AB}(t)] \). The position \( z_c(t) \) and force \( F(t) = (A_1 \Delta p_1(t) - A_2 \Delta p_2(t))/2 \) are considered as controlled outputs.

2.4. Valve Characteristic

The volume flow through a hydraulic valve is usually modelled as an ideal turbulent resistance with variable cross-section. At the given test rig, the two valves have an analogue voltage signal as input, respectively. The volume flow through a hydraulic resistance with variable cross-section area is given by

\[ q_i = B_{1,V}(v_i) \cdot \sqrt{\Delta p_v}, \]

with \( B_{1,V}(v_i) \) as the valve conductance, which depends directly on the valve input signal \( v_i \). The pressure difference \( \Delta p_v \) is defined between the pressures in front of and behind the valve. For instance, the volume flow for the first cylinder is given by

\[ q_{1,A} = \begin{cases} 
0 & \text{for } v_1 = 0, \\
B_{1,V} \cdot v_1 \cdot \sqrt{p_p - p_{1,A}} & \text{for } v_1 > 0, \\
B_{1,V} \cdot v_1 \cdot \sqrt{p_{1,A} - p_t} & \text{for } v_1 < 0,
\end{cases} \]

and

\[ q_{1,B} = \begin{cases} 
0 & \text{for } v_1 = 0, \\
-B_{1,V} \cdot v_1 \cdot \sqrt{p_{1,B} - p_t} & \text{for } v_1 > 0, \\
-B_{1,V} \cdot v_1 \cdot \sqrt{p_p - p_{1,B}} & \text{for } v_1 < 0,
\end{cases} \]

with the tank pressure \( p_t \), and the supply pressure \( p_p \) of the pump. As a result of the given symmetry of the cylinders and neglected leakage effects, the relationship \( p_p = p_{1,A} + p_{1,B} - p_t \) leads to \( q_{1,A} = -q_{1,B} \). The corresponding input of the system is given by \( q_{1,AB} = 2 \cdot q_{1,A} \). Based on an experimental identification, the valve characteristic depicted in Fig. 3 has been derived for the valve of the first cylinder.

3. CASCADED CONTROL DESIGN

For control design, a cascaded structure has been chosen. The mechanical subsystem (1) is controlled by an outer control loop, whereas the two hydraulic subsystems (8) are controlled by underlying control loops. In this way, the complexity of the control design is reduced and a fast control of the difference pressures is ensured. The proposed control structure is depicted in Fig. 4. In the following, the design of the inner control loops for the difference pressures using backstepping techniques is presented. Afterwards, the outer control
loop for the cylinder position as well as the cylinder force with backstepping methods is described. An overview of backstepping control is given by [9] or [10]. In the last section, the P-type ILC algorithm is presented for a correction of the desired cylinder position.

3.1. Differential Flatness

A nonlinear system – which is usually given in state-space representation, i.e., $\dot{x} = f(x, u)$ – is denoted as differentially flat [11] if appropriate flat outputs $y = y(x, u, \dot{u}, ..., u^d)$ exist that:

(i) allow for expressing all system states $x$ and all system inputs $u$ as a function of these flat outputs $y$ as well as their time derivatives, i.e. $x = x(y, \dot{y}, ..., y^{(\beta)})$ and $u = u(y, \dot{y}, ..., y^{(\beta+1)})$,

(ii) are differentially independent, i.e., they are not connected by differential equations.

If the first condition is fulfilled, the second condition is equivalent to $\dim(y) = \dim(u)$. In this paper, a cascaded control approach is considered consisting of fast inner control loops with the difference pressures $\Delta p_i$ as flat outputs, and with the position $z_c$ as well as the force $F = (A_1 \Delta p_1 - A_2 \Delta p_2)/2$ as flat outputs for the outer control...
loop. The corresponding error dynamics is advantageously stabilised by backstepping techniques.

3.2. Inner control loops for the difference pressures of the hydraulic cylinders

In the hydraulic subsystems, the first time derivative of the flat output candidate \( y_{f,p}(t) = \Delta p_i(t) \) results in

\[
\Delta \dot{p}_i(t) = -\frac{2 E A_i V_{i,0} \dot{z}(t)}{V_{i,0}^2 - (A_i \cdot \dot{z}_c(t))^2} + \frac{E V_{i,0}}{V_{i,0}^2 - (A_i \cdot \dot{z}_c(t))^2} q_{i,AB}(t)
\]  

(14)

and is affected by the control input. Hence, equation (14) can be solved for the input variable \( q_{i,AB}(t) \). This results in the following inverse model depending on the flat output and its first time derivative

\[
q_{i,AB}(t) = \frac{\upsilon_{\Delta p,i}(t) \cdot (V_{i,0}^2 - (A_i \cdot \dot{z}_c(t))^2)}{E V_{i,0}} + 2 A_i \dot{z}_c(t) \, ,
\]  

(15)

with \( \upsilon_{\Delta p,i} = \Delta \dot{p}_i \) as the stabilizing control input. For the hydraulic subsystems the dynamics of the tracking error \( e_{\Delta p,i}(t) = \Delta p_{i,d}(t) - \Delta p_i(t) \) have to be stabilized

\[
e_{\Delta p,i}(t) = \Delta p_{i,d} - \Delta p_i \, ,
\]  

(16)

where the stabilizing input \( \upsilon_{\Delta p,i} = \Delta \dot{p}_i \) have to guarantee the convergence of the pressure tracking error \( e_{\Delta p,i} \) to zero via special feedback design. Therefore, a quadratic Lyapunov function is chosen

\[
V_p(e_{\Delta p,i}) = \frac{1}{2} e_{\Delta p,i}^2 > 0 \, .
\]  

(17)

Its time derivative has to be negative definite

\[
\dot{V}_p(e_{\Delta p,i}) = e_{\Delta p,i} \cdot \dot{e}_{\Delta p,i} = e_{\Delta p,i} \cdot (\Delta p_{i,d} - \Delta p_i) = -c_{i,p} e_{\Delta p,i} \, .
\]  

(18)

Accordingly, the bracket in (18) is set to \(-c_{i,p} \cdot e_{\Delta p,i}\), where the positive parameter \( c_{i,p} > 0 \) can be used to specify the error dynamics. Solving for the stabilising control input results in

\[
\upsilon_{\Delta p,i} = \Delta p_i(t) = \Delta \dot{p}_i(t) = \Delta p_{i,d}(t) + c_{i,p} \cdot e_{\Delta p,i}(t) \, .
\]  

(19)

The stabilising control law is given by equation (19) and can be inserted into the inverse dynamics (15). This leads to

\[
q_{i,AB}(t) = \frac{\left(\Delta p_{i,d}(t) + c_{i,p} \cdot e_{\Delta p,i}(t)\right) \cdot (V_{i,0}^2 - (A_i \cdot \dot{z}_c(t))^2)}{E V_{i,0}} + 2 A_i \dot{z}_c(t) \, .
\]  

(20)

3.3. Outer control loop for the cylinder position and the cylinder force

The mechanical system part also represents a differentially flat system with the position \( z_c \) of the first hydraulic cylinder and the force \( F = (A_1 \Delta p_1 - A_2 \Delta p_2)/2 \) as flat outputs. Subsequent differentiations of the first flat output – until one of the control inputs \( \Delta p_1 \) or \( \Delta p_2 \) appears – lead to

\[
y_1(t) = \dot{z}_c(t) \, , \quad \dot{y}_1(t) = \ddot{z}_c \quad \text{and} \quad \ddot{y}_1(t) = \dddot{z}_c(t) = \frac{1}{m} \left[ -\dot{z}_c(t) b_D + A_1 \Delta p_1(t) + A_2 \Delta p_2(t) - F \hat{u}(t) \right] \, ,
\]  

(21)
whereas the second variable directly depends on the control inputs
\[ y_2(t) = F(t) = \frac{(A_1 \Delta p_1(t) - A_2 \Delta p_2(t))}{2}. \]  

(22)

The inverse dynamics can be obtained by solving equations (21) and (22) for the input variables \( \Delta p_1(t) \) and \( \Delta p_2(t) \). Hence, the input vector \( \vec{u}(t) \) depending on the desired force and the control input \( \nu_c(t) = \dot{z}_c(t) \) is given by
\[
\vec{u}(t) = \begin{bmatrix} \Delta p_1(t) \\ \Delta p_2(t) \end{bmatrix} = \begin{bmatrix} \frac{1}{2A_1} (m \cdot \nu_c(t) + 2F(t) + bD \dot{z}_c(t) + F_U(t)) \\ \frac{1}{2A_2} (m \cdot \nu_c(t) - 2F(t) + bD \dot{z}_c(t) + F_U(t)) \end{bmatrix}.
\]

(23)

The first subsystem of the outer control loop to be stabilized results from the definition of the tracking error \( e_1 = \dot{z}_{c,d} - \dot{z}_c \). Its time derivative becomes
\[
\dot{e}_1 = \dot{z}_{c,d} - \dot{z}_c.
\]

(24)

Here, the virtual control input is given by \( \alpha_c \approx \dot{z}_c \). As before, a quadratic Lyapunov function \( V_1(e_1) = 1/2 \cdot e_1^2 \) is used to stabilize the error dynamics. The virtual control input can be obtained by specifying a negative definite time derivative \( V_1(e_1) \)
\[
V_1(e_1) = e_1 \cdot \dot{e}_1 = e_1 \cdot \left( \dot{z}_{c,d} - \alpha_c \right),
\]

(25)

with positive parameters \( c_1 > 0 \) and \( c_2 > 0 \). In this way, \( \alpha_c \) becomes
\[
\alpha_c = \dot{z}_{c,d} + c_1 \cdot e_1 + c_2 \cdot e_1^2.
\]

(26)

For the next design step, the error variable \( e_2 \) is introduced
\[
e_2 = \alpha_c (\dot{z}_{c,d}, e_1) - \dot{z}_c = c_1 \cdot e_1 + c_2 \cdot e_1^2 + \dot{z}_{c,d} - \dot{z}_c.
\]

(27)

The corresponding error dynamics is given by
\[
\dot{e}_2 = c_1 \cdot \dot{e}_1 + 3 \cdot c_2 \cdot e_1^2 \cdot \dot{e}_1 + \dot{z}_{c,d} - \nu_c,
\]

(28)

with the stabilising input \( \nu_c = \dot{z}_c \). Again, an extended Lyapunov function
\[
V_2(e_1, e_2) = V_1(e_1) + \frac{1}{2} \cdot e_2^2
\]

(29)

is used for stabilisation. Its time derivative has to be made negative definite
\[
V_2(e_1, e_2) = e_1 \cdot \dot{e}_1 + e_2 \cdot \dot{e}_2 = -c_1 \cdot e_1^2 - c_2 \cdot e_1^4 + e_2 \cdot (\dot{z}_{c,d} - \nu_c + g_1(e_1, e_2)),
\]

(30)

with \( c_3 > 0 \) and \( c_4 > 0 \). By setting the bracket in (30) equal to \( -c_3 \cdot e_2 - c_4 \cdot e_2^3 \) and solving for the input \( \nu_c \), the stabilising control law follows as
\[
\nu_c = \ddot{z}_{c,d} + g_2(e_1, e_2).
\]

(31)

The control input \( \nu_c \) has to be inserted in the inverse dynamics (23), where the disturbance force is neglected using \( F_U(t) = 0 \). The nonlinear error dynamics of the closed loop can be specified by the design
3.4. Iterative Learning control

ILC is a powerful means for the reduction of tracking errors in repetitive motion tasks. For this purpose, the tracking error of the current iteration is employed to improve the control behaviour for the future iterations [14]. Repetitive tasks can be found in many industrial applications, such as automated manufacturing systems, chemical processes or robotics. Since iterative learning control was introduced by [15], many different algorithms have been derived. An overview of ILC is presented in [1] or [16]. In principle, ILC can be implemented as a pure feedforward control strategy. To account for unrepeatable disturbances acting on the plant during each iteration, the ILC approach can be combined with an additional feedback controller as described in subsections 3.2 and 3.3. There are also approaches that directly include feedback actions in the ILC update law, e.g., NOILC, cf. [17], [18]. The P-type ILC, which is employed in this contribution, represents a very popular form of ILC thanks to its simplicity. However, a main disadvantage of the P-ILC approach is given by its insufficient convergence properties present in many practical applications, cf. [14], [19]. To overcome this problem, here, a zero-phase filtered ILC with phase-lead compensation as proposed in [20] or [14] is employed. The implemented ILC control law depends on the tracking error \( e_j(k) = y_d,j(k) - y_j(k) \) at time step \( k \) and iteration \( j \).

For the system under consideration, \( y_j(k) \) corresponds to the position \( z_j(k) \) of the hydraulic cylinder. The P-ILC update law is given by

\[
 u_{j+1}(k) = u_j(k) + \Phi_1 f(e_j(k + l)) ,
\]

where \( \Phi_1 \) is the ILC learning gain. Now, to improve the behaviour of the learning process, a zero-phase low-pass filter \( f \) is used to counteract high-frequency noise and uncertainties that may cause an undesired learning behaviour. Furthermore, a phase-lead compensation is used to increase the learning bandwidth and to improve the error convergence. The design parameters for the ILC are given by the learning gain \( \Phi_1 \), the cut-off frequency \( \omega_0 \) of the low-pass filter, and the linear phase lead \( l \). The resulting ILC learning law becomes

\[
 u_{j+1}(k) = u_j(k) + \Phi_1 f(e_j(k + l)) .
\]

To obtain a criterion for the choice of the available design parameters, the ILC update law is transferred into the complex \( z \)-domain

\[
 U_{j+1}(z) = U_j(z) + \Phi_1 z^l F(z) E_j(z) ,
\]

where \( F \) denotes the filter transfer function. The iterative error dynamics results in

\[
 E_{j+1}(z) = (1 - \Phi_1 z^l F(z) G(z)) E_j(z) ,
\]

where \( G(z) = \frac{Y(z)}{U(z)} \) represents the transfer function of the closed-loop system. Consequently, the condition for monotonic error decay becomes [14], [21]

\[
 \left| 1 - \Phi_1 e^{j\omega T_c} F(e^{j\omega T_c}) G(e^{j\omega T_c}) \right| < 1 .
\]

In (36), \( z \) has been substituted by \( e^{j\omega T_c} \), where \( T_c \) characterizes the sampling time. For the implementation, Nyquist plots of \( G_0 = \Phi_1 e^{j\omega T_c} \hat{G}(e^{j\omega T_c}) \) can be investigated, where \( \hat{G} \) denotes the measured or estimated frequency response of the real system \( G \). The condition (36) is fulfilled for all the frequencies for which the Nyquist plot remains inside a unit circle with centre \((1,0)\). The frequency at which the Nyquist plots leaves the unit centre for the first time represents the maximum value possible for the cut-off frequency. With increasing values for the phase lead \( l \), the Nyquist plot can be maintained within the unit circle even for higher frequencies.
In this way, the introduction of the linear phase-lead allows for higher maximum values of the cut-off frequency.

3.5. P-type ILC for the hydraulic cylinder system

As depicted in Fig. 4, the ILC algorithm is employed to modify the desired trajectory of the cylinder position $z_c$. The zero-phase filtering of the tracking error $e_{z,j}(k) = z_{c,d}(k) - z_{c,j}(k)$ can be calculated offline after every iteration. In the given case, a second-order low-pass filter is utilized for this purpose. The original frequency response of the closed loop system is estimated by the frequency response of the simulated system. In Fig. 5, the Nyquist plots are depicted for a gain $\Phi_1 = 1$ and a phase lead $l = 0$ as well as for a gain $\Phi_1 = 0.5$ and a phase lead of $l = 28$. It becomes obvious that the maximum possible cut-off frequency for $l = 28$ ($\omega_{c,\text{max}}(l = 28) = 135$ rad/s) is significantly higher than the one for $l = 0$ ($\omega_{c,\text{max}}(l = 0) = 40$ rad/s). Due to uncertainties related to an approximated frequency response, the cut-off frequency in the implementation at the test rig is chosen smaller than $\omega_{c,\text{max}}$. In the implementation, a gain $\Phi_1 = 0.5$, a phase-lead $l = 28$ and a cut-off frequency $\omega_c = 80$ rad/s have been chosen.

4. EXPERIMENTAL RESULTS

In the following, experimental results for the coupled hydraulic cylinders are presented to point out the benefits of P-ILC control design techniques. Each electro-hydraulic valve is connected to a separate hydraulic pump, with a supply pressure of $p_{1,\text{pump}} = p_{2,\text{pump}} = 120 \cdot 10^5$ Pa. The desired trajectory for the position as well as its first time derivative are shown in Fig. 6. The desired value for the force is chosen as zero according to
In the first iteration, the position errors depicted in Fig. 7 are obtained. Note that the P-ILC does not have an impact in the first iteration because the initialization error is zero. In the subsequent iterations, the resulting position error $e_z = z_{c,d} - z_c$ is used as input for the P-ILC. It can be seen from Fig. 7 that the steady-state error is below $|e_z| < 0.05$ mm and, hence, very small. Only in transient phases, a small position error occurs. The corresponding difference pressure dynamics for both cylinders are depicted in Fig. 8.

The main focus in this contribution is on the benefits of the proposed P-ILC. Therefore, a long time experiment with 50 iterations is performed. To visualize the convergence of the algorithm the root-mean square error is calculated according to

$$e_{\text{RMS}} = \sqrt{\frac{1}{N} \sum_{k=1}^{N} (z_{c,d}(k) - z_c(k))^2}$$

in each iteration. In Fig. 9, the achieved rms errors are depicted in dependence of the iteration number. During only eight iterations the root-mean square error decreases significantly and remains constant afterwards with values $e_{\text{RMS}} < 1 \cdot 10^{-5}$ m.
To point out the decrease of the maximum position errors, which can be seen in Fig. 7, the tracking error of the first, third, fifth and seventh iteration are depicted in Fig. 10. It can be seen that the tracking error decreases in the transient phases of the trajectory. After seven iterations, the tracking error for the cylinder position is below $|e_{z,\text{max}}| < 0.07$ mm in both steady-state and in transient phases.

![Figure 10. Tracking error for the cylinder position for selected iterations.](image)

5. CONCLUSIONS

In this contribution, a P-type iterative learning control algorithm is designed for precise position control of a system consisting of two coupled hydraulic cylinders. The ILC adapts the desired position for an underlying cascaded control structure. For this structure, a model-based backstepping design is employed. In the outer control loop, a decoupling control of the cylinder position and force is performed. Two fast inner control loops determine the difference pressures in both cylinders. The control performance and the efficiency of the proposed control structures are pointed out by experimental results from an implementation on a dedicated test rig at the Chair of Mechatronics, University of Rostock. The maximum position error in the transient phases of the first iteration, where the P-ILC does not affect the results, is approx. $|e_{z,\text{max}}| \approx 1.5$ mm. After only seven further iterations, the maximum position error remains below $|e_{z,\text{max}}| \approx 0.07$ mm in steady-state phases as well as in transient phases.

References


DESIGN OF DISTURBANCE OBSERVER OF ELECTRO-HYDRAULIC LOADING SYSTEM FOR HELICOPTER MANIPULATING BOOSTER

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ABSTRACT

Addressing the simulating issue of the helicopter manipulation booster under high-frequency aero-dynamic load with dynamic load superimposed on large static load, this paper has developed an electro-hydraulic loading system to simulate the load with sinusoidal dynamic load superimposed on large static load. The mathematical model of the electro-hydraulic loading system is firstly established. Then, double loops cascade composition control strategy is applied in loading system controller design. The inner-loop controller is expected to make the actual plant track a nominal model approximately; the outer-loop controller is expected to realize the desired force tracking performance. The disturbance observer based approach is adopted in inner-loop controller design to suppress the surplus force to be caused by the external motion and other disturbances, also to enhance system robustness to parameter perturbations and uncertainties. The low pass filter $Q(s)$ is designed by $H_{\infty}$ mixed sensitivity optimization method. The simulation results indicate that the surplus force is effectively suppressed using the proposed controller with $Q(s)$ filer, and meanwhile the robustness of the system can be also guaranteed.

KEYWORDS: electro hydraulic servo loading, surplus force, disturbance observer, robust control

1. INTRODUCTION

Helicopter manipulating booster is an actuating component in the helicopter manoeuvring system, which undertakes the transmitting of the manipulation displacement signal and the amplifying of the manipulating force, and meanwhile to meet the requirements to overcome the aero-dynamic load acted on the main rotor and the rail rotor. The booster is requested to have high accuracy and fast speed response from the driver’s command under the aero-dynamic load, which is passed from the helicopter rotor and tail driven by the booster. According to the characteristics of the aero-dynamic load acting on the helicopter rotor and tail, corresponding load spectrums are compiled. Through electro-hydraulic loading system, which is applied to simulate aero-dynamic loads on the ground, booster control performance with load can be evaluated, and this has vital significance in product performance test and improvement.

There is an unavoidable problem in loading system, how to suppress the interference introduced by the active movement of loaded plant. This active movement causes disturbance/forced flow in hydraulic load cylinder, generates a force output, which namely surplus force. In force closed loop, surplus force seriously influences the force tracking accuracy, reduces the closed-loop bandwidth of the loading system, and
seriously causes gradation in force control performance. Therefore, there are two major problems in loading system to be solved: force tracking and surplus force suppression.

Main research methods address to how to suppress surplus force at present are as follows: 1) robust control methods. Such as quantitative feedback theory (QFT) \[1,2\], \(H_\infty\) mixed sensitivity method \[3\], variable structure control \[4\], fuzzy control \[5\], Neural Network Control \[6\], and so on. The active motion of loaded plant is regarded as an external disturbance. The maximum disturbance is regarded as disturbance suppression index. System robustness is usually achieved at the price of the nominal performance, so the controllers designed are conservative; 2) nonlinear control methods \[7-9\]. Such methods focus on the nonlinear characteristics and parameters uncertainties, the active motion is also regarded as disturbance, the designed feedback controllers are also conservative; 3) compensation based control methods. Such as velocity synchronization method with introducing the command input of the position servo valve of the loaded plant \[10\], hybrid control method \[11\], and traditional velocity coefficient compensation method based on structure invariance principle. Feed forward compensation is utilized to attenuate surplus force in those methods without considering system robustness; 4) two degree of freedom based methods. Such as disturbance observer (DOB) based method \[12\], it utilizes a dual-loop control structure, and is designed for low frequency torque loading of electric actuator.

The difficulties in implementation of electro-hydraulic loading system for helicopter manipulating booster is mainly reflected in high frequency dynamic load superimposed on large static load, the static load reaches 15kN, the dynamic load amplitude reaches 10kN, the frequency of dynamic load is requested to 30Hz, even up to 80Hz.

At present the design theory and control strategy, about elastic load type electro-hydraulic loading system for fixed wing actuator, are relatively mature, but research about high frequency dynamic load superimposed on large static load type electro-hydraulic loading system, for manipulating booster of main rotor and tail rotor of rotorcraft, is relatively few. This paper mainly studies the disturbance observer (DOB) based inner loop controller with double loops cascade composition control strategy.

2. SYSTEM OVERVIEW AND ANALYSIS

2.1. System Composition and Working Principle

The studied hardware-in-the-loop electro-hydraulic loading system for helicopter manipulating booster is shown in Figure 1. As depicted, the manipulating simulation system on the left is a low-power electro-hydraulic position servo system, used to simulate the operating action on the control rod by driver’s hand. The loading target, helicopter manipulating booster, is in the middle. The booster is connected to control rod through the lever linkage, the rod action is reflected on the booster piston rod with amplifying the manipulating force. On the right is the electro-hydraulic loading system, the loading hydraulic cylinder piston rod is connected to the booster piston rod by force sensor. Load spectrums are imposed on the booster when the loading hydraulic cylinder follows the movement of booster. Symbols \(x_M\) and \(x_L\) represent the displacement of the booster piston rod and the loading hydraulic piston rod, respectively.
Motion spectrums are compiled in sinusoidal form, to simulate the operating action on the control rod by hand. When motion spectrum is at the maximum frequency 2Hz, its magnitude does not exceed 6mm; and when motion spectrum is at the maximum magnitude 15mm, its frequency does not exceed 1Hz. Load spectrums compiled to simulate aero-dynamic load are in the form of sinusoidal dynamic load superimposed on large static load, their frequency range is generally 20Hz to 80Hz.

During the whole hardware-in-the-loop simulation process, booster (position servo system) and loading system (force servo system) work synchronously under the control of central simulation computer. For the booster, loading force is a strong disturbance to its output displacement, which seriously impacts the output precision of the system. For loading system, the speed of booster is also a strong disturbance to its force output, which affects the tracking precision of the system output force similarly.

2.2. System Model and Analysis

The mathematical model of the electro-hydraulic loading system can be set up by means of theoretical analysis or experiment identification. The theoretical model has an important significance in the design of the controller and the selection of system parameters. Before modelling, it is assumed that the loading system works under ideal condition.

The load force $F_L(N)$ applied to the booster can be measured, and as depicted in Figure 1 it can be expressed as

$$F_L = K_s(x_L - x_M)$$

where $K_s$ is elastic stiffness of the force sensor (N/m).

The linear flow equation of load valve is given by

$$q_L = K_x x_L - K_p p_L$$

where $x_L$ is the valve spool displacement (m); $K_x$ and $K_p$ are flow gain ($m^2/s$) and flow-pressure coefficient ($m^3/(N\cdot s)$) of the valve, respectively; $p_L$ and $q_L$ denote the load pressure (N/m$^2$) and load flow ($m^3/s$), respectively.

The load-side hydraulic cylinder flow continuity equation is
\[ q_L = A_i \frac{dx_i}{dt} + V_L \frac{dp_L}{dt} + C_{ul} p_L \]  

(3)

where \( A_i \) is effective area of hydraulic cylinder piston \((m^2)\); \( C_{ul} \) is total leakage coefficient of hydraulic cylinder \((m^3/(N \cdot s))\); \( \beta_e \) is effective bulk modulus \((N/m^2)\); \( V_L \) is the total control volume of the loading system \((m^3)\).

The force equilibrium equation of the loading hydraulic cylinder is

\[ A_i p_L = m_L \frac{d^2x_i}{dt^2} + B_L \frac{dx_i}{dt} + F_L \]  

(4)

where \( m_L \) is equivalent mass of load \((kg)\); \( B_L \) represents load equivalent viscous damping coefficient \((N \cdot s/m)\).

Combining the above four equations and then solving them, the transfer function of load force \( F_L \) with the valve spool displacement \( x_v \) as input, and the displacement of booster \( x_M \) as disturbance is as follows

\[
F_L(s) = \frac{A_i K_q x_v(s)}{K_m} = \frac{V_i m_L s^3 + \left( m_L + \frac{V_i B_L}{K_s} \right) s + \left( B_L + \frac{A_i^2}{K_m} \right)}{4 \beta_s K_m K_s s^3 + \left( m_L + \frac{V_i B_L}{K_s} \right) s + \left( B_L + \frac{A_i^2}{K_m} \right)} s + 1 \]  

(5)

where \( K_m = K_e + C_d \).

From Eq. (5), besides the valve spool displacement \( x_v \), \( F_L \) is also related to booster velocity \( s x_M \). This component of \( F_L \) is exactly the surplus force in loading system.

Defining

\[
G_p(s) = \frac{A_i K_q}{K_m} \frac{V_i m_L}{4 \beta_s K_m K_s s^3 + \left( m_L + \frac{V_i B_L}{K_s} \right) s + \left( B_L + \frac{A_i^2}{K_m} \right)} s + 1 \]  

(6)

\[
G_s(s) = \frac{V_i m_L s^3 + \left( m_L + \frac{V_i B_L}{K_s} \right) s + \left( B_L + \frac{A_i^2}{K_m} \right)}{4 \beta_s K_m K_s s^3 + \left( m_L + \frac{V_i B_L}{K_s} \right) s + \left( B_L + \frac{A_i^2}{K_m} \right)} s + 1 \]  

(7)

\[
G_n(s) = \frac{x_v(s)}{I(s)} = \frac{K_m}{\omega_n^2 + \frac{2 \zeta_n}{\omega_n} s + 1} \]  

(8)

where \( K_m \) is the valve gain \((m/A)\); \( \omega_n \) and \( \zeta_n \) are the valve natural frequency \((rad/s)\) and the valve damping ratio, then equation (5) can be rewritten as

\[ F_L(s) = G_p(s) x_v(s) - G_s(s) s x_M(s) \]  

(9)

A simplified structure diagram of the loading system can be expressed in Figure 2. In Figure 2, \( u_r \) represents the current command input \((A)\), \( F_d \) represents the surplus force input \((N)\), \( F_d = G_d(s)x_M \).
It is known from Eq.(5), if only there exists a movement of the loaded plant, the surplus force is inevitable, and cause interference on the output loading force. The motion amplitude and frequency of booster are measurable and bounded. From the mathematical model, the magnitude and frequency of surplus force must be bounded. Surplus force is a known, measurable and strong output disturbance; it influences the force tracking accuracy, reduces the closed-loop bandwidth of the loading system, and seriously causes gradation in force control performance. Therefore, surplus force suppression is the primary problem in loading system, and should to be solved firstly.

Electro-hydraulic loading system is a nonlinear and strongly time-varying system because of variable parameters and external interferences. System robustness must be taken into consideration, otherwise might cause system instability and dissatisfy system performance requirements.

Two problems of surplus force suppression and force tracking can be solved separately, if controllers of loading system are designed based on double loops cascade composition control strategy. The inner loop controller is utilized to eliminate surplus force and other disturbances, enhance system robustness, and make the actual plant behave as a nominal model approximately; the nominal model based outer loop controller is utilized to realize the desired force tracking performance.

For surplus force suppression and system robustness, the inner loop is designed based on the disturbance observer (DOB) method.

3. LOADING SYSTEM DOB DESIGN

The disturbance observer (DOB) method firstly observes the output differences between output of actual plant and nominal model, which are caused by external disturbances and model perturbations, estimates the disturbances amount through the inverse nominal model, then introduces equivalent compensation into the control, feeds back the cancellation into the control input, and thus achieves complete inhibition of the equivalent disturbance.

The disturbance observer (DOB) structure of the studied electro-hydraulic loading system is depicted in Figure 3. $d$, $F_s$, $\hat{d}$ and $\zeta$ represent equivalent disturbance, surplus force, estimated disturbance and measurement noise, respectively. $P_s(s)$ and $Q(s)$ represent the nominal model of actual plant $P(s)$ and the low-pass filter.
3.1. Disturbance observer Analysis

The transfer functions from each input to output $F_i$, in Figure 3, are derived

\[ G_{u,F_i} = \frac{F_i}{u} = \frac{PP_u}{Q(P-P_u) + P_u} \]  
\[ G_{d,F_i} = \frac{F_i}{d} = \frac{PP_u(1-Q)}{Q(P-P_u) + P_u} \]  
\[ G_{\xi,F_i} = \frac{F_i}{\xi} = -\frac{PQ}{Q(P-P_u) + P_u} \]  
\[ G_{e,F_i} = \frac{F_i}{F_d} = -\frac{P_u(1-Q)}{Q(P-P_u) + P_u} \]  
\[ F_i = G_{u,F_i}u - G_{d,F_i}\xi + G_{d,F_i}d - G_{e,F_i}F_d \]  

From transfer functions $G_{d,F}$ and $G_{e,F}$, in order to suppress surplus force and disturbance, the maximum magnitude of $1 - Q(s)$ should be small for all frequencies. From transfer function $G_{\xi,F}$, in order to eliminate noise, the maximum magnitude of $Q(s)$ should be small for all frequencies. This is a conflicting requirement.

So design of $Q(s)$ is a trade-off between disturbance suppression and noise elimination.

It is noticed that there is a same multiplicative element $1 - Q(s)$ from $d$ and $F_d$ to the output $F_i$, that means after appropriate transformation surplus force can be regarded as a part of external disturbances. The surplus force frequency is less than 2Hz according to the actual situation, so the surplus force can be regarded as a low-frequency disturbance. When $Q(s)$ is designed, it must to be concerned that its cut-off frequency must be higher than the disturbance frequency of the surplus force, besides the constraints of the traditional DOB.

From Figure 3, the sensitivity function $S_{DOB}$ and the complementary sensitivity function $T_{DOB}$ of the inner loop are derived, respectively [14, 15]

\[ S_{DOB} = \frac{P_u(s)(1-Q(s))}{Q(s)(P(s) - P_u(s)) + P_u(s)} \]  
\[ T_{DOB} = \frac{P(s)Q(s)}{Q(s)(P(s) - P_u(s)) + P_u(s)} \]
The term $1 - Q(s)$ reflects the sensitivity characteristics of the DOB system. If the sensitivity function $\|1 - Q(s)\|_\infty$ and the complementary sensitivity function $\|Q(s)\|_\infty$ are designed small enough, the optimization performance of disturbance observer can be guaranteed. This is also a conflicting requirement. So design of $Q(s)$ is a trade-off between performance and robustness.

The key of the DOB design is the design of $Q(s)$. If $Q(s)$ is designed appropriately, the disturbance observer can not only suppress the surplus force, but also fulfill three fundamental objectives in feedback system design: disturbance rejection, robust stability and robust model matching between the actual plant and the nominal plant model.

### 3.2. Design Method of $Q(s)$

Under the structure of DOB in Figure 3, analysis and design of $Q(s)$ are not a simple matter when the performance requirements and robustness requirements must be considered together. If we take

$$Q(s) = \frac{P_n(s)K(s)}{1 + P_n(s)K(s)}$$

(16)

where $K(s)$ is stable, and then the DOB can be transformed equivalently to a structure in Figure 4 [16]. In this way, the design of $Q(s)$ can be transformed to the design of robust internal-loop compensator $K(s)$. Therefore, $Q(s)$ can be designed systematically under the framework of robust internal loop.

![Figure 4. Equivalent structure of disturbance observer.](image)

In Figure 4, $K(s)$ is the robust internal-loop feedback controller. If $K(s)$ is designed for the nominal model $P_n(s)$ in order to satisfy a given performance and robustness criterion, optimal $Q(s)$ of DOB is systematically designed which has the optimality under the given specific conditions, because the transfer function of the unity feedback system with $P_n(s)$ and $K(s)$ is $Q(s)$ [16].

The $H_\infty$ mixed sensitivity method is utilized to derive the optimal robust internal loop compensator $K(s)$, then the optimal $Q(s)$ is derived by Eq.(16).

From Eq.(16), the sensitivity function $S_Q$ and the complementary sensitivity function $T_Q$ of $Q(s)$ are obtained, respectively

$$S_Q = \frac{1}{1 + P_n(s)K(s)}$$

(17)

$$T_Q = \frac{P_n(s)K(s)}{1 + P_n(s)K(s)}$$

The $H_\infty$ mixed sensitivity problem can be described as

$$\min_{K_{\text{robust}}} \left\| W_S S_Q \right\|_\infty < 1$$

(18)
4. CONTROLLER DESIGN AND SIMULATION

4.1. Design of $Q(s)$

With parametric modeling the actual plant $P(s)$ is derived, and its order is five. The method of the leading pole is adopted in order reduction, the high resonant frequencies of the denominator are removed, the system order is reduced to two, and then a two order nominal model $P_n(s)$ is obtained as

$$P_n(s) = \frac{287670}{s^2 + 540s + 7500}$$

(19)

Figure 5. Bode diagram of actual plant and nominal model.

By the contrast of the magnitude and phase frequency characteristics of $P(s)$ and $P_n(s)$, as shown in Figure 5, the nominal model can well reflect the actual characteristics of the actual plant in system working frequency range (<540rad/s).

The weighting function $W_1(s)$ represents system uncertainties. It is determined by the unstructured model uncertainties, namely the high frequency un-modelled dynamics, and the uncertainties of model parameters. It reflects the inherent characteristics of the actual plant, and is obtained as

$$W_1(s) = 10^{-5} (s + 5) (s + 1.2)$$

(20)

The weighting function $W_3(s)$ represents the spectrum characteristics of the interference. Considering the characteristics of surplus force and equivalent interference, it is obtained as

$$W_3(s) = \frac{700}{(0.2s + 1)^2}$$

(21)

MATLAB Control System Toolbox and Robust Control Toolbox are utilized to solve the mixed sensitivity problem described in Eq. (18). Then $G(s)$ is obtained. After calculated by Eq. (16), $Q(s)$ is obtained as

$$Q(s) = \frac{94160s + 8638000}{s^3 + 4775s^2 + 94160s + 8638000}$$

(22)

The magnitude frequency response curves of $Q(s), W_3^{-1}(s), 1 - Q(s)$ and $W_3^{-1}(s)$ are shown in Figures 6 and 7, respectively. Obviously, the robust stability condition and the interference suppression requirement of DOB...
system are both satisfied. In order to compare with the previous traditional Q-filter, a classic binomial coefficient model filter with the same order and the same relative order is considered as

$$Q_{31}(s) = \frac{3\tau s + 1}{(\tau s)^3 + 3(\tau s)^2 + 3\tau s + 1}$$  \hspace{1cm} (23)$$

where \( \tau = 0.00564 \), so its high frequency characteristic coincides with \( Q(s) \).

\[ \text{Figure 6. Frequency magnitude responses of } Q(s), W_3^{-1}(s) \text{ and } Q_{31}(s). \]

In Figure 6, \( Q_{31}(s) \) also satisfies the robust stability condition of the system. In Figure 7, in low frequency range, the magnitude difference between \( 1 - Q_{31}(s) \) and \( 1 - Q(s) \) is about 4.2dB, this shows that the interference suppression performance of \( Q(s) \) is 1.7 times higher than \( Q_{31}(s) \), moreover \( Q_{31}(s) \) does not satisfy the interference suppression requirement.

From Figure 6, it is obvious that the maximum frequency bandwidth of \( Q(s) \) is directly related to weighting function \( W_j(s) \), namely the frequency bandwidth is determined by the robust stability conditions of the inner loop, the appropriate design of \( W_j(s) \) has actual significance.

\[ \text{Figure 7. Frequency magnitude responses of } 1 - Q(s), W_1^{-1}(s) \text{ and } 1 - Q_{31}(s). \]
For the same actual plant, once the nominal model is selected, the model errors are sure. The design of $W_s(s)$ depends on the model errors, so choosing a more approximated nominal model of the actual plant is better for improving the frequency bandwidth of $Q(s)$, and can achieve better interference suppression effect.

4.2. Simulation of Surplus Force Suppression

In order to verify the effectiveness in surplus force suppression of the designed inner loop controller based on DOB, simulations are performed in the model built according to Figures 2 and 3. In simulation process, some unified settings are made: 1) loading force control input $F$ is 0(N); 2) motion spectrum $x_{M1} = 7\sin(14\pi t)$ (mm/s), its amplitude is higher than the maximum amplitude at maximum frequency; 3) motion spectrum $x_{M2} = 15\sin(8\pi t)$ (mm/s), its frequency is higher than the maximum frequency at maximum amplitude; 4) the outer loop controller is constant 1.

In order to compare the effects of surplus force suppression, simulations with an often used velocity feed-forward compensation function $G_{com}(s)$, based on structure invariance principle, also have carried out. Surplus force without any compensation is represented by the fine dotted line in the figures.

Simulation results show that, surplus force is mostly eliminated with compensator $G_{com}$, but such compensation belongs to open loop compensation, the system robustness obviously cannot be guaranteed. With DOB, the results are more outstanding than those with $G_{com}$, and system robustness is also guaranteed.
5. CONCLUSIONS

To address the force tracking issue in the load simulator used to simulate the aerial load simulator in the helicopter manoeuvring system, a composition controller with double loops cascade composition control strategy was designed. Two main problems of the loading system, i.e., the surplus force suppression and the loading force tracking, had been solved separately using the controllers in inner loop and outer loop. The proposed disturbance observer based inner loop controller with the low pass filter \( Q(s) \) to be designed by \( H_\infty \) mixed sensitivity optimization method can acquire the satisfactory loop performance compared with those designed by traditional method. The surplus force is eliminated approximate completely (lower than 5%), and the system robustness is guaranteed at the same time. So, in the outer loop controller design, the plant can be treated as a nominal model with non-disturbance and non-perturbation, and which simplified the design of the controller of the outer loop.

ACKNOWLEDGEMENT

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REFERENCES


ADAPTIVE DAMPING CONTROL OF DRILLING STRING FOR OFFSHORE PLATFORM PASSIVE COMPENSATOR UNDER DIFFERENT SEA CONDITIONS

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ABSTRACT

The passive heave compensator is widely used in offshore drilling platform nowadays. The traditional passive compensation system is designed in one sea condition and the performance of it decreases when the sea waves become severe. This paper finds out a new idea to use the compensation system adaptively by changing the damping ratio between drilling string and drilling fluid. To achieve the goal, the influencing factors of the damping ratio have been studied, and the square relationship between damping ratio and velocity of the fluid has been obtained. Nonlinear and linear model have been built to analyse the performance of compensation system. And it has been proved that by changing the fluid velocity adaptively, damping control is a feasible method to improve the performance. By using the compensation ratio as the control target, the new model of the system works better than the traditional compensation system when sea condition changes.

KEYWORDS: passive compensator, damping control, compensation rate

1. INTRODUCTION

Offshore drilling platform plays an important role in energy exploitation. While the oil sources in shallow sea become rare, drilling platform has moved to deeper sea area. In this condition, the impact on it from the sea wave is very sharp. Most of the motions could be reduced by the combination of mooring constraints except heave motion. In this case, heave compensator has been widely used in platform to mitigate the influence of the sea wave.

Two forms of the compensator have been applied, passive compensator and active compensator. The system of passive compensator consists of a group of air banks and passive compensator cylinders, which can be regarded as air springs. The system of active compensator has an actuator, such as a hydraulic cylinder which is controlled by the controller and a motion detector. The main characteristic of active compensator is the environmental adaptation which means it can function well in rough sea conditions. However it also adds extra power source and the energy consumption is more than passive compensator. In this paper the passive compensator is the main object of study.

The structure of the offshore drilling platform with passive compensator will be introduced at first. The drilling system consists of 2 parts: the over-water structure and the under-water structure.
Crown block, compensator cylinders, wire rope, drilling fluid pump, air banks and the platform itself are the main parts over-water. The crown block is connected with the drilling string through the wire rope. And the force from the string is the load of the crown block. Two compensator cylinders provide support to the crown block and the string. The cylinder is linked to the air bank. The most common air banks applied in passive compensator are hydro pneumatic accumulators with gas cylinders, which can be regarded as a gas spring. The pump provides drilling fluid to the string and the fluid flows back to the tank after taking back the cuttings from bottom of the well.

Drilling string, drilling bit and marine riser are the main parts under-water. Marine riser plays an important role in the under-water part. It can insulate the string from sea water and provide a flow back channel for drilling fluid. The drilling fluid flows from the top of the string to the drilling bit. Across the holes on drilling bit, it can flow back to the top of the string and into the tank.

In previous studies, the characteristics of the environment and over-water structure, such as the compensator cylinder have been widely studied in both passive and active system. But the under-water character is also important to the compensation performance. The damping rate between the drilling string and the fluid has a significant influence when we analysis the performance. However, the calculation method of the damping ratio is unreliable as the result changes maximum 100 times from one method to another. And the influencing factors are also not the same in different methods.

The environment of the platform has been studied by Korde[1]. Irregular wave acting on the platform has been introduced and the power spectral density function is firstly used in drilling platform compensator study. Hatleskog[2] studied the nonlinear model of passive compensator and the influence of the load variation of the sea bed. In his study, the string is modeled by the lumped-mass method and the damping ratio calculation between the string and the fluid is given. However, the influencing factors in his study are not same as that of the others’. In another paper of Hatleskog[3], the passive cylinder is regarded as the air spring and the model is simplified a lot. The research findings of Do[4] is based on an active heave compensation system. A disturbance observer is designed to observe the elastic and damping force acting on the string. It is a new method to deal with the under-water structure, but it can only be applied in active compensator. Huang[5] built a new structure of the active compensator using an active compensation motor and a passive compensation hydraulic motor. The stiffness of the load has been introduced, but the damping ratio of the load is ignored. The cable damp is considered. And an active damper is applied to attenuate the wave of sea.

In Figure 1, part 1 is crown block; part 2 is compensator cylinder; part 3 is wire rope; part 4 is drilling fluid pump; part 5 is drilling bit; part 6 is marine riser; part 7 is drilling string; part 8 is air bank.

In passive compensation system, there are also some parameters that can be controlled. The number of working air banks can change the stiffness of the air spring, which will change the compensation performance. In addition, the drilling flow, which is used to cool down the drill bit and transfer the drilling cuttings, can be controlled by drilling pump. With some limits, these parameters can be designed to be adaptive and it is the main idea of this paper. According to the study in this paper, adaptive parameter control in passive system is an effective way and does not require extra power source.

This paper conceptualizes a new idea of passive compensator. The adaptive idea has been applied to drilling fluid pump control based on the relationship between damping ratio of the string and the velocity of the fluid. In 2.1, the environment of the system will be introduced, which contains the introduction of sea wave and the relationship between wave and sea condition. The input of different sea conditions to the system is the crux of adaptive design and analysis, which will be discussed in this part. In 2.2, damping ratio will be calculated. The model will be built in FLUENT and the influencing factors will be analysed in the model. The relationship between flow velocity of drilling fluid, which can be controlled by the drilling pump, and the damping ratio will be obtained. In 2.3 and 2.4, the model of nonlinear and linear will be introduced. The lumped-mass method is applied in modeling. And the compensator is modeled as an air spring in linear mode. Nonlinear model is more accurate but time consuming. So the linear model corrected by the nonlinear model is built to accelerate the calculation process. In 2.5, the simulation result and the comparison of the two models will be given. And the linear model is corrected to have the same performance with the nonlinear
one. In part 3, the idea of damping control will be introduced in detail including the program. The control target will be discussed at first in 3.2, and then we obtain the compensation performance of the system with damping control and without damping control, followed by the comparison of the two above mentioned compensation performances.

Figure 1. The structure of the offshore drilling platform with passive compensator.

2. MODELING AND CALCULATING

2.1. Irregular wave

In order to model the real environment which the platform works in, the characteristic of the sea wave should be introduced at first. As a compensation system, the essential task is to mitigate the influence of the sea wave acting on the platform. There are many researchers who studied sea wave, and in this part the basic theory of the sea wave is introduced.

The sea wave is a kind of random wave, which means it consists of series of sinusoidal waves. So it can be showed as a stationary random process. Sinusoidal waves of different frequency, amplitude and phase are used to describe the wave front of a selected point:

\[
A(t) = \sum_{i=0}^{\infty} A_i \sin(\omega_i t + \phi_i)
\]  

(1)

Where \( A(t) \) is the height of the selected wave front, \( A_i \) is the amplitude of each harmonic wave, \( \omega_i \) is the angular frequency of each wave, and \( \phi_i \) is the initial phase of each wave.

Besides the 3 basic parameters, significant wave height and average wave height are also widely used. Wave height is the height calculated from the peak to the trough of one independent wave. In 20mins of observation time, significant wave height \( A_{1/3} \) can be defined as the average of the top 1/3rd of the total wave heights. And the average height is calculated from all the wave heights, which is written as \( \bar{A} \).
Using the sum of harmonic waves makes it hard to describe the overall feature of a chosen sea wave. So power spectral density function is applied to describe overall feature. Power spectral density function, which is usually written as $S(\omega)$, is the distribution regularities of sea wave energy due to the frequency. The relationship between $S(\omega)$ and $A_i$ is shown in the following function:

$$A_i = \sqrt{2S(\omega_i) \Delta \omega_i}$$

(2)

Which means the height of the wave front can also be described in the form below:

$$A(t) = \sum_{i=0}^{\infty} \sqrt{2S(\omega_i) \Delta \omega_i} \sin(\omega_i t + \varphi_i)$$

(3)

The average height also has a relationship with $S(\omega)$:

$$\bar{A} = \sqrt{2\pi \sum_{i=0}^{\infty} S(\omega_i) \Delta \omega_i}$$

(4)

The accurate description of $S(\omega)$ is usually the empirical formula. One of the widely used formula is called the Pierson-Moscowitz function[7] which is written as the form below:

$$S(\omega) = \frac{0.78}{\omega^3} \exp[-\frac{3.12}{\omega^4 A_{1/3}^2}]$$

(5)

The parameters of sea wave are different when the sea condition changes. Table 1 shows the relationship between sea conditions and the wave heights.

<table>
<thead>
<tr>
<th>Sea condition level</th>
<th>Sea wind velocity (kn)</th>
<th>The average wave height (m)</th>
<th>The significant wave height (m)</th>
<th>Dominant frequency (Hz)</th>
<th>Frequency band (HZ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>1.1</td>
<td>1.8</td>
<td>0.15</td>
<td>0.167-1</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>1.6</td>
<td>2.6</td>
<td>0.124</td>
<td>0.143-1</td>
</tr>
<tr>
<td>2</td>
<td>14</td>
<td>2.3</td>
<td>3.6</td>
<td>0.105</td>
<td>0.128-0.667</td>
</tr>
<tr>
<td></td>
<td>16</td>
<td>2.9</td>
<td>4.7</td>
<td>0.092</td>
<td>0.114-0.5</td>
</tr>
<tr>
<td>3</td>
<td>18</td>
<td>3.7</td>
<td>5.9</td>
<td>0.082</td>
<td>0.1-0.4</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>4.6</td>
<td>7.3</td>
<td>0.074</td>
<td>0.09-0.333</td>
</tr>
<tr>
<td>4</td>
<td>22</td>
<td>5.5</td>
<td>8.8</td>
<td>0.067</td>
<td>0.082-0.294</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>6.6</td>
<td>10.5</td>
<td>0.062</td>
<td>0.074-0.27</td>
</tr>
<tr>
<td>5</td>
<td>26</td>
<td>7.7</td>
<td>12.3</td>
<td>0.057</td>
<td>0.069-0.25</td>
</tr>
<tr>
<td></td>
<td>28</td>
<td>8.9</td>
<td>14.3</td>
<td>0.053</td>
<td>0.065-0.222</td>
</tr>
<tr>
<td>6</td>
<td>30</td>
<td>10.3</td>
<td>16.4</td>
<td>0.049</td>
<td>0.059-0.213</td>
</tr>
<tr>
<td></td>
<td>32</td>
<td>11.6</td>
<td>18.6</td>
<td>0.046</td>
<td>0.057-0.2</td>
</tr>
</tbody>
</table>

Table 1. The relationship between sea conditions and the wave heights
If the sea condition is known, $A_{1/3}$ can be found in Table 1, which means the power spectral density function can be obtained from equ.(5).

2.2. Damping ratio

The system could be simplified as two spring damping systems, the drilling string and the passive compensator, which couple with each other. As we have introduced above, the structure of offshore platform passive compensator is simple. The main part of it is the passive compensator cylinder and the air bank. However, the modeling of the drilling string is complex and full of disputes, while the principle of model simplifying is quite different. Also, very little information is present in the literature, for the calculation of damping rate between the drilling string and the fluid. But damping rate is a basic parameter of spring damping system.

In this part, the simulation of the damping rate between string and drilling fluid is introduced. We use 2D model in FLUENT to simulate. Figure 2 illustrates the flow condition of deep water drilling.

![Diagram](image)

Figure 2. The flow condition of drilling fluid in drilling string.

Red arrows 1 refer to the inner fluid and blue arrows 2 refer to the external fluid, which have opposite directions. 3 is the drilling string which transfers slowly in 4 the marine riser.

The inner fluid is driven by the pump put in the crown, while the external fluid flows from the shaft bottom with the silt cut by the drilling bit. Thus the string is surrounded by the drilling fluid and this special flow condition makes the damping rate unique between them. It is the combined action of the flow in 2 opposite directions. In addition, the velocity of the 2 opposite flows is different but correlative due to the flow continuity equation, as showed in equ.(6).

$$v_{in} \frac{\pi d^2}{4} = v_{out} \frac{\pi (B^2 - D^2)}{4} + q_s$$

In this equation, $v_{in}$ is the velocity of the inner fluid which is always called the velocity; $v_{out}$ is the velocity of the external fluid and called as velocity in annular; $d$ is the inner diameter of the string, $D$ is the string external diameter and $B$ is the inner diameter of the marine riser. The last item of the equation on the right is the flow added by the silt and in this paper it will be ignored for simpler analysis. Thus the relationship between the velocity and the velocity in annular is obtained.

Using the parameters given by China Shipping Heavy-duty Machine Group Co. 704 Research Institute showed in Table 2, the characteristics of the damping rate can be simulated in FLUENT and given directly.
There are some fundamental assumptions that need to be introduced at first. The drilling fluid is the non-Newtonian fluid which has different rheological behavior compared with the Newtonian fluid. However, as the flow of this condition is stable and the velocity is high, the viscosity of Newtonian fluid and non-Newtonian fluid are nearly equal to each other[8]. So Newtonian fluid model can be used to study the characteristics of drilling fluid in this situation. In addition, the string is regarded as a circular tube for simplifying the simulation. However in reality the string translates slowly in the same direction with the flow velocity and spins to transfer the torsion from the motor of the crown to the bit.

Table 2. Parameters of the string and fluid

<table>
<thead>
<tr>
<th>Inner diameter of the marine riser (mm)</th>
<th>External diameter of the string (mm)</th>
<th>Inner diameter of the string (mm)</th>
<th>Length of a single string(mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>160</td>
<td>150</td>
<td>130</td>
<td>10000</td>
</tr>
</tbody>
</table>

Density of the inner fluid (kg/m3) | Viscosity of the inner fluid (kg/ms) | Density of the external fluid (kg/m3) | Viscosity of the external fluid (kg/ms) |
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1200</td>
<td>8.00E-03</td>
<td>1440</td>
<td>1.50E-02</td>
</tr>
</tbody>
</table>

According to the equation given by the other researchers [2][9], the damping ratio is influenced by the following factors: the length of the string L, the flow velocity \( v_{in} \), the inner diameter d, the external diameter D and the parameters of the drilling fluid such as density, viscosity. If the fluid is given and regarded as Newtonian fluid, the parameters of it will be stable. Thus the influence of the fluid itself is ignored.

In this situation, 2D model is applied. The finite element model has been built in Gambit as shown in Figure 3. The length of Line AI indicates the unit length of drilling string L. The length of line DE indicates the inner diameter of the string d. The length of line CF indicates the external diameter of the string D. And the length of line BG indicates the inner diameter of marine riser.

The boundary type of face ABJI, face FGON, face CDLK and face ABJI are wall. Face LM, face BC and face FG are velocity inlet. Face DE, face JK and face NO are outflow. The zones of DEML, BCKJ and FGON are fluid and have been meshed. Zone CDLK and EFNM are drilling string. Zone DEML is the inner drilling fluid. Zone BCKJ and FGON are the external drilling fluid. Zone ABJI and GHPO are the marine riser. Red lines refer to inlet face. Blue lines refer to outlet face.

Figure 3. 2D model in FLUENT.
First of all, models have been defined in FLUENT. The Multiphase model has been applied and the number of phases is 2. The RNG k-epsilon model has also been applied. Then two different materials have been defined with the parameters in Table 2. Two phases have also been defined. The Gravity has been put into the model and the direction is IA. The initial velocity of fluid in zone DEML $v_{in}$ and the initial velocity of fluid in zone BCKJ, FGON $v_{out}$ have been given. And the relationship between the two velocity has been given in equ.(6). The PISO algorithm has been applied to solve the calculation. After initializing, the phase of zone DEML is the basic phase, and the phase of zone BCKJ, FGON is the second phase. Finally, the force monitors have been applied. The drag coefficient on line DL and CK have been monitored, which are written as $C_{in}$ and $C_{out}$.

After iterating and simulating, $C_{in}$ and $C_{out}$ can be obtained. The 3D damping ratio $C$ can be calculated by equ.(7).

$$C = \pi dC_{in} - \pi dC_{out}$$  \hspace{1cm} (7)

Change L from 500mm, 1000mm to 2000mm to build three models in FLUENT and keep the other parameters the same. And also ensure that the sizing grid, multiphase model, viscous model and arithmetic in FLUENT are the same. The influence of L is illustrated in Figure 4 which shows the linear dependence.

Change $v_{in}$ from 0m/s to 4m/s in one model. The length of the string in model is 1000mm. Keep other parameters and forms of the model as the same. The influence of the fluid velocity is illustrated in Figure 4 also. Keep the other parameter constant and change inner diameter from 65mm, 97mm, 130mm to 140mm with external diameter 150mm. Also change external diameter from 141mm, 146mm, 150mm to 156mm with inner diameter 130mm. The results are illustrated in Figure 4.

Figure 4. The simulation result in FLUENT of the relationship between damping ratio and other parameters.

The black line refers to the relationship between damping ratio and length of the string. The red line refers to the relationship between damping ratio and flow velocity. The blue line refers to the relationship between damping ratio and inner diameter. The green line refers to the relationship between damping ratio and external diameter.
In the real condition, if drilling has started, the parameters $D$, $d$ and $L$ cannot be changed, which means velocity control is the only method to change the damping ratio adaptively. And velocity could be controlled easily by the revolving speed adjustment of the pump.

The damping ratio has been obtained and showed in Figure 5. Black line is the simulation result, and red line is the fitted curve. The damping ratio is proportional to the square of $v_n$.

$$C = 260v_n^2L$$  \hspace{1cm} (8)

![Figure 5. The square relationship between velocity of the drilling fluid and damping ratio.](image)

2.3. Non-linear Model

The model of the passive heaving compensation system could be built after the study of the damping ratio. There are 2 methods to build the model. One is the non-linear model built in AMESim which will show the characteristics of the non-linear elements, such as the air bank and the friction in the cylinders. But the simulation time is consuming, especially for the random sea wave input. Also the control strategy is hard to get from the simulation. Thus the linear model has been built which ignores the non-linear factors to simplify and expedite the simulation process. The non-linear model is built at first. Then the linear model is built and the correction factor is given to reflect the non-linear influence.

The parameters of the passive compensator, which are applied to the two models, have been given by 704 Research Institute and is shown in Table 3.

<table>
<thead>
<tr>
<th>Inflation pressure of the air bank (MPa)</th>
<th>Piston diameter of hydro pneumatic accumulator (mm)</th>
<th>Length of hydro pneumatic accumulator (mm)</th>
<th>Piston diameter of passive compensator cylinder (mm)</th>
<th>Length of passive compensator cylinder (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15.3</td>
<td>500</td>
<td>6000</td>
<td>420</td>
<td>7620</td>
</tr>
<tr>
<td>Mass of crown block (kg)</td>
<td>Mass of travelling block (kg)</td>
<td>Length of drilling string (m)</td>
<td>Mass of drilling string (kg)</td>
<td>Elasticity modulus of drilling string (GPa)</td>
</tr>
<tr>
<td>15000</td>
<td>15000</td>
<td>13000</td>
<td>441000</td>
<td>210</td>
</tr>
</tbody>
</table>

The modeling methods of the drilling string are lumped-mass method and consistent-mass method. While the consistent-mass method is quite hard to achieve on AMESim, we choose the lumped-mass method. The mass of system will be lumped in one point, so the calculating time can be shorter.
The principle is to ensure the same dynamic performance before and after modeling, which means the unit step response, inherent frequency, magnitude-frequency characteristics and phase-frequency characteristics will be the same. In some situations, the static load is put on the system to maintain the same static characteristics.

As to the string, three values should be analysed: the lumped-mass, the stiffness and the equivalent damping ratio. The ratios of the values after simplifying should be given and then the static load can be calculated from the ratio.

When the drilling begins, the drilling bit is fastened with the well bottom. While the weight of the drill string and the pressure on the drill bit given by the seabed are high, the bottom of the string is considered fixedly connected with the well bottom. In this condition, the model in Figure 6 can be used.

![Figure 6. The lumped-mass model](image)

The lumped-mass model is illustrated on the left in Figure 6. Part m refers to the lumped-mass, k refers to the stiffness of the string and C refers to the equivalent damping ratio. m’ is the static load to balance the system.

From the parameters given in Table 3, the stiffness of the drilling string can be calculated as follows:

\[
k_0 = \frac{E \pi (D^2 - d^2)}{4L} = 64860 \text{ N/m}
\]  

And the damping ratio is obtained by the simulation in FLUENT. Using the following parameters to simulate: L=13000m, d=130mm, D=150mm, B=160mm, \(v_m=2\text{ m/s}\).

The damping ratio is :

\[
C_0 = 4227300 \text{ Ns/m}
\]

According to Rayleigh’s method, any object with continuous mass can be simplified to a lumped-mass model. The model consists of two parts: the dynamic model and the statistics load. The dynamic model has been built according to the energetic principles to show the dynamic load caused by the string, while the statistic load has been put as the initial condition.

The principle of mass, statistic load and stiffness is as follow:

\[
m = \frac{m_0}{3} \\
m' = \frac{2m_0}{3} \\
k = k_0
\]
The principle of the equivalent damping ratio has not been given in Rayleigh’s method, so the ratio after simplification should be analyzed.

Two different models have been built in AMESim to find out the ratio of damping. As the consistent-mass cannot be built in AMEsim, the model can be simplified by the following method. Put N mass points and add a spring between each two mass points. Each mass point \( m_0 \) is \( m/N \), and the stiffness of spring is \( Nk \). In addition, the coefficient \( C_0 \) is \( C/N \) for the friction of damping is equally distributed along the length. If N is big enough, the model can be regarded as a consistent-mass string.

The model is illustrated in Figure 6 on the right. The long mass points model is the consistent-mass spring model and the short mass points is the lumped-mass spring model. The same force input is added on the two models and the dynamic characteristics of both model will be compared and discussed. The consistent-mass model contains 100 mass points and each point is 3950kg. In addition, the stiffness of the springs which connect each of the two mass points is \( 6.49 \times 10^6 \text{N/m} \) and the coefficient \( C_0 \) is 4227300 \text{Ns/m} . The mass point of the lumped-mass model is 131666kg, \( m/3 \) as we discussed, and the stiffness is 64860N/m. 4 different coefficients of viscous friction, \( C_1, C_2, C_3 \) and \( C_4 \), have been put on the lump-mass model, and the values are as follow:

\[
C_1=C_0=4227300\text{Ns/m}, \quad C_2=C_0/2=2113650\text{Ns/m}, \quad C_3=C_0/3=1309100\text{Ns/m}, \quad C_4=C_0/4=1056825 \text{Ns/m}.
\]

First of all, the unit step input is put on both string models. The overshoot and adjust time of one step input have been obtained. In addition, change the input to a periodic force and get the inherent frequency from Bode diagram. This data is shown in the Table 4.

<table>
<thead>
<tr>
<th>Model</th>
<th>Lumped-mass model with ( C_1 )</th>
<th>Lumped-mass model with ( C_2 )</th>
<th>Lumped-mass model with ( C_3 )</th>
<th>Lumped-mass model with ( C_4 )</th>
<th>Consistent-mass model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overshoot of one step input (%)</td>
<td>20</td>
<td>50</td>
<td>60</td>
<td>70</td>
<td>60</td>
</tr>
<tr>
<td>Adjust time of one step input (s)</td>
<td>6</td>
<td>12</td>
<td>18</td>
<td>19</td>
<td>17</td>
</tr>
<tr>
<td>Inherent frequency (Hz)</td>
<td>0.065</td>
<td>0.053</td>
<td>0.052</td>
<td>0.05</td>
<td>0.056</td>
</tr>
</tbody>
</table>

It can been found that the characteristics of \( C_3 \) model is similar with the consistent-mass\(^{\prime}\). So \( C \) of the AMESim model is:

\[
C = \frac{C_0}{3} = 1409100 \text{Ns/m} \tag{11}
\]

The conclusion is that the lumped-mass and damping ratio become 3 times less than the real one while the model stiffness stays the same after using lumped-mass method. And the static load is 2/3 as that of the real mass. This conclusion can be used in both non-linear model and linear model.

Figure 7 illustrates the non-linear model in AMESim by using the Mechanical module, Hydraulic Component Design module and Pneumatic Component Design module. Part 1 is gas cylinder; part 2 is sea wave input; part 3 is drilling string; part 4 is hydro pneumatic accumulator; part 5 is passive compensator cylinder; part 6 is statistic load; part 7 is crown block.
2.4. Linear model

To build the linear model, the air bank should be linearized and the friction in the passive compensator cylinder should be ignored. Then the differential equations to describe the system should be established and the transfer function of the system could be obtained.

A linear air bank could be regarded as a spring, which is often called gas spring. Gas spring has the same stiffness characteristic as spring. The deflation of the air bank is an adiabatic process which could be described as equ.(12).

\[ p_g V_g^n = p_{g0} V_{g0}^n \]  

where \( p_g \) is the system pressure, \( V_g \) is the volume of the air bank, \( p_{g0} \) is the inflation pressure, and \( V_{g0} \) is the initial volume of the bank. The parameter \( n \) is constant 1.4. If the flow is stable as the compensator system, equ.(12) could be linearized. Equ.(13) describes the linear flow function:

\[ q_g = \frac{V_{g0}}{p_{g0} n} \frac{dp_g}{dt} \]  

where \( q_g \) is the gas flow from the air bank.

The forces put on the piston of the hydro pneumatic accumulator reach a balance as the equ.(14) shows:

\[ dF_g = dp_g S \]  

where \( S \) is the area of the piston.

The definition of flow is:

\[ q_g = S v_g \]  

where \( v_g \) is the velocity of the piston.

From equ.(13), equ.(14) and equ.(15), the stiffness \( K_g \) of the gas spring is given:
\[ K_s = \frac{dF_g}{dx_g} = \frac{p_{g0}n}{V_{g0}} S^2 \]  

(16)

While \( p_{g0} = 15.3 \text{MPa}, \ V_{g0} = 20000 \text{L}, \ n = 1.4, \ S = 0.19 \text{m}^2 \), we have:

\[ K = 38663.1 \text{N/m}. \]

Then we could analyze the system including the string modeled in 2.3.

According to the Newton second law, the forces on the crown block can be described as follow:

\[ m_s \frac{d^2 x_g}{dt^2} = F_g - F_k - F_c \]  

(17)

where \( F_g \) is the counter-acting force from the piston of hydro pneumatic accumulator, \( F_k \) is the elastic force from the string and \( F_c \) is the damping force from the string. \( m_s \) is the sum of the crown block mass, traveling block mass and the equivalent mass of the string. The forces could be described as follow:

\[ F_g = k_g (x_c - x_g) \]

\[ F_k = k x_g \]

\[ F_c = C \frac{dx_g}{dt} \]  

(18)

where \( x_c \) is the input sea wave acting on the platform.

Using Laplace transform to integrate and simplify the differential equations, the transfer function of the sea wave to the displacement of the crown block can be obtained:

\[ G(s) = \frac{X_g}{X_c} = \frac{\eta k_g}{m_s s^2 + C s + k_g + k} \]  

(19)

where \( \eta \) is the non-linear corrected parameter.

And equ.(19) is the linear model of passive compensator of the offshore drilling platform. The simulation could be performed in MATLAB easily.

2.5. Simulation

The input signal should be given at first. The wave analysis studied in 2.1 shows the relationship between sea condition and wave height. In the non-linear simulation, 3 typical sea waves under the wind speed of 10kn, 20kn and 30kn are the input signal to the compensation system. While in the linear simulation, the whole 16 sea wave heights are used as the input to show the compensation performance under all the sea conditions for a shorter simulation time cost. The compensation rate \( \Phi_1 \) is the comparison of the average amplitude of the crown block and half of the average height. The compensation performance decreases if \( \Phi_1 \) is growing. Sometimes we use \( \Phi_2 \) to measure the performance and it is more convenient to use. If \( \Phi_2 \) grows, the performance also increases. The relationship between \( \Phi_1 \) and \( \Phi_2 \) is easy to find in equ.(20):

\[ \Phi_1 = \frac{2 \overline{x}_g}{A} \]

\[ \Phi_2 = 1 - \frac{2 \overline{x}_g}{A} \]  

(20)

The irregular input model in AMESim is the Harmonics Signal Source Module which consists of a series of sinusoidal wave. That means that the power spectral density function should be discretized at first. Divide
the frequency band into 25 parts, and then use the equ.(5) to get \( S(f_i) \) in every discrete frequency point. After using equ.(2) to get the amplitude of each sinusoidal wave, the expression of the irregular input under one sea condition could be obtained. In addition, the initial phase of each sinusoidal wave is the stochastic production. The sea wave parameters of each sinusoidal wave under wind of 30 kn is as an example showed in Table 5.

<table>
<thead>
<tr>
<th>( f_i ) (Hz)</th>
<th>0.036</th>
<th>0.042</th>
<th>0.047</th>
<th>0.052</th>
<th>0.057</th>
<th>0.062</th>
<th>0.068</th>
<th>0.073</th>
<th>0.078</th>
<th>0.083</th>
<th>0.089</th>
<th>0.094</th>
<th>0.099</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A_i ) (m)</td>
<td>0.20</td>
<td>0.70</td>
<td>1.21</td>
<td>1.50</td>
<td>1.58</td>
<td>1.40</td>
<td>1.26</td>
<td>1.12</td>
<td>0.99</td>
<td>0.88</td>
<td>0.78</td>
<td>0.69</td>
<td></td>
</tr>
</tbody>
</table>

Table 5. Discretized sinusoidal wave of 30 kn wind speed

These parameters are added into the AMESim Harmonics Signal Source Module to model the input irregular wave. And the displacement of crown block could be gotten from the model directly. Calculating the average amplitude, the compensation rate in non-linear model will be obtained.

The irregular input model in the linear model is much simple. Enlarge the frequency band from 0.01Hz to 10Hz and average it into 200 parts. The calculation of amplitude of each sinusoidal wave is simple in MATLAB. The displacement of the crown \( x_g \) should be calculated also.

From the transfer function \( G(s) \) showed in equ.(19), the amplitude-frequency characteristic could be obtained. It is the modulus of \( G(j\omega) \). Thus the influence of each sinusoidal wave in discrete frequency point acting on the compensation system is shown in equ.(21):

\[
x_{gi}(t) = |G(\omega_i)|A_i \sin(\omega_i t + \varphi_i)
\]

As to the average displacement, there is need of some mathematical techniques. We define \( S_g(\omega) \) as the power spectral density function of the \( x_g \) as showed in equ.(22).

\[
S_g(\omega_i) = \frac{2\pi x_{gi}^2}{\Delta \omega_i}
\]

And the average of \( x_g \) could be gotten from equ.(23):

\[
\bar{x}_g = \sqrt{\pi \sum_{i=0}^{\infty} |G(\omega_i)|A_i^2}
\]

And then the compensation rate in linear model will be gotten.

Put the sea waves of 10kn, 20kn and 30kn wind speed on both non-linear and linear model, and keep the other parameters the same. In addition, add the non-linear corrected parameter \( \eta=2.5 \) into linear model. Changing the damping ratio from 2.25e6Ns/m, 4.22e6 Ns/m to 6.15e6Ns/m, the influences of sea conditions and damping ratio could be gotten from the 3D contour maps in Figure 8.

The result of nonlinear model is on the left and the right one is the result of linear model. The horizontal axis is the damping ratio changing from 2.5×10^6Ns/m to 6×10^6Ns/m. The vertical axis is the significant wave height \( A_{1/3} \) changing from 2m to 16m. The red part in the diagram means bad compensation performance and the blue part means the good performance as we use \( \Phi_1 \) here.
From *Figure 8*, three conclusions could be obtained:

(i) The compensator with one group of parameters can not suffer from different sea conditions. The compensation rate reduces sharply when the wave height increase and the maximal situation is 2.83 times when the wind speed changes from 10kn to 30kn. So the traditional passive compensation will easily fail to work if the environment is changed.

(ii) Increasing the damping ratio is a feasible method to remit the influence of the bad sea conditions. Keeping sea condition the same, changing the damping ratio can increase the compensation rate by a maximum of 3.17 times;

(iii) The simulation results of linear and non-linear model are basically the same, especially in the moderate environment. So the linear model can be used to analyse the compensation rate under the whole sea conditions in order to study the damping control strategy.

3. ADAPTED CONTROL

3.1. The principle of control

According to the simulation results, sea conditions influence the performance of the compensator sharply. And increasing the damping ratio between string and drilling fluid is a feasible method to mitigate the displacement of crown block and the string, which has been studied in 2.5. There are 4 factors influencing the damping we studied, which are the length of the string, the velocity of the flow, the inner and external diameter of the string. However, the length and diameters are not feasible to change, which means the velocity control is the only method of improving the flow of the drilling fluid pump. The pump is a plunger pump and the friction pairs are easy to wear in high rotation speed. So the high damping ratio conditions could not be kept the same in all the sea conditions.

In this paper, the compensation rate and the displacement of the crown are regarded as the most important indexes. So keeping the rate or the displacement stable is the control target. Compensation rate $\Phi_2$ is shown in equ.(20) and displacement of crown is shown in equ.(23).

In the following part, two of the control target will be analysed.

3.2. The control strategy

First of all, we compare the control target and feed the program of each target. The program is given in *Figure 9*. The program on the left is for the control target of crown block displacement. The program on the
right is for the control target of compensation rate $\Phi_2$. If the sea condition is changed, the wave height which is the input of the compensator system will also be changed. For a given system, $G(s)$ could be obtained. The given input parameters are $A_{1/3}$, control target (the displacement of the crown block or the compensation rate), precision $e$ and the disperse unit $\Delta \omega$. If the absolute difference of the target value and the calculated value is smaller than the precision, the current $v$ is allowed. If not, increase or reduce $v$ to give a new damping ratio and system transfer function.

Keeping $\Phi_2=85\%$ and putting other parameters into the program on the left in Figure 9, the velocity can be obtained. Also keeping $x_g=0.17m$, the velocity can be given from the right program. The result of two control targets is illustrated in Figure 10. The input variable on the horizontal axis is $A_{1/3}$ which refers to the sea condition. The black line is the velocity from keeping compensation rate stable. And the red line is from keeping displacement of the crown block stable.

The controlled velocity of 0.17m displacement changes quicker and sharper than the velocity of 85% compensation rate. Also the velocity of 0.17m displacement breaks through the limit even in 3m/s, which will break the friction pairs. Thus the suitable control target is the compensation rate.

The compensation ratio of given velocity 0.897m/s and controlled velocity is illustrated in Figure 11. And the displacement of the crown block of the 2 situation is given in Figure 12. Without damping ratio control through changing the velocity of the fluid, the passive compensator can not adjust to the different sea conditions in one system parameter. Thus, changing the fluid velocity is a method to be adaptive to the environmental changes.

*Figure 9. The programs flow chart to calculate the suitable velocity of drilling fluid.*
The velocity of fluid (m/s)
The significant wave height (m)
85% compensation rate
0.17m displacement

Figure 10. The velocity calculated from two control targets.

Figure 11. The compensation rates from adaptive controlled system and uncontrolled system.

Figure 12. The displacements of crown block from adaptive controlled system and uncontrolled system.
4. CONCLUSION

In this paper, a new control idea for offshore platform passive compensator has been discussed. The base is the influencing factors study of damping ratio between string and drilling fluid. It has been proved that the length, external diameter, inner diameter of the string and the fluid velocity affect the damping ratio. However, only velocity control is feasible. Using the square relationship between velocity of the drilling fluid and damping ratio between string and fluid, the compensation system works adaptively when the sea condition changes. Moreover, the control strategy and program has been given in this paper by using the new idea. The compensation rate of new system can almost stay the same, while the compensation rate of traditional system drops sharply when the significant wave height increases.

There is also another idea to make the system adaptive which needs to be discussed. The volume of the air bank can also be changed. The method of adaptive damp control can be applied in volume control as well in future works. And the model of the damping ratio in FLUENT is a simple model which ignores many factors. 3D model is needed to build in the following study and the dynamic model can be added in new model. Finally, the damping ratio study is essential and experimental study is a good support to model simulation.

REFERENCES

HARDWARE-IN-THE-LOOP ELECTRONIC CONTROL SYSTEM FOR A UNIVERSAL TEST RIG FOR HYDRAULIC SERVO CYLINDERS MOTION SYNCHRONIZATION AND SERVO VALVES TESTING

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ABSTRACT

This paper presents the details of a universal hydraulic test rig, with an eye on the rig electronic control system. It has been developed for testing the motion synchronization of two loaded, unconnected, and independently controlled hydraulic servo cylinders, and for simulating and evaluating the performance of these cylinders when cross coupling controllers are introduced. The rig design allows also testing servo valves of different sizes and brands as well as testing the performance of two servo cylinders at the same time. Results of testing the motion synchronization of two servo cylinders are presented. Measurements for the no load flow, blocked load pressure, and blocked load internal leakage at different input signals for malfunctioning and repaired servo valves of different sizes and brands are also presented. The rig has been used as an off-line simulator to verify the quality of repair and the assurance of coincidence of the hydraulic and electronic null positions of the repaired servo valves. The presented results confirm the fidelity and fertility of the rig.

KEYWORDS: Hydraulic, servo, cylinder, valve, electronic control, synchronization

1. INTRODUCTION

Hydraulic servo cylinders and servo valves of different sizes and different brands are extensively used nowadays in modern heavy industries. In thin slab casting plants, for instance, these valves are used, e.g. in furnaces, oscillating moulds, and rolling mill stands. Two servo cylinders, each driven independently in a closed loop control system incorporating a hydraulic servo valve, are widely used to oscillate the heavy moulds of continuous casting machines [1]. In this case the required accurate position synchronization of the two cylinders during mould oscillation can be realized only when the two command signals of the control loops are identical and the system components are functioning properly. A Fuzzy Logic Controller (FLC) had been proposed in [2] as a Cross Coupling Controller (CCC) to enhance the motion synchronization in this application in the event of malfunctioning of any of the components. Theoretical analysis and practical implementations showed the merits of this CCC [2]. A Proportional Controller (PC) had also been proposed in [3] as a CCC, but the theoretical analysis revealed that the FLC renders better synchronization.
Unexpected failure or malfunctioning of servo valves or cylinders would cause production sudden interruption. This would have an adverse effect on the production line and the plant economics. Testing of operating cylinders and valves during the plant overall maintenance periods would help avoiding their unexpected sudden failures. Due to variety of sizes and brands, the frequent testing of these components can’t be easily realized in many cases, since the available test rigs are generally designed to test components of one and in some cases two manufacturers.

A test rig for investigating the motion synchronization of two non-connected hydraulic servo cylinders, when the cylinders are loaded, is presented in this paper. The rig software section allows testing the motion synchronization when each cylinder is independently controlled, or when cross coupling controllers are also used. Servo valves and cylinders of various sizes from different manufacturers can be tested using the rig [4]. Two different servo cylinders can be tested under load at the same time on the rig. For investigating the performance of hydraulic cylinders, a test rig had been developed by Forester [5]. On Forester’s rig a double rod servo cylinder driven by a two stage servo valve was used to apply the load on the tested cylinder. The closed loop force control loop of the loading cylinder of the rig had a simple proportional controller.

2. TEST RIG DESCRIPTION

The hydraulic circuit diagram of the test rig is shown in figure 1 [3]. The figure shows the hardware section which includes the hydraulic power supply unit for the valves test bench and the tested cylinders. The geometric volume of the pump of this unit is 250 cm³/rev. Both of the pump flow rate and maximum supply pressure can be controlled electrically in closed loops [6]. The pump delivery line is equipped with two accumulators, each of 32 liter volume, and feeds the valves test bench and the servo valves driving the cylinders to be tested. A Leakage Unit (LU) is seen in the figure, in the tank of which the oil leaking during exchanging the tested valves and cylinders is collected after being filtered. A small gear pump automatically supplies the collected oil to the main tank when the oil level in the LU tank reaches a certain value. The servo valves to be tested are mounted on a specially designed main block. On this main block valves or sub-plates for valves of sizes up to NG 32 from different manufacturers can be mounted, using intermediate adapter plates. On the other hand, each servo cylinder to be tested can be loaded by means of a double rod servo cylinder of piston and piston rod diameters 100 mm and 60 mm respectively, and stroke 100 mm. Each loading servo cylinder is equipped with a pressure sensor, and is driven by means of a two stage electro-hydraulic servo valve in a closed loop force control system. The two servo valves of the loading cylinders are supplied with oil from the hydraulic power supply unit shown in figure 1, which has a pressure and flow compensated axial piston pump of geometric volume 100 cm³/rev [6].

Figure 2 shows the mechanical parts and a schematic drawing for the rig when it is used for testing the motion synchronization of two servo cylinders; namely cylinders (1) and (2). The hardware plane of the rig includes the tested cylinders, two loading servo cylinders (3) and (4), four servo valves that control the cylinders, and transducers (11) and (12). Servo valves (5) and (6) control the motion of the tested cylinders in closed loop position control systems. Tested cylinders are connected to loading cylinders via connecting blocks (20) which reciprocate over linear ball bearings. The mass of each block is 300 kg and can be varied according to test requirements. The force of each loading cylinder is controlled by means of a servo valve (7) or (8) type 4WSE2DE1D [7] in a closed loop force control system and can follow any required pattern. Load cells (10) are used to allow automatic shutdown of the rig if excessive unsafe loads are generated. Simulation of internal leakages in the cylinders or their driving servo valves can be carried out using throttle valves interconnected between the relevant hydraulic lines in parallel with the cylinders or the valves.

The software plane which includes the electric and electronic control part of the test rig has the following major components: a PLC Simatic S7-300 for processing and control of the digital and analog I/O signals, a power supply module for input 220 VAC / output 24 VDC – 10A, a power supply module for input 220 VAC / output ± 15 VDC - 4A, a three phase autotransformer with input 380 VAC / output 220VAC – 52A, an amplifier card type VT-VSPA2-50 for the supply of command signals till 2A, and a Siemens touch screen flat panel type
MP277 as a human machine interface. The test rig pumps are rotated by means of switches on the touch screen. The main pump delivery pressure and flow rate are remotely controlled via two potentiometers feeding the input of the electronic control card type VT-DFPE controlling the pump pilot valve.

Position control of the tested cylinders is carried out utilizing an interface electronic control card (IEC1). The IEC1 is a dSPACE controller board type DS1104, endowed with eight A/D and eight D/A interface [8]. It is plugged to a PC server and used as an interface module between the software plane, which includes a Simulink control program for the closed loop control of the tested cylinders positions, and the hardware plane which includes these cylinders. The actual position of each cylinder is measured via a position transducer (11). The transducer output signal is converted to ± 10 V, processed to the signal block diagram of Matlab Simulink environment through the analogue digital converter (A/D) of the IEC1, and then compared with the reference signal \( x_r \). The resulting error is multiplied by a gain \( K_{po} \), converted to analogue signal via a digital analogue converter (D/A) and processed to the servo valve. Simulation of a servo valve disturbance in the form of a disturbed input signal or delay in response can be simulated by multiplying the input current to the servo valve by a constant or a delay factor in the software plane.

Figure 3 shows the components and the signal processing loop of the closed loop position control of the cylinder under test, while figure 4 shows the signal block diagram of the system during motion synchronization testing.

The load pressure of each loading cylinder; is measured via a pressure transducer (12) type EDS344-3-250, [9]. The resulting signal is converted from 4-20 mA to ± 10 V, and then processed to the signal block diagram in Matlab-Simulink environment through the A/D of the interface electronic control card (IEC2) type DS1104. It is then compared with the reference signal \( F_r \) and multiplied by a proportional controller gain \( K_p \), converted through the D/A and processed to the corresponding servo valve. Figure 5 shows the components and the signal processing loop of the closed loop force control of the loading cylinders, while figure 6 shows the signal block diagram of this system. The signal processing components have been assembled and interconnected in the control panel as shown in figure 7.
Figure 1. Hydraulic circuit diagram of the test rig.
Figure 2. Test rig mechanical parts and the HiLcontrol for motion synchronization testing.
Figure 3. Closed loop position control signal processing.

Figure 4. Signal block diagram of the position control system during motion synchronization testing.
Figure 5. Closed loop force control system.

Figure 6. Signal block diagram of loading cylinders force control system.
3. RESULTS OF TESTS

Results of the tests carried out using the rig for motion synchronization of two symmetrical servo cylinders, and for testing different servo valves are presented in what follows.

3.1 Servo cylinders motion synchronization

The synchronization of motion of the two servo cylinders used for oscillating the mould of the continuous casting machine of the thin slab casting plant at AlEzz AlDekheila Alexandria Steel Co. (EZDK) was tested using the rig when these cylinders showed unsatisfactory synchronized motion during operation. They are double-rod cylinders with piston and piston rod diameters 160 mm and 100 mm respectively, and 100 mm stroke. The displacement of each cylinder is controlled in the plant and on the rig via a three stage servo valve type 4WSE3EE1615/200B8T315 [7], and through an independent closed loop control system. The control loop in the test rig has been set identical to that used in the plant, [1] and [3]. Each tested cylinder (TC) was set initially to its fully retracted position, with each loading cylinder (LC) set at its fully extended position. Each TC was then driven in a position control mode to its mid stroke position. Each LC was moved simultaneously to its middle position, with the pin connections of the DiA of the IEC1 and of the ND of the IEC2 coupled electrically for achieving the synchronized position control mode. In this middle position the control system of each LC had been turned to closed loop force control mode, with the reference force $F_r$ set to a constant value equal to 45 kN via Matlab-Simulink environment to represent the mould weight. The TCs were then made to oscillate in a closed loop position control mode according to the reference value $x_r = 3.5 \sin 10 \pi t$ mm. This input signal is the most prevailing signal of cylinders oscillation in the plant. During experiments the force resulting from each loading cylinder was measured. Figure 8 shows the recorded forces of the loading cylinders. The forces of the two cylinders are seen to be nearly the same with an average value of 43.5 kN and oscillations of about 4 kN amplitude and 5 Hz frequency.
The recorded displacements of the two cylinders during the test are shown in figure 9. Position synchronization errors are evident in this case, with the maximum value of the error amounting to 1 mm. According to [1], at this value of synchronization error the machine in the plant is automatically stopped.

![Figure 9](image)

**Figure 9.** Displacements of the two cylinders taken from the plant.

These cylinders were tested further using the rig to investigate their internal leakage characteristics. Figure 10 shows the results of this test. Cylinder (B) was found to experience excessive internal leakage compared to cylinder (A). Cylinder (A) was consequently recommended to be further used, and cylinder (B) to be changed.

![Figure 10](image)

**Figure 10.** Cylinder internal leakage variation with supply pressure.

Cylinder (B) had been replaced by a new one, and the motion synchronization investigation had been carried out again. The obtained results are shown in figure 11. Practically acceptable motion synchronization of the cylinders is evident. These cylinders showed satisfactory performance when they were used later in the plant.

![Figure 11](image)

**Figure 11.** Displacements of two tested cylinders, one replaced.
Experiments had also been carried out to evaluate the merits of the CCCs proposed in [2] and [3] on motion synchronization of two cylinders, when one of the two independent closed loop drive and control systems suffers from disturbances. Two identical cylinders and valves had been installed on the rig, and one cylinder and its control valve were subjected to disturbances. A throttle valve had been inserted between the two lines leading from the servo valve to its driven cylinder to simulate a cylinder or valve internal leakage, and a delay of 5 ms had been introduced to the input signal of this servo valve. The throttle valve was opened to allow a 36 l/min flow rate at a pressure drop 10 MPa across it. The system was operated in the absence of a CCC, and when a CCC with FLC or PC is applied. The displacements of the two cylinders and the synchronization errors were recorded in the three cases. The resulting errors are shown in figure 12. It is seen that the maximum synchronization error amounting to about 1 mm in the absence of CCC is reduced to about 0.75 mm when a CCC with PC is applied and further down to about 0.6 mm when FLC is used. The proposed CCCs are seen experimentally, using the HIL simulation technique, to reduce the practically unacceptable synchronization error of 1 mm to acceptable values, which complies with the theoretical findings in [3].

![Graph showing synchronization errors in the presence and absence of CCC](image)

**Figure 12.** Synchronization errors in the presence and absence of CCC

### 3.2 Testing of servo valves

The valves to be tested are to be mounted on the main block at the valve test bench using adaptor plates that match valves of different types and sizes from different manufacturers. Valves of sizes up to NG 32 from manufacturers like Bosch Rexroth, MOOG, EMG, Vickers, Kawasaki, Yuken and others can be tested using this stand.

In the following, examples of test results for three electro-hydraulic servo valves (A, B and C) are presented. The valves are from three different manufacturers, where the valve A is of size NG 16 and the valves B and C are of size NG 10. The presented test results for each valve show the variation of the no-load flow (NLF) and the blocked load internal leakage (BLIL) with the increase of the valve input signal [7] and [8]. The valve supply pressure had been set to 7±0.2 MPa during the NLF tests of the valves, to 14±0.2 MPa during testing the BLIL of valves A and B, and to 10±0.2 MPa during testing the BLIL of valve C. During the tests, the valve input signal had been increased from zero to 100% of the nominal value. Each of these valves was tested as it is coming from the plant as a malfunctioning valve, and after repairing it. Figures 13 to 18 show the obtained results. The dashed lines in the figures represent the valves performance before repair. The shown values are out of the acceptable range stated in the manufactures technical data catalogues and test sheets. The valves had been repaired and tested, and the test results are shown in solid lines in the figures, which depict satisfactory valves performance.
The rig has also been used as an off-line simulator to verify the repair quality of the servo valves in an environment similar to the practical one. Repaired and tested servo valves were installed on new cylinders for checking their functionality. In very few cases a deviation between the hydraulic and electronic null positions
of the valves was recorded. This deviation was eliminated in most cases through loop gains adjustment, and rarely by changing the valve pilot stage nozzles or the valve main spool position transducer; i.e. the LVDT.

4. CONCLUSION

The details of a hardware-in-the loop electronic control system of a test rig developed for testing motion synchronization of two hydraulic servo cylinders and for testing different types of servo valves have been presented. The rig can also be used for testing the performance of two hydraulic cylinders at the same time. Loading of cylinders during the tests is carried out by means of hydraulic servo cylinders, which allows applying different loading patterns during the tests. Motion synchronization of two cylinders, each driven by a servo valve in an independent closed loop control system, can be tested in the presence or absence of a cross coupling controller. The rig allows simulation of the system practical disturbances either in the hardware or software planes. Results of testing the motion synchronization and testing servo valves of different sizes from different manufacturers confirm the high reliability of the test rig. Tests carried out, using the rig, on repaired servo valves under conditions similar to those in the plant were conducted to assure the repair quality and probable needed final off-line adjustment of these valves. These final fine tuning or in rare cases internal part replacement can be made before installing the valve in the machine. With such a rig, considerable positive impact on operational and maintenance costs of hydraulic servo systems can be realized.

ACKNOWLEDGEMENT

The authors would like to express their gratitude to the management of EZDK Co. for their support and for funding the development and manufacture of the test rig.

REFERENCES

DESIGN OF A VANE PUMP POWER SPLIT TRANSMISSION FOR A HIGHWAY VEHICLE

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ABSTRACT

The demand for low emissions and better fuel economy requires increased vehicle drive system efficiency. A Vane Pump Power Split Transmission (VPPST) can fulfill this need. A VPPST functions much like a conventional power split hydro mechanical transmission (HMT) but without the planetary gear. It consists of a Vane Power Split Unit (VPSU) and a variable displacement hydraulic motor. The VPSU is a double acting vane pump with a floating ring. The Input shaft of the VPSU is directly coupled to the engine and the output shaft, fixed to the floating ring, is connected to the drive train. The control flow of the VPSU is fed to a variable-displacement motor mounted on the VPSU output shaft. Unlike an HMT, where the transmission ratio can be adjusted by changing the displacement of the pump or motor, or both, in a VPPST, the transmission ratio is adjusted by controlling the displacement of the motor thus affecting the pressure of the control port of the VPSU. The resulting infinitely variable transmission allows for optimum engine operation by decoupling the engine speed from the drive speed. When the control flow is zero, the input and output shafts lock up at the same rotational speed. The transmission also has an integral clutch that allows de-clutching the engine from the drive train by retracting the vanes of the VPSU. In this paper, the design of a VPPST sized for a Class 1 pickup truck is demonstrated.

KEYWORDS: Vane pump, power split, drivetrain, transmission.

1. INTRODUCTION

The growing demand of fuel efficient vehicles, low carbon footprint technologies and drivability requires more efficient vehicle drives. This creates opportunities to integrate new technologies to achieve better performance with energy efficiency.

The automatic transmission is a widely acceptable solution. However it is unable to maintain optimal efficiency over the entire engine operating range. In contrast, a Continuously Variable Transmission (CVT) can decouple the engine speed from the wheel speed, making the engine run more efficiently. The hydraulic form of a CVT is a hydrostatic transmission that uses a hydraulic pump to drive a hydraulic motor. By changing the displacement of one of units, a variable transmission ratio is created. Due to its high power density, durability, continuously variable ratio and smooth operation, the Hydro Static Transmission (HST) has been widely used in off-road applications such as agricultural, construction and forestry machinery.
With continuously variable transmission and energy storage, full engine management becomes possible. The high power density of the hydraulic powertrain allows for lower vehicle weight, more regenerative braking and faster acceleration. The EPA’s world first series hydraulic hybrid delivery vehicle has 60-70% better fuel economy and 40% or more reduction in CO₂ emissions [1]. Altair’s series hydraulic hybrid city bus delivers 30% or more fuel efficiency than other diesel-electric buses available today [2].

The inefficiency of pumps and motors at low displacement dramatically reduces the transmission efficiency. However by combining an HST and a gearbox in one unit, the hydro-mechanical transmission has the advantages of both, the continuously transmission ratio of an HST and the high efficiency of a gearbox. Since the power is transferred though both the mechanical path and hydraulic path, it is also called a power-split transmission.

Although an HMT is more efficient than an HST, the transmission is more bulky due to the planetary gear set. Therefore a more compact and cost-effective power split transmission is needed. The VPPST consisting of a VPSU and a variable displacement hydraulic motor, is an attractive alternative. The architecture is shown in figure 1.

2. VANE PUMP POWER-SPLIT UNIT

The VPSU is based on a double-acting vane pump. Since it has a balanced design, it has smaller bearings, a longer lifetime and quieter operation than a gear or piston pump. The rotor with the vanes is coupled to the input shaft and the floating ring is coupled to the output shaft. As it is based on a double-acting vane pump, the VPSU intakes and discharges oil symmetrically from two pumping chambers.

The VPSU combines both the pumping and motoring function in one single unit, making it function like a conventional HST, but it is more compact. The pumping unit, consisting of the input shaft, rotor/vane assembly and the floating ring, is the same as a conventional vane pump except for the floating ring. The
motoring unit consists of the floating ring and the output shaft. The schematic of the VPSU and its hydraulic symbol are shown in figure 2 and details of the VPSU is shown in figure 3. Unlike a conventional HST where the transmission ratio is adjusted by changing the displacement of the pump, the motor, or both, the ratio of the VPSU is adjusted by changing the pressure at the control port. By changing the control pressure, the output shaft torque and speed of the VPSU can be adjusted.

The control flow at the control port $Q_c$, is determined by the relative rotary speed between the input and output shaft

$$ Q_c = (\omega_p - \omega_m)D_p\eta_v $$

where $\omega_p$ and $\omega_m$ are the input and output shaft speeds, $D_p$ is the displacement of the VPSU, $\eta_v = f(p_c, \omega_p - \omega_m)$ is the volumetric efficiency of the transmission, which is pressure and relative speed dependent, and $p_c$ is the control pressure at the control port. In the HMT, the control pressure is regulated by the displacement of the hydraulic motor.

The power equation for the VPSU is

$$ T_p\omega_p = T_m\omega_m + p_cQ_c $$

The mechanical and volumetric efficiencies of the unit can be found by analyzing the equivalent unit shown on figure 2. The details are given in [3]. The characteristics of VPSU will be determined in upcoming experiments. Experimental results on fan drives give some indication of the potential for energy saving. These experiments show that the input power with the VPSU is nearly 20% lower than with a conventional vane pump given the same output fan speed and torque.

The vane can be fully retracted through the tapered pins actuated by pilot pressure, decoupling the output shaft from input shaft and making the transmission function as clutch. The parasitic loss in the clutch is mainly the viscous drag on the rotor and is very low. The integrated clutch in the transmission is an attractive feature.

By increasing the control pressure to a high level, the control flow becomes zero and the input and output shafts have same rotational speeds. This lock up function creates an efficient direct drive for high speed.

### 3. POWER REQUIREMENT

The power requirement is determined by the driving resistance [4]. The driving resistance is made up of wheel resistance ($F_R$), air drag resistance ($F_D$), gradient resistance ($F_G$) and acceleration resistance ($F_A$). So the total tractive effort ($TE$) is sum of all resistances

$$ TE = F_R + F_D + F_G + F_A $$
where $K_1$ is the rolling resistance coefficient, $M$ is the weight of the vehicle, $\theta$ is the gradient angle, $\rho$ is the air density, $C_w$ is the drag coefficient, $V$ is the vehicle speed, $A$ is the frontal area, $a$ is the acceleration and $g$ is the acceleration of gravity.

The performance of vehicle is analyzed in four distinct modes:

- **Cruising Speed**: Flat road and zero acceleration
- **Acceleration Performance**: Flat road and zero initial velocity
- **Gradient Performance**: Zero acceleration
- **Towing performance**: Additional weight

The power required for each mode is analyzed for a Class 1 pickup truck. The parameters of pickup truck are given in table 1.

<table>
<thead>
<tr>
<th>Table 1. Class 1 pickup truck parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gross Vehicle Weight</td>
</tr>
<tr>
<td>Gross Combined Weight</td>
</tr>
<tr>
<td>Diameter of Wheel</td>
</tr>
<tr>
<td>Frontal Area (A)</td>
</tr>
<tr>
<td>Rolling resistance Coefficient ($K_1$)</td>
</tr>
<tr>
<td>Drag Coefficient ($C_w$)</td>
</tr>
<tr>
<td>Air Density ($\rho$)</td>
</tr>
</tbody>
</table>

The torque required at different cruising speed with towing and without towing is shown in figure 4. The maximum cruising speed of a pickup truck without towing is 160 Km/hr (100 mph), so it needs 1134 Nm of torque.

A pickup truck without towing must accelerate from 0 to 60 mph (96.5 Km/hr) in 5.9 to 7.5 sec, requiring an acceleration of 4.54-3.57 m/s². The torque required to achieve an acceleration of 4.54 m/s² is plotted in figure 5. The maximum torque required is 5785 Nm.

The maximum gradient of a highway is in the range of 3 to 5%. However, in a mountainous region the gradient is higher. Figures 6 and 7 show the torque required to climb at different gradients and speeds with and without towing.

![Cruising Speed](image_url)

*Figure 4. Torque required for cruising with and without towing*
Figure 5. Torque required for 4.54 m/s² acceleration

Figure 6. Torque required for different gradients without towing

Figure 7. Torque required for different gradients with towing
From the power analysis, it can be observed that, the maximum torque is required during acceleration and climbing a high gradient road. On a high gradient road, pick-up trucks climb at low speed. So, to achieve all of the performance required, the torque at the wheel of the pickup truck should be greater than 5785 Nm. and the maximum speed of vehicle should be 160 Km/hr.

The transmission matches the engine torque to the wheel torque. The specification of the engine selected for the analysis is given below.

Power: 225.2 kW @ 6500 rpm (302 hp @ 6500 rpm)
Torque: 377 Nm @ 4000 rpm (278 lb-ft @ 4000 rpm)

4. POWER SPLIT ARCHITECTURE

The key component in this design is VPSU which splits power into a hydraulic path to run a variable displacement motor and a mechanical path to rotate the output shaft of the VPSU. The power sharing between mechanical path and hydraulic path is controlled by the hydraulic motor displacement [5][6]. The hydraulic power is then added to the drive shaft to amplify the torque. The schematic of power split transmission is shown in figure 8.

![Figure 8. Schematic of power split transmission](image)

The input shaft of VPSU is coupled to an IC engine and output shaft is connected to the drive axle through appropriate gear box. The gear ratio is used to obtained desired wheel speed and wheel torque from the engine. The position of the gearbox and selection of appropriate gear ratio is very important in sizing of pump and motor. Different architecture with gear boxes at different positions are noted below.

4.1. ARCHITECTURE 1

The output shaft of VPSU is connected to an intermediate gear box, having gear ratio G2. The hydraulic path of VPSU is fed to a variable displacement motor which is mounted on the output shaft of an intermediate gear box. The motor has the same speed as the output shaft of the intermediate gear box .It also amplifies the torque from $T_{12}$ to $T_2$. The power of transmission is transferred to the wheel through a final drive gear ratio G1. The speed and torque relationship between the engine and wheel is derived from static conditions, as given by eq. (9) and eq. (12) and the fraction of mechanical power, hydraulic power fraction and loss fraction is formulated in eq. (15), (17) and (18). The schematic is shown in figure 9, where $\omega_e, \omega_2$ and $\omega_w$ are the engine, transmission output shaft, and wheel shaft speeds. $T_e, T_2$ and $T_w$ are the engine, transmission output and wheel shaft torque. $D_p$ and $D_m$ are the displacement of the VPSU and variable motor, $p$ is the pressure in the hydraulic path, and $Q_1, Q_2$ and $Q_m$ are the flows as shown figure 9.

The VPSU is a single unit performing both pumping and motoring. For ease of understanding the equivalent model shown in figure 2. The overall efficiency of the pumping unit ($\eta_p$) and the motoring unit ($\eta_{pm}$) is assumed to be 0.9. Hence, the overall efficiency of VPSU is 0.81. For calculation, the volumetric efficiency
of the pumping unit ($\eta_{p,v}$), the volumetric efficiency of the motoring unit ($\eta_{pm,v}$), the mechanical efficiency of the pumping unit ($\eta_{p,m}$), and the mechanical efficiency of the motoring unit ($\eta_{pm,m}$) is taken as the square root of the overall efficiency of each unit. The overall efficiency of the variable displacement motor ($\eta_m$) is assumed to be 0.85 and all gear boxes are assumed to be lossless. The efficiency of each unit needs to be modified based on the future experimental observations.

![Figure 9. Schematic of Architecture 1](image)

At the final drive, with gear ratio G1

$$\omega_2 = \omega_w \times G1 \quad (4)$$

$$T_w = T_2 \times G1 \quad (5)$$

Similarly at the Intermediate gear, with gear ratio G2

$$\omega_1 = \omega_2 \times G2 \quad (6)$$

$$T_{12} = T_1 \times G2 \quad (7)$$

The control flow $Q_m$ is given by

$$Q_m = Q_1 - Q_2$$

where flow is defined as the product of displacement of the pump or motor and the speed of rotation.

$$Q_m = \frac{D_m \omega_2}{\eta_{m,v}} = \frac{D_p \omega_e \eta_{p,v} - \frac{D_p \omega_1}{\eta_{pm,v}}}{\eta_{pm,v}} \quad (8)$$

From eq. (4), (6) and (8)

$$\omega_e = \left( \frac{D_m}{D_p \eta_{m,v}} + \frac{G2}{\eta_{pm,v}} \right) \frac{G1 \times \omega_w}{\eta_{p,v}} \quad (9)$$

The torque $T_2$ is the sum of the VPSU output shaft torque and the torque created by the hydraulic motor. The torque in a hydraulic component is the product of pressure and displacement. The hydraulic pressure $p$ can be found from

$$p = \frac{T_e \eta_{p,m}}{D_p} \quad (10)$$

The resulting torque $T_2$ is

$$T_2 = T_{12} + \eta_{m,m} p D_m \quad (11)$$

From eq. (5), (7), (10) and (11)
\[ T_w = G1\eta_{p,m}(G2\eta_{pm,m} + \frac{D_m}{D_p}\eta_{m,m})T_e \] (12)

The mechanical fraction \( f_m \) is the ratio of the power of the VPSU output shaft to the engine power.

\[ f_m = \frac{T_1\omega_1}{T_e\omega_e} \] (13)

From eq. (4), (5), (6), (7), (9) and (12):

\[ T_1\omega_1 = T_e\omega_e G1G2\eta_{p,m}\eta_{pm,m} = T_e\omega_e \left( \frac{G2\eta_{p,m}\eta_{pm,m}\eta_{p,v}}{\frac{D_m}{D_p}\eta_{m,v} + \frac{G2}{\eta_{pm,v}}} \right) \] (14)

From eq. (13) and (14):

\[ f_m = \frac{G2\eta_{p,m}\eta_{pm,m}\eta_{p,v}}{\frac{D_m}{D_p}\eta_{m,v} + \frac{G2}{\eta_{pm,v}}} \] (15)

And hydraulic fraction \( f_h \) is the ratio of hydraulic power of the VPSU to the engine power

\[ f_h = \frac{PQ_m}{T_e\omega_e} \] (16)

From eq. (8), (9), (10) and (16):

\[ f_h = \eta_{p,m}\eta_{p,v}(1 - \frac{G2}{\eta_{pm,v}\left(\frac{D_m}{D_p}\eta_{m,v} + \frac{G2}{\eta_{pm,v}}\right)}) \] (17)

And loss fraction is

\[ f_l = 1 - (f_m + f_h) \] (18)

4.2. ARCHITECTURE 2

In this architecture, the motor is coupled to the VPSU output shaft through a gear box of gear ratio G3. The schematic is shown in figure 10. The speed relationship of the engine and wheel shaft is given by eq. (19) and the torque relationship by eq. (20). The mechanical fraction is given by eq. (21).
4.3. SIZING OF TRANSMISSION

The engine operates in the range of 600 to 6500 rpm and generates a maximum torque of 377 Nm. During acceleration, the wheel rotates from zero to 1120 rpm at 160 Km/hr and needs a maximum torque of 5785 Nm. The maximum torque occurs in the lowest gear and the maximum speed occurs at the highest gear. So the maximum gear ratio will be

\[ G_{\text{max}} = \frac{T_{w,\text{max}}}{T_{e,\text{max}}} = \frac{5785}{377} = 15.34 \]

To exceed this requirement, \( G_{\text{max}} \) was chosen to be 15.5, by which 5843.5 Nm of torque will be available at the wheel assuming maximum engine torque.

By choosing the optimum fuel efficiency point from the engine map, the minimum gear ratio \( (G_{\text{min}}) \) required to achieve 160 Kmh/hr is calculated to be 2.57.

The input shaft of the VPSU is coupled to the engine. So the size of the VPSU is calculated from the maximum torque capacity of engine as given by eq. (10). With \( T_{e,\text{max}} = 377 Nm \), the displacement of the pump at different pressures is plotted in figure 11.

From the figure it can be noted that, if we select a maximum operating pressure of 250, 300 or 350bar then the displacement of motor would be 80.54, 67.11 or 57.53 cc/rev.

The parameters for sizing of the motor and intermediate gear ratio for both designs are shown in table 2.
### Table 2. Sizing parameters

<table>
<thead>
<tr>
<th>Sizing Parameter</th>
<th>Architecture 1</th>
<th>Architecture 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max Speed(Dm=0)</td>
<td>( G1G2 = 2.57 \times \eta_{pm,m} \eta_{p,v} )</td>
<td>( G1 = 2.57 \times \eta_{pm,m} \eta_{p,v} )</td>
</tr>
<tr>
<td>Maximum Torque</td>
<td>( G1 \eta_{p,m} (G2 \eta_{pm,m} + \frac{D_m}{D_p} \eta_{m,m}) = 15.5 )</td>
<td>( \left( \frac{D_m G_3}{D_p} \eta_{m,m} + \eta_{pm,m} \right) G1 \eta_{p,m} = 15.5 )</td>
</tr>
<tr>
<td>Maximum motor speed</td>
<td>1120*G1</td>
<td>1120<em>G1</em>G3</td>
</tr>
<tr>
<td>(@1120 RPM)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

At high speed, the vehicle needs less torque. So, all the engine power can be directed to the mechanical path with zero power in the hydraulic path. This can be achieved by setting the displacement of the motor to zero resulting in the minimum gear ratio, or direct drive. With the minimum gear ratio, the ratio of final drive \( G_1 \) is calculated. In design 2, \( G_1 \) is constant. However, in design 1, the final drive ratio \( G_1 \) depends on \( G_2 \), too.

At maximum torque, most of the power must go through the hydraulic path. The maximum torque amplification will be at maximum displacement of the motor. The ratio of \( D_m \) is calculated from maximum gear ratio (15.5) by using the equation shown in the second row of table 2.

The maximum speed of the motor is also a constraint on the design. The hydraulic variable displacement motor can operate at a maximum speed of 2500 to 5000 rpm. In figure 12 and 13, the size of the motor with maximum rotational speed and intermediate gear ratio for architecture one and two is plotted.

**Figure. 12. Size of motor and intermediate gear for architecture 1**

**Figure. 13. Size of motor and intermediate gear for architecture 2**
5. CONCLUSIONS

The VPPST has better advantage over other power split architectures. The key component of the VPPST is the VPSU, is a double acting vane pump. The VPSU has the advantages of durability, balanced design, smaller bearings and quiet operation. The unit is compact, having pumping and motoring together. It has an integrated clutch function, too. The efficient direct drive mode can be used when vehicle needs more speed less torque, as cruising on a highway. Through the hydraulic power split architecture, the engine can be operated at its most efficient point to consume less fuel.

In architecture one, the motor must have a through shaft, so it must be a radial piston or axial piston motor. It cannot be a bent axis motor. However in architecture two, the motor is connected to the driveshaft through a gear box so, there is greater flexibility to choose from different motor types.

One drawback of architecture one and two is a large motor is required. For example, for a 2926 kg pickup truck having additional towing capacity of 2290 kg with a 225 kW engine requires a motor of 220 cc/rev and max speed of 4500 rpm for a 350 bar pressure. This issue can be resolved in two ways. One way is to replace single motor with two smaller motors, one fixed and the other variable displacement. The second way is to put a two or three stage fixed gear box between the VPPST and the final drive.

In this paper, a static analysis has been used to size components. However the design needs to be tested under dynamic conditions. A dynamic model of the drivetrain is under development. The engine performance for different drive cycles will be analyzed under dynamic conditions. A method to control swash plate angle and engine management to minimize fuel consumption will be developed in the future.

REFERENCES

MODELING AND SIMULATION OF THERMAL HYDRAULIC COUPLING IN ELECTRO-HYDROSTATIC MODULES INVOLVING FIXED-DISPLACEMENT VANE PUMPS

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ABSTRACT

The thermal-hydraulic model of an electro-hydrostatic module involving a variable speed, fixed-displacement, internal-drain vane pump is developed at system level for analysis and for support of preliminary sizing. A generic model structure is proposed for the pump energy losses that are made dependent on velocity, pressure and temperature. Leakage and friction models are implemented accordingly. Model structure and implementation are proposed giving preference to re-using standard submodels of the AMESim software library and to balance model complexity and realism. Then, the proposed approach is deployed for simulating the temperature rise during static injection in moulding machines that are powered and controlled via an electro-hydrostatic module. In this particular phase, the pump operates at extremely low delivery flow, inducing a rapid temperature rise that may directly impact service life. For validation, a very penalizing approach is used which runs the model in open loop in response to motor electromagnetic torque, ambient temperature and hydraulic load resistance. Conclusions are drawn concerning the model accuracy and the mains issues encountered, regarding model structure, knowledge models and parameters preliminary determination.

KEYWORDS: Electro-hydrostatic, injection moulding, static injection, thermal-hydraulic, vane pump, AMESim

1. INTRODUCTION

Injection machines involve many actuators that typically operate in sequence according to the injection process, as displayed on Figure 1. In the well-established designs, the injection moulding machine is made of servo-hydraulic actuators (SHAs) that are hydraulically supplied at constant pressure from a centralised hydraulic power generation. SHAs are electrically signalled though metering valves that play the role of signal-to-power interfaces. Although this solution offers real advantages, it mainly suffers from the bad efficiency of valve control over a full mission (energy is consumed for full force, pressure in excess is wasted as pressure drop at valves).
Figure 1. REP injection moulding machine [1] and example of typical cycle’s phases for rubber parts

Toward "more electrical" products, electrical drives have been regarded with attention over the last decade for application in injection moulding machines [2]. Indeed, electric drives have made significant progress in terms of availability, cost, force or power density, power management and energy saving. Such drives typically combine an inverter and a DC brushless motor that operates functionally under the principle of power-on-demand (the consumed energy functionally corresponds to that demanded by the load). In a full hydraulic-less design, any hydraulic component is removed with resort to mechanical power transmission devices like gears or nut-screws, making electro-mechanical actuators (EMAs). However, in-service experience showed that maintenance costs in EMAs can be much higher than in SHAs: replacing rolling elements (nut-screws, bearings, etc.) in heavy duty machines is much more expensive than replacing hydraulic seals.

An interesting approach consists in combining the low maintenance cost of SHAs and the low energy consumption of EMAs to make electro-hydrostatic actuators (EHAs) where a centralized, fixed-displacement pump is driven at variable-speed by the electric motor. End-user power transformation is made by hydraulic cylinders where mechanical motion is required on the machine. This concept can be implemented technologically by use of gear pumps, pistons pumps or vane pumps.

Vane pumps have been preferred for their appreciated high efficiency, extremely low ripple and low noise emission. The integrated electro-hydrostatic module (EHM) under study associates a Parvex NX860 series motor and a Parker T7 series vane pump, Figure 2.

As the pump is not drained, one most critical phase of moulding for EHM design is the static injection where high pressure has to be hold (from seconds for plastic, to minutes for rubber) in the injection ram that remains quite steady. In this situation, the pump operates at very low outlet flow (e.g. 1 l/mn for 60 l/mn rated flow) that only evacuates a minor fraction of the energy generated by the pump internal energy losses (friction and leakage). As pump temperature rises, the internal leakage increases, and again contributes to

<table>
<thead>
<tr>
<th>Cycle’s Phase</th>
<th>Time duration [s]</th>
<th>Pressure [bar]</th>
<th>Flow rate [l/min]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mould closing/locking</td>
<td>3</td>
<td>210</td>
<td>68</td>
</tr>
<tr>
<td>Clamping of the unit</td>
<td>4</td>
<td>250</td>
<td>34</td>
</tr>
<tr>
<td>Dynamical injection</td>
<td>35</td>
<td>250</td>
<td>68</td>
</tr>
<tr>
<td>Static injection</td>
<td>60</td>
<td>250</td>
<td>12</td>
</tr>
<tr>
<td>Vulcanisation</td>
<td>105</td>
<td>130</td>
<td>34</td>
</tr>
<tr>
<td>Mould reopening</td>
<td>2</td>
<td>105</td>
<td>68</td>
</tr>
<tr>
<td>Part ejection</td>
<td>2</td>
<td>210</td>
<td>34</td>
</tr>
</tbody>
</table>
increase the pump temperature. Therefore, during the static injection phase, the pump temperature may reach excessive values that directly impact its service life.

For this reason, it is of particular importance to manage properly the operation of the pump during the injection phase, e.g. by adequate definition of the pump efficiency at extremely low speed or by increasing the outlet flow with resort to an external hydraulic bypass.

On the way to full simulation-based design, developing a realistic thermal-hydraulic model of the EHM at a system level can significantly contribute to shorten preliminary studies in response to injection machine designers' call for tender. Although the idea of using an EHM for injection moulding machine is not new, the published research work dealing with this technology essentially focuses on velocity control (e.g. [3]). In parallel, with the rise of more electrical commercial aircrafts, thermal aspects have been continuously addressed since the early 2000s for flight control actuators, e.g. [4]. In these applications, the pumps are drained and of axial-piston type and the thermal balance is addressed for long time operation (a flight mission); the scope of the thermal hydraulic simulation is far from the need related to static injection in moulding machines. On its side, the present work addresses the development of such a model that enables the thermal divergence to be investigated and the production rate and energy consumption of the moulding machine to be quantified. Section 2 is dedicated to the modelling and simulation of the pump energy losses while section 3 deals with the full hydro-thermal model of the EHM.

2. MODELLING THE ENERGY VANE PUMP LOSSES

The study carried out by the authors in [5], touched on the importance of reproducing by simulation the internal leakages and frictional torques of vane pumps when the EHM’s power sizing and temperature management are taken into account. These parasitic effects, initially assumed as linear in the analytical preliminary models, were then simulated through look-up tables composed by experimental data. A parametric model was also proposed, based on first degree polynomial equations. The frictional torques and the internal leakages were, at that modelling level, considered to depend on hydraulic line pressure drop and on the pump speed only. In practice, they also depend on dynamic fluid viscosity that is highly impacted by the fluid temperature. Such effects have been modelled in [6] from measurements performed on vane pumps and of internal gear pumps where the oil temperature was measured at pump outlet. However these measurements were considered not sufficient for the present need. It was shown in Parker laboratory that the oil temperature inside pump can be far different from temperature measured at the pump outlet. This is also confirmed in [7] in particular through the temperature measurements taken of eight thermocouples placed in the cam-ring and in the side-plate of the pump.

Figure 3 displays the measured internal leakages and frictional torques for a T7 Series Parker vane pump as a function of line pressure drop, pump speed and fluid temperature inside the pump. Two thermocouples were used and placed respectively in the drain pump volume and into the pressure plate (high pressure side) and they revealed a temperature gap lower than 1 °C.

It is clearly observed that the internal leakage is nearly independent from the pump speed while it varies linearly with the line pressure drop, at a given fluid temperature. At constant pressure and velocity, it is confirmed that the internal leakage varies in the same direction as the fluid temperature.

On its side, the frictional torque characteristic is more complex. It is necessary to distinguish two different regions of influence from the operating conditions. In the first region characterized by line pressure drops up to 100 bar, the frictional torques appear approximately independent from pressure with a low dependency from fluid temperature and exhibit a nearly monotonous increase with speed. At higher pressures, friction becomes more sensitive to pressure. When the fluid temperature increases, the frictional torques reduce as a consequence of lower viscosity but this effect is not observable at higher pressures where normal load at contacts is increased.
2.1. General architecture of the energy losses model

The structure of the system-level model of the vane pump with energy losses model is firstly addressed with support of a causal Bond Graph, Figure 4, following the approach proposed in [8].

The functional power flow is pointed out using bold power bonds. Friction is modelled as R dissipative effect. Introducing the RS field enables friction to be made sensitive to temperature and power loss to be output as a power source for the thermal model. Then, pressure influence is introduced using the modulated RS field (MRS) and is taken from an effort detector. The same approach is applied to leakage where the R dissipative effect structurally indicates the algebraic dependence of leakage on pressure while influence of temperature and velocity is introduced by changing the R element into a modulated RS field. A flux detector enables getting the shaft to body relative velocity signal. Causalities are forced for the MRS elements in order to meet numerical requirements and are propagated to facilitate the introduction of shaft inertia.

Figure 3. Energy losses in a Parker T7 vane pump at 3 temperatures vs. line pressure drop and pump speed (left: internal leakage, right: frictional torque)
Thermal model and dynamic parasitic effects such as shaft inertia and fluid compliance are not explicitly displayed on the figure that focuses on energy losses. It is important to keep in mind that when it is intended to address air release and cavitation, considering the outlet/inlet pressure difference is not acceptable as absolute pressure directly impacts these effects (inlet and outlet pressures are to be entered separately as modulating signals in the MRS elements).

When experiments are available, the MRS characteristic of friction and leakage can be simply defined as 3-D look-up tables. Each of them depends on velocity, pressure and temperature. However, developing a parametric model can be of interest for understanding the contribution of each input signal. Such a model is developed in the next section.

2.2. Parametric model of pump leakage and friction

This section intends to develop a parametric model based on the radial gap theory [9] where the effect of temperature is introduced by expressing the fluid viscosity as a function a temperature, according to the Reynolds equation [10]. In this approach, influence of pressure on viscosity is neglected in comparison with influence of temperature.

2.2.1. Internal pump leakage

According to the proposed approach, the internal pump leakage \( Q_{ip} \) is given by the following expression as a function of shaft to body angular velocity \( \omega_s \), pump pressure difference \( \Delta p \) and temperature \( T \):

\[
Q_{ip} = \beta_{ip}\omega_s^a + \gamma_{ip}\frac{\Delta p^b}{\mu_0e^{-\sigma(T-T_0)}}
\]  

where parameters are identified from experiments using a least square method \( (\beta_{ip} = 4.94 \times 10^{-5} \text{ International System Unit}, \gamma_{ip} = 3.14 \times 10^{-14} \text{ ISU}, \ a = 0.19 \text{ and } b = 0.94) \). The reference absolute viscosity \( \mu_0 \) is taken from data sheets at reference temperature \( T_0 \) and \( \sigma = 0.037 \) °C. The relative error between parametric leakage model and measured leakage is globally lower than 10% and never exceeds 25% for the whole tested domain of Figure 2.

2.2.2. Frictional pump torques

In the same manner, the model of frictional torque \( C_f \) originally proposed in [11] is modified to introduce the dependency of viscosity to temperature:

\[
C_f = \alpha_f + \beta_f\mu_0e^{-\sigma(T-T_0)}\omega_s^2 + \gamma_{fp}\Delta p^b
\]
where parameters are again identified from experiments ($\alpha_{fp} = 0.92$ Nm, $\beta_{fp} = 8.33 \times 10^{-6}$ ISU, $\gamma_{fp} = 2.79 \times 10^{-12}$ ISU, $C = 2.64$ and $D = 3.17$). The relative error between parametric friction model and measured friction is globally lower than 15% and never exceeds 28% for the whole tested domain of Figure 2.

In conclusion, the parametric model of pump energy losses involves 9 parameters. It can be used with profit for control design (the control loops admit modelling errors). However, as it introduces a non-negligible modelling error, it is not consistent with accurate simulation of thermal-hydraulic coupling in open loop model operation. This is certainly due to the presence of numerous and coupled effects (e.g. dilatation, centrifuge effects, local cavitation, etc.) that make energy losses difficult to be accurately modelled in practice. For this reason, look-up tables are preferred for the development of the thermal-hydraulic pump model. They enable frictions and leakages to be reproduced within the range of experimental data acquired in laboratory. Unfortunately, the rapid temperature increase does not allow taking measurements at extremely low shaft velocities. It is important to note that out of the experimental range, data have to be extrapolated that introduces modelling uncertainties.

2.3. Thermal pump model through virtual prototyping

A thermal pump model was developed in LMS-AMESim environment [12] in order to simulate the heat exchanges between hydraulic fluid, pump body and external environment and finally to evaluate the time history of the fluid temperature inside pump. As far as possible, the model was built from AMESim libraries in order to take benefit of well established, numerically robust and documented sub-models. The energy losses of the vane pump were modelled by means of look-up tables for the above-mentioned reasons.

Figure 5 displays the numerous coupling or interactions that must be taken into account to get realistic thermal simulation.

![Figure 5. Vane pump thermal-hydraulic virtual prototype](image-url)
The virtual prototype principally consists of two parts. The bottom of figure reproduces the pump energy losses, the pump/circuit thermal-hydraulic interface (right-hand side) and the pump/motor mechanical interface (left-hand side). The second part, in the top of figure, represents the pump body and its thermal interaction with electric motor (left-hand side) and environment (upper side).

In the first part, the vane pump is modelled according to the Bond-Graph of Figure 4. Two thermal-hydraulic volumes are added at suction and delivery sides where mass and energy balances are simultaneously applied. Fluid temperature is measured in the suction volume and in the discharge volume, in accordance with the effective placement of the thermocouples in the real pump of the test bench. From a causal point of view, the thermal-hydraulic model computes torque, flow and heat flow from shaft/body angular velocity, pressure at pump ports and ambient temperature. One important choice was related to the location where heat is injected in the pump model. The heat generated by the pump leakages was transferred directly to hydraulic fluid filling the suction pump volume because it is mainly carried by the fluid that recirculates internally from the high pressure to the low pressure domain. The heat due to pump friction was transferred to hydraulic fluid filling the discharge volume. This choice was driven by considering that the pump friction is mainly produced in the regions where the back pressure increases the contact loads between sliding and mating parts.

In the top part of the model, the pump body is considered as a single solid made of cast iron which exchanges heat with motor, fluid and ambient: internal forced convection between body and fluid, thermal conduction and heat carriage between pump and motor through shaft and coupling, natural convection and radiation with environment. In practice, the main issues encountered during the development of the thermal-hydraulic pump model were related to the modelling and parameterisation of resistive heat exchange between components.

3. FULL THERMAL MODEL

In order to evaluate correctly the time evolution of the fluid temperature inside pump during typical injection moulding machine’s cycles, in particular in holding pressure phases, it is necessary to connect the thermal-hydraulic pump virtual prototype, described in section 2, to the model of electric motor and hydraulic circuit. In this way, it is possible to simulate the complete heat transfers between hydraulic fluid, pump, electric motor, hydraulic circuit and external environment. Figure 6 shows the full thermal-hydraulic model of EHM used to emulate the real test bench performance that will be described in section 3.2.

The complete virtual prototype is composed by a thermal-mechanical modelling of the electric motor, a thermal-hydraulic modelling of the hydraulic circuit and a thermal modelling of the base support where the module is installed.

As the inverter can output a signal image of the electromagnetic torque, it was decided to avoid modelling the inverter, the motor electromagnetics and the control loops in order to keep the model simple at preliminary design stage. Therefore the electric motor was modelled as a perfect power transformer plus copper losses, friction and rotor inertia.

Such a decision sets up a high challenge for the validation of the EHM model. Indeed, when the inverter and the control loops are modelled, as generally done, any modelling error impacting the fastest states (e.g. electromagnetic torque - or current) has a low influence on slower state variables (e.g. velocity). In the present case, as no loop is modelled, the model is forced to operate in full open loop where modelling errors sum: impact of friction on speed for a given drive torque, impact of leakage on output flow for a given shaft velocity, impact of output flow on output pressure for a given hydraulic load…and in addition impact of temperature on every effect.

Motor copper losses were calculated from the motor electromagnetic constant and the windings resistance. The mechanical power losses due to friction were also introduced from motor supplier data. The motor body exchanges heat with pump body and with its base support through thermal conduction. At the same way of
the pump body, it interacts with the ambient air by means of natural convection and radiation with environment walls.

The model of the hydraulic circuit consists of thermal-hydraulic pipes with simulated compressibility and friction effects, load control valve, pressure relief valve, tank and sensors.

![Thermal Hydraulic Circuit](image)

**Figure 6.** Complete thermal virtual prototype of EHM (top: hydraulic load and test bench, bottom; electrohydrostatic module)
3.1. Virtual testing of EHM virtual prototype: determination of heat transfer parameters

As far as possible, it was intended to get a first estimation of the heat transfer coefficients from existing models associated with generic heat transfer schemes [13] to [16] and to check whether or not the pre-calculated values were close to identified ones. Model development was also driven firstly by available submodels of heat transfer.

3.1.1. Internal forced convection

Internal forced convection coefficients $h_{\text{int, conv}}$ are usually extracted from the calculation of the Nusselt number $Nu$ according to equation (3):

$$h_{\text{int, conv}} = \frac{Nu \ k_f}{L_c}$$

(3)

where $k_f$ is the fluid thermal conductivity and $L_c$ the characteristic length.

There are a lot of models for calculation of the Nusselt number versus flow conditions (Reynolds number $Re$) and fluid temperature condition (Prandtl number $Pr$). However, any model applies to very specific geometries or flow that do not match with the real operation of a hydraulic pump. In the present work, Nusselt correlation for internal forced convection between pump body and hydraulic fluid was estimated with the Dittus-Boelter equation associated to turbulent flow in tubes, obtained by modification of the Colburn equation (4) where $n$=0.4 for heating of the fluid flowing in the tube [13]:

$$Nu = 0.125 \ f \ Re \ Pr^n$$

(4)

The (4) is valid for Reynolds numbers higher than $1 \times 10^4$ and for Prandtl numbers in the range $0.7 \leq Pr \leq 160$. The friction factor $f$ was determined through the simple power law relation (5) that applies to turbulent flow in smooth tubes [13].

$$f = 0.184 \ Re^{-0.2}$$

(5)

The properties of the fluid were calculated at the bulk mean fluid temperature given by:

$$T_{\text{mean}} = \frac{T_{\text{fluid}} + T_{\text{pump-body}}}{2}$$

(6)

This approach enabled the effect of flow and temperature to be considered in the calculation of the convective coefficient but required the friction factor $f$ and the characteristic length $L_c$ to be set as model parameters.

3.1.2. Natural convection and radiation

The natural convection over surface mechanics was taken into account to evaluate the heat transfer between the external environment and respectively: pump body, motor body and base support. The heat transfer coefficients $h_{\text{conv}}$ were calculated by the same expression for the $h_{\text{int, conv}}$ (3) where $k_f$ was now the thermal air conductivity and the $Nu$ were determined by means of empirical correlations dependent on the Rayleigh number $Ra$ which is the product of the Grashof number $Gr$ and the Prandtl number $Pr$ [13]:

$$Ra = Gr \ Pr = \frac{g \beta (T_s - T_{\infty}) L_c^3}{\nu^2}$$

(7)

where $g$ is the gravitational acceleration, $\beta$ is the coefficient of volume expansion, $T_s$ is the surface temperature, $T_{\infty}$ is the fluid temperature far from the surface and $\nu$ is the kinematic viscosity of air.

Table (1) summarises all the empirical relations for the calculation of $Nu$, applied for each system component. The pump body was compared to a sphere having the same external surface area, the motor body was
compared to a horizontal cylinder having the same side area and finally, the base support was compared to a horizontal plane having the same surface area and perimeter.

Table 1. Nusselt numbers calculation for natural convection over the surface of: pump and motor body and base support

<table>
<thead>
<tr>
<th>System Component</th>
<th>Shape</th>
<th>Nusselt number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump body</td>
<td>Sphere</td>
<td>$N_u = 2 + \frac{0.589 R_a^{1/4}}{1 + (0.469/Pr)^{9/16}R_a^{4/9}}$</td>
</tr>
<tr>
<td>Motor body</td>
<td>Horizontal cylinder</td>
<td>$N_u = \left{ 0.6 + \frac{0.387 R_a^{1/6}}{1 + (0.559/Pr)^{9/16}R_a^{8/27}} \right}^2$</td>
</tr>
<tr>
<td>Base support</td>
<td>Horizontal plane</td>
<td>$N_u = 0.15 R_a^{1/3}$</td>
</tr>
</tbody>
</table>

The heat radiation was also considered because, even if it usually represents the smallest contribution to total heat exchanges, it cannot be negligible when the released and absorbed powers are evaluated in a temperature prediction approach [15]. The following expression was used for the heat radiation transfer $\dot{Q}$:

$$\dot{Q} = -\sigma_s \varepsilon A_e (T_s^4 - T_w^4)$$

where $\sigma_s$ is the Stefan-Boltzmann constant, $\varepsilon$ is the surface emissivity and $A_e$ is the emitting surface. It was assumed $\varepsilon=0.18$ for pre-polished aluminium of motor body [14], $\varepsilon=0.6$ for cast iron of pump body and $\varepsilon=0.56$ for rolled steel sheets of base support, from literature.

3.1.3. Thermal conduction

Heat transfer at the interface between the module components was evaluated by means of the thermal conduction and contact relations [13]. In detail, it was assumed to be heat conduction between pump and motor body simultaneously through both the coupling and the common shaft. The value of the thermal contact conductance depends on a lot of parameters as the surface roughness, material properties, temperature and pressure at the interface. The difficulty in determining the heat transfer in this case was solved by assuming an overall constant thermal contact conductance, based on typical values for dissimilar metal surfaces pressed at high pressure each other [13].

Concerning the thermal conduction between motor body and base support, the thermal resistance network for heat transfer through a two-layer plane wall was applied. In this way, it was firstly possible to estimate the total thermal resistance as the arithmetic sum of the individual thermal resistances of motor body and support.

3.2. Test bench and laboratory tests

In order to reduce the tests complexity, the EHM was placed on top the tank, Figure 2. In the next future, it is planned to develop an immersed solution that will reduce temperature rise and noise emission to ambient. During the tests, the pump outlet pressure was closed-loop controlled by action on the motor torque using proportional control plus feedforward action to compensate for friction, inertia and hydrostatic torques. In both of tests an increase of motor speed $\omega$ is so automatically produced, thanks to constant pump output pressure, in order to compensate the effects of reduction of output flow rate due to fluid temperature increase. The $\omega$ increase so enables to refrigerate the inside of pump by sending cooler input flow rate. The controller was then implemented into a SpeedGoat host controller that drove directly the motor inverter as a torque demand signal. The control laws were implemented in the Matlab/Simulink Real-Time Workshop environment with a sampling rate of 1 kHz. The output flow at operating point, emulating the static injection,
was fixed by setting opening of the hydraulic load valve prior the test running. In order to facilitate discussions at pump factory level, the pump operating point during static injection was defined by the effective volumetric efficiency at which it operates at the beginning of the test, when the pump is driven to generate a constant output pressure of 160 bar at a very low delivery flow. Two values were selected:

- $\eta_{vol} = 10\%$ that requires the pump to start test at 124 rpm (4.1% rated speed) shaft velocity at 45°C (Test-01)
- $\eta_{vol} = 30\%$ that requires the pump to start test at 146 rpm (4.9% rated speed) shaft velocity at 45°C (Test-02)

Five temperatures sensors were installed (inside tank $T_{tank}$, at pump outlet $T_{out}$, in drain pump volume $T_{f-in-p}$ and into the pressure plate, at pump housing surface $T_{p-body}$ and at motor housing surface $T_{m-body}$) in addition to the motor speed $\omega$, the line pressure drop $\Delta p$, the output pump flow rate $Q_{out}$, the estimated motor torque $C_{est}$.

### 3.3. Comparison and validation

Figure 7 compares the measured and simulated temperatures of $T_{f-in-p}$, $T_{p-body}$ and $T_{m-body}$. In the virtual prototype the initial conditions of temperatures were set according to the measurements. The flow control valve static hydraulic characteristic was modelled using look-up tables providing the mass flow rate as function of $\Delta p$ and $T_{out}$ that were acquired from the experimental data. The laboratory Test-01 was stopped when the limit condition of 90 °C was nearly reached by the $T_{f-in-p}$ (308 s) while Test-02 was stopped when the $T_{f-in-p}$ was not far from steady-state conditions (453 s).

As expected, both experimental results highlight that the measured $T_{f-in-p}$ increases more rapidly than $T_{p-body}$ and $T_{m-body}$. $T_{f-in-p}$ is characterized by two different thermal time constants: the first one of about $3/4$ s while the second one is much higher, over 7 min. The pump body has a thermal constant time of at least 16 min while the motor body thermal time constant is about 27 min from catalogue.

Although the virtual model is excited with a very constraining (open-loop) approach, the simulated $T_{f-in-p}$ remain close to the measured ones. The maximum relative errors are lower than 5 % in Test-01 (3 °C of temperature gap) and lower than 2 % in Test-02 (1.4 °C of temperature gap). The simulated and measured $T_{f-in-p}$ differ principally in the initial extent due to an overestimation of heat absorbed by fluid, while the final values result are very accurate: the temperature gap is lower than 0.4 °C.

It is interesting to remark that in Test-02, the pump transfers heat to motor for all test duration while in Test-01 the motor firstly transfers heat to pump body. In this case the simulated $T_{p-body}$ differ from the measured
ones, due to the higher simulated absorbed powers, of about 5 °C for Test-01 and 2.3 °C for Test-02. The simulated $T_{in\text{-body}}$ are instead very close to measured ones with temperature gaps that remain lower than 1.3 °C.

Table 2 shows the values of the heat transfer coefficients selected in the complete EHM virtual prototype. It is possible to observe that the internal forced convection coefficients $h_{int\text{-conv}}$ varied during each simulation because the equation (4) for calculation of $N_{ir}$, as function of $R_i$ and $P_s$, was implemented in the prototype by means of a “supercomponent” built by starting from basic components of AMESim library (Fig. 2). In Test-01 the initial $h_{int\text{-conv}}$ is lower than in Test-02 due to lower starting $\omega$ that makes the initial $R_i$ lower. After that the increases of both $\omega$ and $T_{f,in-p}$ increase greatly the $h_{int\text{-conv}}$ in both Test-01 and in Test-02: it is remarkable that the increase of 100% of $\omega$ and $T_{f,in-p}$ with reference to starting point conditions for Test-01, made the $h_{int\text{-conv}}$ to be increased of 146%. The natural convection coefficients were then assumed invariant for all simulations time durations: their values were initially determined by the empirical relations in table 1 and then scaled by an incremental factor that took into account the realistic convection in laboratory where drafts could occurred [14]. These final values are coherent to values proposed in [15].

<table>
<thead>
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<th></th>
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<tbody>
<tr>
<td>Internal forced convection</td>
<td>from 1202 to 2964</td>
<td>from 1211 to 2208</td>
</tr>
<tr>
<td>Natural convection for pump body</td>
<td>25</td>
<td>25</td>
</tr>
<tr>
<td>Natural convection for motor body</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Natural convection for base support</td>
<td>25</td>
<td>25</td>
</tr>
<tr>
<td>Pump/motor thermal contact conductance</td>
<td>8600</td>
<td>8600</td>
</tr>
<tr>
<td>Motor/support thermal contact conductance</td>
<td>15400</td>
<td>15400</td>
</tr>
</tbody>
</table>

Figure 8 shows the percentage of absorbed and released power contributions simulated by virtual model at the end test times (in the top), and the overall absorbed and released energies during all time duration (in the bottom) for respectively pump body, motor body and fluid. Figures relative to Test-01 are on the left while relative to Test-02 are on the right.

Concerning the pump body, it is notable that the percentages of each absorbed/released power are similar for the two tests and that the volumetric losses, associated to pump internal leakages, represent the major contribution of absorbed power. The increase of internal energy of the fluid produces an enthalpy flow rate towards the pump body that represents the second contribution, more important than the mechanical losses due to pump frictions. Another fundamental observation is the importance of pump/motor thermal conduction that contributes with a percentage over the 80 % of total pump released powers.

The major source of heat absorbed by the motor body is instead provided by the pump while the electrical power of the motor represents a minor contribution. The thermal conduction with the base support finally permits to transfer the most of quantity of heat previously stored.

The figures relative to absorbed and released powers of the fluid show that the volumetric and mechanical losses, assumed initially stored in the fluid by the model, are completely released to pump body at the end of simulation. Only in the Test-02 it remains about 200 W of volumetric losses again stored in the fluid. The major percentage of released power is the enthalpy flow rate transported by the fluid flowing out of pump. Concerning the overall net energies, it was calculated 12 kJ and 4 kJ for the fluid at the end of each simulation to increase its temperature respectively of 41 °C and 24 °C: it corresponds to an average energy value of 0.29 kJ/°C and 0.17 kJ/°C. This difference is due to different value of $h_{int\text{-conv}}$ that governs the heat flow between pump and fluid (tab. 2): in fact $h_{int\text{-conv}}$ firstly lower in Test-01 than in Test-02, becomes higher at
the end of simulation by increasing the released power and by increasing finally the total net absorbed energy.

Figure 8. Contributions of absorbed and released powers at the end simulation time (top of each figure) and absorbed and released energies (bottom of each figure) for pump body, motor body and fluid. Data of Test-01 in left column and data of Test-02 in right column.
At a final overview on the complete EHM virtual prototype performance, it is possible to assert that the model provides the time evolution of temperature respectively of fluid, pump body, motor body and base support by assuming a uniform temperature distribution inside each of them. In fact pump, motor and support are modelled through an overall mass without modelling the sub-components like the rotor, cam-ring, vanes, cap and housing for pump. Consequently the model does not simulate the temperature gradients and so a temperature error can be produced between simulated and measured results, last ones obtained through thermocouples located in specific points on bodies’ surface. It was the case of pump body. Another observation can be made about the sensitivity of the virtual model to extrapolation of energy pump losses out of the range of experimental data: the pump experimental internal leakages and frictional torques were implemented in the model by means of look-up tables whose range of operating conditions is showed in figure 3. The model so extrapolates linearly the energy pump losses out of this range but it could produce not negligible errors that could modify significantly the final results. Just thinking to the internal leakages that represent the major contribution to power absorbed by pump body, and a little error of leakage flow evaluation, it would generate big errors on volumetric losses.

Nevertheless the model limitations just exposed, the virtual prototype enables to simulate accurately the time evolution of temperatures, especially of the fluid inside pump that represents a fundamental parameter to be controlled to guarantee the adequate operation of the vane pump during the hardest phases of a typical injection moulding machine’s cycle. The results of temperature simulation also permit to evaluate with good reliability the heat transfers between fluid, pump and motor body, base support and external environment by taking also into account the variation of the convection coefficients as function of operating conditions.

4. CONCLUSION

The reported work aimed to develop and validate a system-level thermal-hydraulic model of an electro-hydraulic module involving a fixed-displacement and internally-drained vane pump to be used in moulding machines. It was in particular intended to reproduce by simulation - and with resort to a minimal set of parameters - the rapid and huge temperature increase during static injection where high pressure has to be hold at (almost) null flow. The thermal-hydraulic model of pump leakages and friction losses has enabled velocity, pressure and temperature to be considered either using parameterized equations or look-up tables. On this basis a thermal-hydraulic model of the pump has been developed by addition of two thermal-hydraulic capacitances (at low pressure side and at high pressure side). Real experiments were performed at high pressure and extremely low delivery flow, forcing the pump to operate at 10 or 30% volumetric efficiency. Although the pump model was validated in a very constraining way that made all modelling errors cumulate, the temperature evolution was correctly predicted. The works showed the significant influence of the heat exchange between the pump and the motor on the fluid temperature and the lack of knowledge models for calculation of convection factors that can apply to the geometry and the operative conditions associated with the application's need.

REFERENCES


A NOVEL CONCEPT FOR A VARIABLE DELIVERY EXTERNAL GEAR MACHINE

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ABSTRACT

This paper presents a design analysis of a novel concept of variable delivery flow external gear machine (EGM) previously introduced by the authors. This innovative design encompasses all the well-known advantages of traditional EGMs (low cost, reliability and efficiency), but also introduces a cost effective solution for varying the amount of fluid displaced per unit revolution of the shaft. The paper summarizes the main features of the design, which is essentially based on a unique implementation of a variable timing concept for the connections of the tooth space volumes with the inlet/outlet ports. To permit high range of flow regulation, this new design utilizes asymmetric profiles for the teeth.

In a previous paper, the authors tested the new design on an existing pump, utilizing the same casing and designing new internal parts (gears and lateral bushings) suitable for the implementation of the new concept. With this concept tests, a potential for flow variation in the range 100% - 68% with high level of energy efficiency – comparable to other existing variable displacement pumps – was shown. The design utilized, however, was characterized by design constraints on the gear profiles given by the requirement of utilizing an existing case (fixed outer radius and fixed inter-axis).

In the present study, a parametric study for the proposed variable delivery flow unit is performed to understand its full potentials. The study is performed through an optimization procedure implemented in modeFrontier® and based on the use of the HYGESim simulation tool for EGMs developed in the authors’ research center. Results shows that assuming the involute profile for both the cost and drive side of the gears, the range of flow variation tends to increase with the number of teeth per gear, and a 62% flow variation can be achievable with a 30 tooth profile.

KEYWORDS: external gear machines, variable displacement, variable delivery, design optimization

1. INTRODUCTION

External gear machines (EGMs) are widely used in hydraulics. Because of their advantages in terms of cost, reliability, customizability and flexibility of installation, they are the preferred unit in several mobile (such as agricultural-, construction- and earthmoving-machines) and fixed applications (such hydraulic presses and forming machines). EGMs are often used as the primary flow units, however, they are also utilized in several auxiliary systems, such as fuel injection systems, hydraulic fan drives, power steering systems and charge pumps in hydrostatic transmissions.
EGMs are typically fixed displacement machines, with a principle of operation which can be shown in Figure 1, which represents the case of a pressure compensated EGM for high pressure applications (up to 300 bar). As represented in Figure 1(A), the fluid is transferred from the inlet to the outlet due to the rotation of the gears, while the displacing action is realized by the meshing of the teeth. As shown in Figure 1(C), at the lateral side of the gears there are grooves (machined in the lateral bushings for the pressure compensated design, or in the casing for non-compensated units) which perform the timing function of connecting each tooth space volumes (TSV) - when trapped between the points of contact of the gears - with the inlet/outlet ports. These grooves ensure the utilization of the full volumetric capacity of the unit, and avoid localized pressure peaks and internal cavitation effects [1,2].

![Figure 1. (A) Operating principle of an EGM. (B) Parts of an EGM (C) Detail of the lateral bushings highlighting the grooves machined on them.](image)

Being inherently fixed displacement, EGMs are unsuitable for efficient system layouts based on fixed speed for the prime mover, such as load-sensing systems, displacement controlled systems etc. use variable displacement (VD) units which are capable of achieving more than 50% reduction in energy consumption as compared to those which use fixed displacement units [3,4].

In order to formulate a cheap, reliable unit which can vary the displacement, researchers have made significant efforts in studying a working design for variable displacement type EGMs. At a system level, [5-7] describe a solution for a “virtually variable flow” unit, by combining fixed displacement pumps with fast on/off switching valves using pulse width modulation. However, this solution requires valves with very high bandwidth (at the limit of what is currently commercially available), and it introduces additional pressure fluctuations peaks which might compromise the reliability of the EGM. Alternatively, at a component level, a few patents have been published which achieve variable displacement by movement of the gears. Particularly, [8-11] represent patents which describe the principle of achieving a displacement variation by moving the gears axially, on the other hand, [12, 13] obtain a change in displacement by moving the gears radially. Nevertheless, all the patents described were not successfully commercialized due to the complexity which arises in implementing a solution to move the gears which happen to be the most loaded parts of the machine. Obtaining a proper sealing of the displacement chambers and a smooth transmission of power between the gears poses another problem to obtain a cost effective and feasible solution.

Another solution presented in [14, 15] proposed the use of an adjustable spool inside the EGM to vary the communication of the fluid in the TSVs with the inlet/outlet port. This solution, in a certain extent similar to the one investigated by the authors of this papers, is limited by the use of conventional profiles of the gears, which doesn’t permit a significant range of flow variation while guaranteeing high energy efficiency.

The design proposed in this work, first introduced in [16], utilises a variable spool introduced as additional element (slider) to vary the timing of the connections of each TSVs, and it is based on unconventional design
for the gears to obtain high range of flow regulation. In particular, the gear profile is based on asymmetric involute and trochoid profiles for the gears. A summary of the concept idea is provided in section 2.

Through an optimization process, the authors in [16] formulated a design for the gears and the of the slider which they successfully tested, showing a potential for flow variation range between 68% to 100% of full displacement by maintaining high energy efficiency (in terms of volumetric efficiency and torque efficiency). The design process, however, was constrained by the requirement of using a commercial pump as reference for the tests. In particular, the gears and the lateral bushings of a commercial unit were replaced with prototype parts derived from the numerical optimization process. This implied assuming a priori a certain value of the gear outer radius and inter-axis.

In this work, the design procedure based on a two-level optimization process (which will be described in section 3) will be utilized without the above specified constraints, thus permitting to investigate the full potentials of the proposed concept, and identify tendency of certain parameters (such as the number of teeth) on the performance level of the proposed solution.

2. DESCRIPTION OF THE NOVEL DESIGN CONCEPT FOR VARIABLE DELIVERY EGM

The novel design concept is based on the attainment of a variable timing of the connections of the TSVs with the inlet/outlet lateral grooves. The variation in the timing of the connections is achieved by the introduction of a single-axis movable element – called the “slider” within the lateral bushings as shown in Figure 2(A). The position of the slider, which could be also placed in the pump casing for a non-pressure compensated EGM, determines the amount of flow displaced by the unit per revolution of the shaft.

![Figure 2. (A) Detail of the bearing block showing the slider (B) Angular range, θ within which the meshing process occurs, points D through S where the TSV is trapped between the contact points of the gears](image)

The displacement of the fluid in the EGM occurs in the angular interval θ, which defines the meshing region. The TSV (assuming a dual flank contact configuration of the gears as shown in Figure 2(C)) is trapped between the points of contact represented by D and S of the line of action, therefore within this region the displacing action is realized by means of the inlet/outlet grooves (the trace of these grooves is indicated in Figure 2(B). In standard EGMs, the commutation between inlet and outlet groove is realized when the volume is minimum (represented by “M” in Figure 3(A)), so that the maximum volumetric capacity of the pump is utilized. Nevertheless, a small degree of cross-porting will be necessary to obtain an optimal performance in terms of minimizing internal pressure peaks, cavitation and fluid borne noise emissions [17, 18]. The slider shown in Figure 2(A) is capable of a single degree of motion as represented by the arrow; hence, it can vary the position of these grooves affecting the angular position at which the commutation between the connections of each TSV with the inlet/outlet ports occur. The effect of different positions of the
slider can be described with Figure 3. In particular, Figure 3(A) pertains to the maximum flow condition, in which the TSV is connected to the inlet and outlet for equal intervals of time, by switching the commutation from outlet to inlet at “M”. A variation of the displaced flow per every shaft revolution is achieved when the slider is positioned closer to the inlet port, Figure 3(B). In this condition, each TSV remains connected to the outlet port for a first portion of the filling process, after it reaches the minimum volume point at 180°, wherein a part of the fluid already delivered to the outlet is taken back into the TSV. Therefore, each TSV displaces to the outlet a reduced volume of fluid, with respect to the maximum flow condition. As shown in Figure 3(A), for the reduced flow condition the TSV is connected to the delivery for a larger angular period (in general between “M” and “S” for a reduced flow condition, until “S” for the minimum flow condition), and the transition from the outlet to the inlet port occur with an additional dead volume (with respect to the maximum flow condition) which can be represented by the difference between volumes at “S” and “M”.

![Figure 3](image)

**Figure 3.** (A) The evolution of TSV with respect to the shaft rotation angle, the switch of the TSVs connection from the outlet to the inlet occurs at “M” for achieving max. displacement and at “S” for achieving min. displacement. (B) Position of the grooves to achieve min. displacement.

With the described principle it is necessary to notice how the variation of the displacement as a result of change in the slider position occurs for all slider positions that realize the described switching between “D” and “S”, within which the fluid TSV is trapped between points of contact of the teeth. Outside this range the slider would realize a direct connection between the inlet and outlet port through the gear depth, being the volume not trapped between points of contact. Conventional gears for EGMs with symmetric teeth have a very small angular range of trapped volume, meaning that points “D” and “S” lay very close to “M”. This makes the conventional EGM design not advantageous in offering large variation in the displacement according to the proposed principle. This is the reason why in [12] the authors focused on an asymmetric profile of the gears that permits to enlarge the range of variation of displacement.

### 2.1. Design of the Asymmetric Gear Profile

In the proposed design, the gears are assumed to be comprised of involute and trochoid profiles above and below the base circle respectively. In order to accomplish the goal of designing asymmetric teeth, two different pressure angles are considered respectively for the drive and opposite coast tooth flanks, as shown in Figure 4. To ensure that the asymmetrical gear profile is physically manufacturable using conventional manufacturing processes like hobbing, shaping, rack-cutting etc., an asymmetrical cutter profile is assumed at first. The tooth profile is then derived on the basis of the shape of the asymmetric cutter as shown in Figure 4(A).

The involute profiles for both the drive and coast side of the teeth can be obtained using the Eqs. (1) – (3). These equations are represented in a generic form for any involute side of the teeth, changing the values of
\( r_b \) and \( \vartheta_g \) for the drive or coast yields respectively the corresponding involute profiles as shown in Figure 4(B) [16,19].

\[
x = (\sin \vartheta_g - \vartheta_g \cdot \cos \vartheta_g) \cdot r_b \cdot \cos \vartheta_g - (\cos \vartheta_g + \vartheta_g \cdot \sin \vartheta_g) \cdot r_b \cdot \sin \vartheta_g
\]

\[
y = (\sin \vartheta_g - \vartheta_g \cdot \cos \vartheta_g) \cdot r_b \cdot \sin \vartheta_g + (\cos \vartheta_g + \vartheta_g \cdot \sin \vartheta_g) \cdot r_b \cdot \cos \vartheta_g
\]

where,

\[
\vartheta_g = \text{inv} \alpha_0 + \frac{\pi}{2 \cdot z}
\]

Similar to the construction of the involute profiles, the trochoid profiles of the teeth are obtained using the generic eqs. (4) – (6), as shown Figure 4(B) [16,19].

\[
x = (r_0 - h_0) \cdot \sin(\xi_g + \nu_g) - r_0 \cdot \xi_g \cdot \cos(\xi_g + \nu_g) - \left[ \frac{r_0 \cdot \xi_g + h_0}{\sqrt{h_0^2 + r_0^2 \cdot \xi_g^2}} \right] \cdot \rho_r \cdot \sin(\xi_g + \nu_g)
\]

\[
y = (r_0 - h_0) \cdot \cos(\xi_g + \nu_g) + r_0 \cdot \xi_g \cdot \sin(\xi_g + \nu_g) + \left[ \frac{r_0 \cdot \xi_g - h_0}{\sqrt{h_0^2 + r_0^2 \cdot \xi_g^2}} \right] \cdot \rho_r \cdot \sin(\xi_g + \nu_g)
\]

where,

\[
\nu_g = \frac{\pi}{2 \cdot z} + \frac{h_0 \cdot \tan \alpha_0}{r_0} + \frac{\rho_r + b_n}{r_0 \cdot \cos \alpha_0}
\]

The value of \( b_n \) controls the backlash in the gear pair generated; therefore setting its value to zero yields gears with zero backlash or dual flank contact.

Figure 4. (A) Asymmetric tool for generating asymmetric gear profiles (B) Drive and coast involute and trochoid profiles of the gears (C) Lateral bushings with a representative inlet/suction and outlet/delivery groove [16]
2.2. Design of the Grooves in the Lateral Bushings

The grooves machined in the slider of Figure 2(A) determine and control the amount of fluid displaced per revolution of each TSV. With an optimal simultaneous connection of the TSV with the inlet and outlet port (often referred as crossport), the grooves can also ensure minimal internal pressure overshoots and localized cavitation effects during the transition of TSV from/to the low pressure and high pressure regions maintaining minimum bypass leakage flows. In the proposed design, a particular “two-winged” shape of the grooves, shown in Figure 4(C), is considered. The main intent for using such architecture with four angular parameters (α’s) is to understand the influence of the pressure angles (drive and coast side) of the gear profiles on the performance of the machine [16]. Particular emphasis is placed on the feasibility of machining the grooves using the conventional milling process for prototyping.

2 HYGESIM SIMULATION TOOL

In this work, the predictions of the performance of the variable displacement type EGM are made using the HYdraulic GEar Machines SIMulator (HYGESIM) tool which is being developed by the authors’ research team. Particularly, the calculations of the delivery flow, tooth space pressures, forces acting on the gears and the input shaft torque are important to accurately understand the performance of the novel design concept and its potentials for delivery flow variation. As described in [20] HYGEsim consists of 3 main modules: geometrical model, fluid-dynamic model, mechanical model and fluid structure interaction model. The geometrical model of HYGEsim is currently capable of processing gears and grooves with the features described in sections 2.2 and 2.3. All the geometrical information (such as the instantaneous volume and connection areas of each TSV) required by the fluid-dynamic model of HYGEsim is provided by the geometrical model. The fluid dynamic model of HYGEsim uses a lumped parameter approach to evaluate the pressure in the tooth space volumes (TSVs) and the mass flow between the TSVs and the inlet and outlet port. The model is implemented in LMS.AMESIM® and is capable of evaluating inlet and outlet flow and pressure, flow through the TSV, instantaneous meshing pressure. The mechanical model accounts for the radial micro-motion of the gear resulting from the radial forces acting on them. An accurate evaluation of radial leakages, at the tip of each tooth, is performed considering the actual position of the gears, resulting by a force balance that include contact forces and pressure forces. Details on the geometric model and the lumped parameter fluid dynamic model inclusive of the evaluation of the gears’ micro-motion are provided in [20]. The fluid structure interaction model of HYGEsim deals with the study of the lubricating gap at gears’ lateral sides, considering the deformations of the solid parts due to fluid pressure or thermal effects [21, 22]. However, in this study, the lateral gap model of HYGEsim is utilized in a simplified way, considering a realistic value for a constant gap height, assuming that the axial balance design problem of the novel design can be addressed as future work.

3 OPTIMIZATION PROCEDURE

The approach used to find the optimal design of the asymmetric gears and the grooves in the slider of the proposed variable delivery EGM are determined by using an optimization method based on a multi-level-multi-objective genetic algorithm. A schematic of the overall optimization workflow is shown in Figure 5(A): Level 1 is the top level which deals entirely with the design of asymmetric gears; while Level 2 is the sub-level which optimizes the grooves in the lateral bushings for a given design of the gears. This optimization uses HYGEsim as a calculation block, for evaluating the performance of the considered design based on several objective functions. The optimization workflow uses a hybrid algorithm of multi-objective genetic algorithm combined with response surface methodology to accelerate the convergence of the design process. This algorithm aids in the fast evolution of new designs at every iteration, since there are more than one objective functions and they lack analytical expressions in terms of the design variables.
This optimization procedure is similar to the one used in the past work done by the authors on the proposed concept [16]. A similar version of the optimization workflow of Figure 5(A) was also used to optimize the design of standard involute gears for EGMs [23]. However, as mentioned above, in [16] the optimization was constrained for prototyping and experimental purposes, hence the results of the optimization were limited. In the present work, the optimization is performed completely relaxing these constraints to determine the maximum potential in displacement variation that can be achieved by using asymmetric gear profiles. The following sub-sections detail the formulation of the objective functions as well as constraints used in this procedure.

### 3.1 Objective Functions

**Maximize reduction in displacement (OF$_1$):** The maximum reduction in displacement is calculated by accurately tracking the location of the points D and S of both the gears TSVs as shown in Figure 5(B). Due to the asymmetric nature of the gears, the location of the points ‘D’ and ‘S’ (which represent the angular location when the TSV is trapped) are not symmetric. The resultant minimum displacement achievable can be expressed as an average of the ones provided by the drive and the slave/driven TSVs independently. The minimum displacement achievable can be calculated using eq. (7) [16].

$$
\beta = \frac{\beta_{\text{drive}} + \beta_{\text{driven}}}{2} = \frac{V_{S_2} + V_{S_1}}{2 \cdot V_M}
$$

Therefore, the objective function for maximizing the reduction in displacement can be expressed as eq. (8).

Maximize: \( OF_1 = 1 - \beta \)  

**Minimize delivery flow ripple (OF$_2$):** The fluctuation of the flow at the delivery is one of the major contributions to the fluid borne noise [24,25]. An accurate estimation of the delivery flow ripple can be quantified using the energy of the signal [24]. The estimate of OF$_2$ can be expressed as the sum of squares of the amplitude corresponding to the fundamental harmonics obtained after a Fast Fourier Transform (FFT) of the flow ripple. Eq. 9 [24] represents the expression for OF$_2$,

$$
\text{Minimize: } \text{OF}_2 = \sum_{k=1}^{N} \pi_k \cdot L(f_k)^2
$$

where \( N \) is the total number of fundamental harmonics of interest. \( \pi_k \) is the total energy of \( k^{th} \) fundamental frequency about a frequency resolution, \( \Delta f \) and \( L(f_k) \) is the FFT of the flow ripple signal.

**Minimize internal pressure peaks (OF$_3$):** Internal pressure peaks occur when the gears are in the meshing zone specifically due to the trapped volume of the fluid which exists between the points of contact of the teeth. At the beginning of the meshing process, the volume decreases leading to significant overshoots of fluid pressure, if an optimal design of the grooves are not considered. OF$_3$ is expressed as a non-dimensional value given by eq. 10 [23,26].
Minimize \(OF_3 = \frac{P_{TSV,\text{peak}} - P_{D,\text{avg}}}{P_{D,\text{avg}}}\) \hspace{1cm} (10)

Where, \(P_{TSV,\text{peak}}\) is maximum pressure reached inside the TSV during the meshing process and \(P_{D,\text{avg}}\) is the average delivery pressure of operation.

Minimize localized cavitation (OF\(_4\)): At the end of the meshing process, the fluid is still trapped between the contact points of the teeth but the TSV starts increasing. This might lead the pressure in the TSVs to fall below the saturation pressure of the fluid. This effect can be reduced by using an optimal design of the grooves machined on the slider. \(OF_4\) is expressed by Eq. (11), as the area of the tooth space pressure curve which falls below the saturation pressure of the fluid [23,26].

\[
\text{Minimize: } OF_4 = \int P_{TSV} \, d\theta, \quad \forall P_{TSV} < P_{\text{sat}} \hspace{1cm} (11)
\]

Maximize volumetric efficiency (OF\(_5\)): The performance of the machine in terms of the volumetric displacement of the pump needs to be maximized considering all the other objective functions simultaneously. The expression for \(OF_5\) is given by eq. (12).

\[
\text{Maximize: } OF_5 = \eta_v = \frac{Q_{\text{avg}}}{n \cdot V_d} \hspace{1cm} (12)
\]

Where, \(Q_{\text{avg}}\) is the mean flow provided by the machine at its delivery while operating at a speed of \(n\) rpm and \(V_d\) is the displacement of the machine.

3.2 Constraints

With the wide range of variability of the several design variables, constraints need to be enforced to ensure the feasibility and the manufacturability of the generated designs.

**Contact ratio:** In order to ensure the smooth transmission of forces between the gears, there should be at least one pair of teeth which is always in contact with each other (for load sharing) during the working of the machine [27]. Therefore the contact ratio for both the sides of the gears, (namely the drive and the coast) should always ensure to be greater than one.

**Interference:** The process when the corresponding drive or the coast sides of both the gears intersect with each other is known as interference. With interference, the gear teeth are weakened due to the removal of the material near the root of the gears [27].

**Pointed teeth:** This constraint restricts the optimization algorithm to generate gear designs which have very pointed tooth tips. Hence this constraint prevents the wear and tear of the casing due to the sharp gears [27].

Additional constraints on the **bending stress** and the **resistance** of the gears to pitting were also considered to ensure that the gears generated are within allowable limits of the stresses [16].

4 RESULTS

This section illustrates the results of the optimization process which took a total of around 500 hours of computational time (on a Dell® Precision T1600 workstation, with Inter® Xeon® CPU @ 3.30GHz). Almost 200 designs of gears combined with around 100 designs of lateral bushings per gear design was analyzed to arrive at optimal condition. The optimal design of the gears at the end of the optimization process was at a Pareto optimum of all the objective functions considered. In order to understand the potentials of using asymmetric gears on the reduction in displacement, a trend-line for the min. displacement as a function of number of teeth per gear is plotted in Figure 6. The dots plotted in Figure 6, represent the minimum displacement for the particular teeth number from all the designs considering its combinations with the design variables. It can be seen that there is a clear linear trend for the minimum displacement as the number of teeth increases. The maximum number of teeth was limited to 30, above this number very few design can achieve thickness able to guarantee proper structural resistance.
An example of a design with 16 teeth which offered a 32% (from 100% to 68%) variation in the displacement is shown as ‘A’ in Figure 6 and Figure 7. This design was previously selected and a successful proof of concept test was performed as described in [16].

![Figure 6. Trend for min displacement as a function of number of teeth per gear](image)

![Figure 7. Profile of asymmetric gear, A, with 16 teeth](image)

A higher reduction in displacement can be realized by the 30 teeth design ‘B’ in Figure 6, represented in details in Figure 8 and Table 1. This design, more aggressive as compared to existing EGM for high pressure applications as concerns number and thickness of teeth, can offer a displacement variation from 100% to 62%, higher than what already tested by the authors [16]. It can be seen from Figures 6 and 7 that the degree of asymmetry is very low, this is because, a lower pressure angle would imply that the line of action of the gears are steep and hence ensuring a larger extent of trapped volume. This effect combined with a higher number of teeth introduces a larger reduction in displacement as shown in Table 1.

However, these results highlight the limit for achieving high flow variation by using asymmetric gears with the proposed design concept. Different non-conventional design (such as those proposed in [28]) will be investigated by the author to achieve further reduction in displacement.

<table>
<thead>
<tr>
<th>Specification of the optimal design</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Module</td>
<td>1.085 mm</td>
</tr>
<tr>
<td>Number of teeth</td>
<td>30</td>
</tr>
<tr>
<td>Drive pressure angle</td>
<td>6.5°</td>
</tr>
<tr>
<td>Coast pressure angle</td>
<td>5.0°</td>
</tr>
<tr>
<td>Pitch radius</td>
<td>16.28 mm</td>
</tr>
<tr>
<td>Addendum Radius</td>
<td>18.02 mm</td>
</tr>
<tr>
<td>Root Radius</td>
<td>14.30 mm</td>
</tr>
<tr>
<td>Facewidth</td>
<td>27 mm</td>
</tr>
<tr>
<td><strong>Max. Displacement</strong></td>
<td>11.5 cc/rev</td>
</tr>
<tr>
<td><strong>Min. Displacement</strong></td>
<td>7.1 cc/rev (62%)</td>
</tr>
</tbody>
</table>

![Figure 8. Optimal profile of the asymmetric gear, B, with 30 teeth](image)

HYGESIM simulations predicting the performance of the design of Figure 8 (in pumping mode) at max. and min. displacement are shown in Figure 9. As clearly shown in the figure, the flow delivered at min. displacement is 62% of the one at max. displacement. HYGESIM predictions for the input shaft torque in the same operating conditions are shown in Figure 10. It can be noticed that the required torque at min displacement is about 62% of that required at max displacement. This resulting reduction in torque is due to the lower total forces acting on the gears as a result of moving the slider towards the inlet side. The predicted lower value of the torque at min displacement shows a reflection on the lower energy consumption.
at min displacement thus proving the concept of a new EGM which has the feature for varying the displacement.

5 CONCLUSION

This paper presents an investigation study on a novel design concept for variable delivery flow external gear machines previously introduced by the authors. The proposed design preserves the typical advantages of external gear machines, and it is based on a simple principle of variable timing of the connections of the internal tooth space volumes with the inlet and outlet ports. This principle is implemented through the use of an additional sliding element (“slider”) placed at gears’ lateral sides. Unconventional asymmetric gears are used to permit a significant range of variation of the displacement.

While in a previous study the authors tested a design capable of achieving a flow variation of 32% (from 68% to 100% of delivery flow), which was implemented in a commercial pump utilized as reference; in this study an analysis of the solution without specific constraints was performed to understand the full potentials of the proposed solution. The optimal design of the gears and the grooves on the slider were identified by using a multi-level, multi-parameter and multi-objective optimization process formulated on purpose for this research. Considered objective of the procedure were: the maximization of flow variation range, the minimization of the flow pulsation, the minimization of the internal pressure overshoots and localized cavitation, the maximization of the volumetric efficiency.

The results of the optimization show that a higher flow variation range is achievable by using a higher number of teeth, and for example a 38% flow variation is realizable with a 30 teeth design. Results show a torque reduction in accordance to the flow reduction, confirming performance similar to other variable displacement positive displacement machines currently used in fluid power applications.

The proposed design concept has potentials to improve the energy efficiency (and therefore the fuel consumption) of current systems based on economical fixed displacement units (such as EGMs) by adding an cost effective solution for realizing flow-on-demand features. However, this range of improvement is directly related to the range of flow variation guaranteed by the proposed design. Although a range in the order of 38% would be of tremendous impact in applications such as charge pumps for hydrostatic transmissions, hydraulic fan drives and fuel injection systems, a higher range has to be researched in future, by investigating the use of novel unconventional profiles (such as multi-involute, or shark fin shaped profiles) suitable for EGM.
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NOMENCLATURE

\[ D \]
- With respect to gears: the point at which the trapped tooth space volume begins to exist
- With respect to grooves: vertical distance of the delivery groove from the line joining the inter-axis of the two gears [mm]

\[ H \]
Horizontal dimension of the delivery and suction grooves [mm]

\[ P \]
Pressure in TSV [bar]

\[ P_{D,avg} \]
Average pressure at the delivery [bar]

\[ P_{sat} \]
Saturation pressure of the operating fluid [bar]

\[ Q_{avg} \]
Average delivery flow rate [lpm]

\[ S \]
- With respect to gears: the point at which the trapped tooth space volume seizes to exist
- With respect to grooves: Vertical distance of the suction groove from the line joining the inter-axis of the two gears [mm]

\[ Z \]
Number of teeth

\[ k \]
Index for the fundamental frequency of the FFT

\[ m \]
Normal module [mm]

\[ \dot{m} \]
Mass flow rate [kg/s]

\[ r_a \]
Addendum radius [mm]

\[ r_0 \]
Pitch radius [mm]

\[ r_b \]
Base circle radius [mm]

\[ r_r \]
Root circle radius [mm]

\[ r_{bd} \]
Drive side base circle radius [mm]

\[ r_{bc} \]
Coast side base circle radius [mm]

\[ t \]
Tooth thickness of the gear cutter [mm]

\[ \alpha_{DL} \]
Angular position of the left wing of the delivery groove [°]

\[ \alpha_{DR} \]
Angular position of the right wing of the delivery groove [°]

\[ \alpha_{SL} \]
Angular position of the left wing of the suction groove [°]

\[ \alpha_{SR} \]
Angular position of the right wing of the suction groove [°]

\[ \alpha_0 \]
Pressure angle [°]

\[ \alpha_a \]
Pressure angle at the addendum circle [°]

\[ \alpha_{0d} \]
Drive pressure angle [°]

\[ \alpha_{0c} \]
Coast pressure angle [°]

\[ \beta \]
Percentage displacement of VD-EGM [%]

\[ \eta_v \]
Volumetric efficiency [%]

\[ \rho \]
Density of the operating fluid [kg/m³]

\[ \rho_r \]
Root fillet radius of the rack cutter [mm]

\[ \vartheta_G, \varphi_G \]
Involute co-ordinate parameters [°]

\[ \nu_G, \xi_G \]
Trochoid co-ordinate parameters [°]
REFERENCES


LEAKAGE PAST ACTIVE CONTACTS IN INVOUITE AND CYCLOIDAL
GEAR HYDROSTATIC UNITS

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ABSTRACT

Patterns of leakage flow through the ‘active’ contact, which separates a high pressure zone (HPZ) or chamber from its adjacent low pressure zone (LPZ) or chamber, in gear hydrostatic (HST) units, are analyzed using CFD method in Ansys Fluent ® environment. Two type HST units- one is the common involute external toothed gear unit and the other is the form closed ‘epitrochoid (cycloid class) generated gear’- namely GEROTOR and ORBIT units, are considered for analyses. An earlier photo imaging evidences, by another research group, exhibiting not only the leakage but other relevant phenomenon such as possibility of cavitation, back flow etc., have evoked the present CFD analyses for theoretical backup of the experimental results. Essentially pressure buildups are also considered. Rigorous analyses were carried out for estimating the geometric shape of leakage paths, which involved gear geometry, kinematic, contact deformations etc. CFD results have good agreement with the earlier experimental results and confirm the usefulness in predicting the leakage phenomenon at the transition contact zones of considered HST units.

KEYWORDS: Leakage, Gear HST units, Active/Transition Contact, Photo imaging, CFD, Ansys Fluent.

1. INTRODUCTION

Gear hydrostatic units i.e., hydraulic pumps and motors, posses reasonably very good performance characteristics, although have relatively low volumetric efficiencies in comparison to vane and cylindrical piston type hydrostatic units. Apart from the side leakages through the passages between the rotary gears and the fixed casing walls, leakage past the contact regions, specially through the ‘active contact’, which separates a high pressure zone (HPZ) or chamber from its adjacent low pressure zone (LPZ) or chamber, has significant share in total leakages.

In a recent investigation report on visualization of the flow processes in involute external gear pumps and cycloidal internal (GEROTOR) gear pumps, Stryczek et al. [1] have presented clear evidences of leakage flow past the active/transition contacts between tooth pairs [Figure 1]. They used a very high speed camera to capture the flow phenomenon while the original cover or side plates of the pumps were replaced by thick Perspex plates for transparent views. Different operating conditions were studied, although the fluid pressure could not be raised to the higher limits of the hydrostatic units to keep the deflection of Perspex cover plate to a reasonably lower value. They also carried out extensive geometric and kinematic analyses to find out the leakage path and probable reasons of flow pattern. Pressure rise in trapped volume [2] could be the
source of such leakages. For optimization of pressure balance and reduction in leakages they also considered relive grooves. However, fluid dynamics were not studied.

![Image](https://example.com/image1)

(a) Involute Toothed Gear Pump  
(b) Epitrochoidal & its Envelope Toothed GEROTOR Pump.

Figure 1. Images of Leakage Past the ‘Active/Transition’ Contact.

![Image](https://example.com/image2)

Figure 2: Involute External Toothed Gear HST Unit.

The flow pattern in the suction chamber of the gear pumps were also studied both experimentally by N. Ert’urk et al. [3]. Theoretical backup was also developed by R. Castilla et al. [4]. However, leakage past the transition contact was not considered in their studies.

The generation of gaps at active / transition contacts depends both on fittings (i.e., tolerances and manufacturing errors) as well as on deformations due to the load and other factors such as thermal, dynamics factors etc. In case of involute toothed gear hydrostatic units, backlash has a greater contribution in such gaps. On the other hand contact deformations are mainly responsible in case of GEROTOR (also ORBIT) units which are basically ‘geometrically form closed’ in nature. Methods of estimation of contact stresses, deformations and gaps in such machines are well established [5, 6, 7]. However, leakage analyses are not reported.

2. CFD MODELLING AND ANALYSIS

We have used CFD analyses in Ansys Fluent ® [8] environment (with two dimensional, double precision, pressure based, laminar flow solver), to establish theoretical backups of flow phenomenon past the transition contacts and quantify the amount of leakages for both involute and cycloidal toothed hydrostatic units. We have studied only pump models with respect to the prime reference [1] of this investigation report. Some of the relevant data of gears, generated gap, hydraulic oil etc. were assumed as detail data were not available. Comparing with the available photographs (images) matching geometric shapes of gears were developed. However, no gear corrections were considered. The sizes of gaps at transition contacts were assumed based on available research reports and literature.

2.1. Involute External Toothed Gear HST Unit.

Working principles of involute of similar toothed external gear HST units namely- pumps and motors (Figure-2) are well established. The ‘active’ or ‘transition’ contact of the mating gears separates suction chamber from impulse chamber. These two chambers are in low (LPZ) and high (HPZ) pressure zones. They are also called inlet and outlet chambers. Depending on pump or motor features either of the inlet or outlet is LPZ or HPZ. There is no separate valve in such units. When two pairs are in contact momentarily these two contacts entrapping a small amount of fluid subjected to vary in plane area bounded by teeth profiles between two contacts. This causes squeezing of entrapped volume of fluid and pressure rise above the system pressure. Added to the general leakage tendency from HPZ to LPZ, through the active contact zone, such squeezing
increases the leakage. The gap at active contact is usually due to backlash and generated by the pressure force separating the metal to metal contact.

2.1.1. : Profile Geometry and Defining Leakage Path

First of all we have chosen three frames of images showing leakage (Figure 3). Matlab programming technique is used to develop the involute tooth shape for gears [9]. For root trochoid we have simply considered a fillet radius. A few trials and errors in selecting gear parameters were needed to get the tooth profile matching with close to the profile shape in images. However, we have not considered any tooth modification except the profile shifting. The profiles of mating gear teeth were developed by transforming the data of the first gear with respect to their respective centres and the necessary rotation of mating gear for contacts in the line of action. Then both gears were given necessary rotations for matching the profiles of teeth in contact with those in image frames. Next the gap was provided by infinitely small rotation of one of the mating gear as if there is some backlash. This needed another set of transformation of gear data. We fixed the amount of gap measuring the distance of gap at the contact point along the line of action. Now the profile data of only required portion of mating gears were stored in a separate file for feeding into the Ansys Fluent® programme for CFD analysis. Figure-4 shows such development of gear tooth profiles.
2.1.2. Fluent Modelling

Defining flow path in Fluent: The stored data (co-ordinates) defining the profiles of gear pair of a particular frame is now fed into Ansys Fluent in the prescribed format of the software. Boundaries of the surface are defined for the purpose of assigning boundary conditions. They are maked as 1 to 8, as shown in the Figure 6(a). Typical mesh generation adjacent to the transition contact C is shown in Figure 6(b). Quadrilateral method of meshing is used. Initially an element size is selected. Gradually the size is reduced, along with variation in some other parametes such as method of meshing, size of the element and relevant centre to get fine mesh around the contact point etc., untill the result becomes satisfactory and remained almost unchanged with even finer elements. In this case the element size was 0.00001m. In ‘cell zone’ conditions, surface body was defined as a fluid and the oil option was selected.

![Diagram](image)

(a) Boundaries. (b) FEM Mesh Generation.

Figure 6 : Configuration of Leakage Path for CFD Calculation in Fluent.

Generation of Gap at Transition or Active Contact: In the case of involute external toothed gear HST units the inlet and the outlet chambers are separated by one or two transition contacts (marked T in Figure 3) depending on angular positions of the mating gears. This is related to contact ratio which is usually not less than 1.4. These transition contacts are with positive contact forces. Contacts in opposite flanks (Marked C in Figure 3) are not force closed and even with gap i.e., no contact as there is backlash. Frame 3, as in Fig. 3(c) is having double transition contacts. Although duration of such double contacts is about 1/3rd. Times of a chamber action cycle but the contraction of the trapped volume of fluid in between the contacts occurs. This is inherent and results in pressure rise. The variation in trapped volume with the rotation of gear can be estimated following the method proposed by Manring and Kasaragadda [10]. Difference in volume for two contact points is the trapped volume. Hence it is possible to estimate the squeezing and the instantaneous pressure rise in the trapped volume. Considering ideal case the pressure rise will be very high. However, such pressure rise is limited to twice the HPZ pressure and only for a time period of 10% of total cycle time (One cycle is one by number of teeth time of a revolution), may be considered as realistic values.

Boundary Condition: The defined boundaries are named as Fluent Identity (Figure 6). Pressure values are assigned at inlet and outlet.
2.1.3. Fluent Results

We have considered a maximum pressure difference of 9 MPa between HPZ and LPZ or more precisely boundaries marked 3 and 8 in Figure 4. The CFD analysis results using Fluent, are presented in Figure 7.

![Flow by FLUENT](image1)

<table>
<thead>
<tr>
<th>Flow by FLUENT</th>
<th>Pressure by FLUENT</th>
<th>Experiment (Involute tothed Gear Pump)</th>
</tr>
</thead>
<tbody>
<tr>
<td>FRAME 1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>FRAME 2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>FRAME 3</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Figure 7**: Visualisation of Flow past Active Contacts In External Involute Toothed Gear Pump.

2.2. Cycloidal Toothed HST Unit

Cycloidal toothed HST units are essentially internal gear units (Figure 8). Usually Constant difference modified Epitrochoid and its Envelope constitute ‘Star’ and ‘Ring’ members respectively [6]. The active lobes of ring are circular arcs and may be integral or separate rollers in partial bushes cut in the ring. Ideally all lobes are in contact with the star lobes and therefore, the contacts are form closed. Such active contacts moves along the ring lobe profile over $2\phi_{\text{max}}$ (leaning) angle and on whole star profile with the rotational motion of star and ring in fixed axis units or in units with either of them in floating axis. The area bounded by star and ring profiles between two consecutive active/transition contacts varies with motion. This variable area and the uniform thickness of the star-ring form the chamber. In these rotary piston machines
(ROPIMAs) the adjacent chambers are separated by metal to metal higher pair contacts which are very prone to inter-chamber leakages.

Depending on the kinematic arrangement such machines may of fixed axes or may be with orbital motion of either of star or ring (usually the star) when the other member is fixed. The fixed axis type is used as both pumps and motors, which are regarded as High Speed Low Torque (HSLT) HST units and named as GEROTOR units. The conventional kidney port is used in these machines. On the other hand with epicyclic kinematic motion the unit works at Low Speed and High Torque (LSHT), used as motor only, and called as ORBIT Motor. Naturally each active contact has to take certain amount of load depending on fluid pressure i.e., torque. A few active contacts get separated when others are balancing the torque and being deformed due to load. The generated gap at active contacts may occur even at a situation when the active contact is separating a high pressure chamber (HPZ) from a low pressure chamber (LPZ).

The leakage through the generated gap at active contacts is undesired. However, elimination of such gaps is practically impossible although could be optimized. Therefore, in estimation of leakage, the flow visualization [1] through imaging is certainly an essential step. However, interpretation and quantifying such leakages would be the next step which is initiated through the CFD analysis in the present investigation.

2.2.1. Profile Geometry and Defining Leakage Path

The design methods, generation of gap etc. are already investigated earlier by the present research group and reported [6, 7, 11-13]. As the actual geometric and few relevant data were not available for the present analysis those are assumed and with trial and error the generated profile were matched with the video images. However, a typical set of results on an ORBIT unit are presented in Figure 10 and Figure 11. The leakage was identified considering the gap at critical situation. Relevant coordinates of the profile fed into Fluent.

2.2.2. CFD analysis by Fluent

Assigning boundaries and boundary conditions: Unlike the external involute (or involute like) tooth gear HST units in GEROTOR and ORBIT units the entrapment of volume of fluid during phase change depends on flow distributor valve features and timings. In the present analysis we have considered only the ideal case, i.e., no entrapment of volume considering zero-lap during phase change. Therefore, the adjacent chambers of opposite phase are having simply inlet and outlet pressures.
Estimation of Gap and Phases: Maiti and Nag [6, 7] have shown earlier how the gap and the orientation of Star-Ring lobes are calculated in such active and critical contacts, which separate the adjacent chambers in opposite phases. The process of calculations is very tedious as estimating stresses and deformations at all active contacts is statically indeterminate problem. It is to be noted that in the case of ORBIT unit the torque and the pressure load equilibrium is achieved by reaction forces at active contacts i.e., star lobe to ring lobe contacts. This is because the start in ORBIT unit is practically of floating in nature. On the other hand in the case of GEROTOR unit the shaft driving the star also takes load. However, if the shaft and the star is coupled through spline then it may behave more or less as ORBIT unit as far as the load sharing by contacts at lobes are concerned. The magnitude of reaction loads will differ for same sizes of star-ring used for ORBIT unit and GEROTOR unit working at same the pressure.
Table 3. Star-Ring and Gap at Critical Contact
(Lobe-4, Ref Fig. 9 to Fig. 11)

<table>
<thead>
<tr>
<th>Distance from gap in downstream direction (m)</th>
<th>Gap in downstream direction (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>9.32E-07</td>
</tr>
<tr>
<td>0.00708791</td>
<td>6.11E-05</td>
</tr>
<tr>
<td>0.01417365</td>
<td>2.47E-04</td>
</tr>
<tr>
<td>0.02125505</td>
<td>5.57E-04</td>
</tr>
<tr>
<td>0.02832994</td>
<td>9.91E-04</td>
</tr>
<tr>
<td>0.03539615</td>
<td>1.54E-03</td>
</tr>
<tr>
<td>0.04245152</td>
<td>2.23E-03</td>
</tr>
<tr>
<td>0.04949388</td>
<td>3.03E-03</td>
</tr>
<tr>
<td>0.05652109</td>
<td>3.96E-03</td>
</tr>
</tbody>
</table>

Fig 12: Mapping of Gap at Critical Contact Point.

Fig 13: Visualisation of Flow past Active Contacts in ORBIT and GEROTOR unit.
Computation and CFD Results: We have considered a differential pressure of 5 MPa between the inlet and outlet of the detected leakage path and gaps as stated in Table-3. Oil characteristics considered same as in case of involute gear HST units (Table-2). Flow visualizations are shown in Figure 13.

3. SUMMARY

CFD Fluent modelling for the visualization of flow through the leakage paths in the transition contacts in involute type external gear HST units and cycloidal gear HST units are established. The results have reasonably good agreement with the experimental visualisation of such flows through high speed photo imaging by another research group. To adapt the present proposed analytical method, thorough knowledge of gear geometry and kinematics are required to define the leakage path and different angular positions of gear set. In case of cycloidal gear units the method of estimating the gap at transition contact is also focused. The CFD analysis part using Fluent in Ansys is simple although requires carefullness in data interpretation at the input of Fluent tool. CFD results present many information which may be useful for predicting the leakage flow rate, cause of cavitation etc. This analysis certainly can be extended to optimize/reduction, if not elimination, of those.

ACKNOWLEDGEMENT

Prof. J. Stryczek and Dr. P. Antoniak of Wroclaw University of Technology (WUT), Poland, are gratefully acknowledged for providing the image clips and few required data, and valuable discussions. Also, Mr. U. Mukherjee, current postgraduate student at IIT, Kharagpur is working in the same direction for his masters thesis under the supervision first author, is acknowledged for his help in confirming some CFD results in recent times.

REFERENCES


Appendix-1

Figure A1. Gear Geometry for Volume Calculation.

Referring to the contact point (marked circled in red in Figure A1), we get instantaneous flow rate as follows.

\[
\hat{Q}_d = \frac{1}{2} \left\{ \hat{r}_{a_1}^2 + \hat{r}_{a_2}^2 \left( \hat{r}_{p_1} / \hat{r}_{p_2} \right) - \hat{r}_{p_1} \left( \hat{r}_{p_1} + \hat{r}_{p_2} \right) - \left( 1 + \hat{r}_{p_1} / \hat{r}_{p_2} \right) \hat{\ell}^2 \right\}
\]

Method of calculating \( \hat{\ell} \) is established by Manring and Kasaragadda [10].

Now this flow rate is also calculated for the next contact in line of contact. The difference of these two flow rates is the squeezing or expanding rate of the trapped volume.
SIMULATION ANALYSIS OF RING GEAR’S MICRO MOTION IN INTERNAL GEAR MACHINES

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ABSTRACT

This paper presents a new modeling approach for studying the oil film characteristics between the ring gear and the case in internal gear machines. In traditional research, the assembly relationship of ring gear and case bore was considered as an ideal cylindrical pair, and the micro motion of ring gear and its influence on oil film height and pressure had not been discussed yet. However, wedge oil film between the ring gear and the case is actually formed due to the pressure acting on ring gear teeth while Internal Gear Machines are running, causing micro motion of the ring gear. The model presented mainly consists of two parts namely the gears meshing model that deals with the pushing force and the oil film model that deals with the supporting force. The gears meshing process is numerically modeled based on the characteristics of the involute, after which the pushing force can be precisely integrated. The oil film pressure field is analyzed based on the lubrication theory of the oil film under different oil film height field, after which the supporting force is precisely integrated. The supporting force is important for balancing the pushing force, being responsible for functions of sealing and lubricating. Consequently, the simulation model is implemented with MATLAB and running in co-simulation to obtain the resultant force of the ring gear, which should stay at its minimum value to achieve of the goal of minimizing shear stress and preventing excessive wear. Finally, the dynamic characteristics of the ring gear’s micro motion and pressure distribution are analyzed.

KEYWORDS: Internal gear machines, Ring gear, Oil film, Hydrodynamic lubrication

1. INTRODUCTION

Internal Gear Machines, IGMs, are the preferred solution to the fluid power industrial field demanding for high control accuracy, low noise level, and compact structural design, such as injection molding machines, hydraulic servo systems, and marine hydraulic systems. Factors of success of this kind of hydrostatic unit are
presented by their low flow ripple, low noise level and high power to weight ratio. Consequently, research on IGMs to improve its performance and working life is to be of great importance.

The ring gear/case interface of IGMs represents one of the most critical design elements of the machines’ lubrication pairs. The oil film is ideally considered as uniform distribution for the ring gear and the case since the assembly is considered as an ideal typical cylindrical pair.

Wedge-shaped oil film is actually formed between the ring gear and the case, which varies with the time-changing forces acting on the ring gear. The velocity gradient of the oil film is therefore changing as the thickness of the oil film varies, and supporting force is generated due to hydrodynamic effect[1].

The micro motion of the ring gear is not taken into consideration as there have been very few attempts on the oil film model of the IGMs. Andrea Vacca, et al, had built a simulation tool named HYGESim to help understand and evaluate the performance of external gear machines (EGMs)[2][3]. The lubricating gap between the lateral bushings and the gears in EGMs was discussed with CFD taking fluid–structure interaction into consideration. In his research, not only the axial micro motion of the lateral bushings, but also the angle of the tilt of the lateral bushings was taken into consideration. Experimental prototype had been built to validate the model and optimize the parameters of the lateral bushings[4][5][6]. Formulation of typical type of lubrication theory and experimentation methods had been used by Hooke C., et al, to examine the mechanism of lubrication and the clearance between the floating plate and the gears in EGMs[7][8][9]. It showed that the hydrodynamic load is greatly affected by the non-flatness and gear tilt which should meet the requirement to maintain the axial balance of the bushes. Clearance between the floating plate and the gears had been optimized with his research[10].

A numerical simulation model was presented in this paper based on the theory of the meshing gears and the theory of the oil film lubrication. Pressure acting on each tooth was integrated by modeling the meshing process so as to get the pushing force precisely. The effects of the ring gear’s micro motion on the thickness of the oil film were taken into account and pressure distribution was displayed to assess the supporting force as a feedback of the pushing force.

2. SIMULATION MODEL

The structure of the simulation model is displayed in Fig. 1. As presented in Fig. 1, the simulation model mainly consists of two sub-models, the gears meshing model and the oil film model. The two sub-models are implemented with MATLAB and running simultaneously in co-simulation.

The following sub-sections describe in detail the simulation model and their implementation with MATLAB.
2.1 Gears Meshing Model

Gear meshing model is the foundation of the simulation model. It begins with the profiles of the internal gear pairs based on the secondary generating method. As is described in the secondary generating method, the locus of the pinion is the encircle curve of the rack cutter. Due to the meshing principles, the internal gear should be conjugated to the pinion, which indicates its locus should be the encircle curve of the generating pinion. Thus, the mathematical model of the pinion as well as the ring gear can be easily obtained.

Fig. 2 displays the generating process of the gear pairs. The cutter coordinate system \( x_tO_ty_t \) is rigidly connected to the rack cutter, which is supposed to describe the profile of the rack cutter. The distance from the pitch line to \( x_t \) axis is \( x_m \), while \( y_t \) axis lies in the middle of the tooth space.

Define one simple equation which might help express the profile of the rack cutter as follows.
Thus the mathematical model of the rack cutter is easily obtained which is helpful for MATLAB programming.

\[
sgw(x) = \begin{cases} 
1, & x \geq 0 \\
0, & x < 0 
\end{cases}
\]

(1)

(b) depicts the generation process of the pinion. The cutter coordinate system \(x_tO_ty_t\) is rigidly connected to the rack cutter whose \(x_t\) axis is tangent to the pinion’s pitch circle. The pinion coordinate system \(x_1O_1y_1\), is rigidly connected to pinion whose origin lies at the center of the pinion and \(x_1\) axis parallel to the \(x_t\) axis. As is known, the cutter coordinate system takes translational action along the cutter while the pinion coordinate system takes rotational action as the pinion rotates. Therefore, the mathematical model of the pinion can be derived by transformation of coordinates.

\[
R_1(\phi_1, x_t) = \begin{bmatrix}
\cos \phi_1 & \sin \phi_1 & -r_1(\phi_1 \cos \phi_1 - \sin \phi_1) \\
\sin \phi_1 & \cos \phi_1 & r_1(\phi_1 \sin \phi_1 + \cos \phi_1) \\
0 & 0 & 1
\end{bmatrix}
\]

(2)

(c) depicts the generation process of the ring gear. The pitch circle of the pinion is tangent to the ring gear’s. The ring gear coordinate system \(x_2O_2y_2\) is rigidly connected to ring gear whose origin lies at its center and \(x_2\) axis parallel to the \(x_t\) axis. The pinion is taken as the generating cutter to generate the ring gear based on the secondary generating method. Consequently, the mathematical model of the ring gear can be derived by transformation of coordinates.

\[
R_2(\phi_1, \phi_2, x_t) = \begin{bmatrix}
\cos(\phi_1 - \phi_2) & -\sin(\phi_1 - \phi_2) & (r_2 - r_1) \sin \phi_2 \\
\sin(\phi_1 - \phi_2) & \cos(\phi_1 - \phi_2) & (r_2 - r_1) \cos \phi_2 \\
0 & 0 & 1
\end{bmatrix}
\]

R_1(\phi_1, x_t)

(3)

The profile of the internal gear pairs together with the filler parts can be depicted with MATLAB (Fig. 3).

(a) displays physical analysis of the gear pairs, the gear shaft, the ring gear and the filler. Three different pressure zones are divided, the Low pressure area, the transitional pressure area and the high pressure area\(^{[11]}\). The pressure in the high pressure area equals to the outlet pressure \(P_o\) while the pressure in low pressure area equals to the outlet pressure \(P_i\). The pressure in the transitional area shows a linear decrease between the adjacent tooth spaces.

The pressure zones need to be accurately calculated to get the hydraulic force acting on the ring gear. The sectors \(H_1C_01\) and \(H_2L_01\) are the gear shaft’s high pressure area and transitional pressure area while the sector \(H_2C_02\) and \(H_2L_02\) are the ring gear’s. Low pressure angle is not discussed here for it has little influence on the result.
As is known, one or two contact points are to be formed during the meshing process. When two contact points are formed, the new one is neglected when calculating the high pressure sector angle since the volume trapped between the new and the old under high pressure. However, when the old one has just disappear, dividing point between the high pressure area and the low pressure area jumps from the old one’s position to the new one’s position, thus causing a sudden decrease of the high pressure sector angle, which results in a sudden decrease of pushing force.

(a) Physical analysis  
(b) MATLAB description

1 Radial unbalance force; 2 Supporting force; 3 Oil film; 4 High pressure area; 5 Transitional pressure area; 6 Low pressure area; 7 Gear shaft; 8 Ring gear; 9 Case; 10 Filler; P1: Contact begins; P2: Contact ends; C: Contact point; L: Low-pressure boundary point; H: High-pressure boundary point; O1: Centre of Gear Shaft; O2: Centre of Ring Gear

Figure 3. Generating process of the gear pairs

(b) depicts the MATLAB description of the physical shape, in which the contact point C, the pressure division point H and L can be accurately achieve based on the coordinates relationship.

For precise calculation of force acting on the teeth, simple cylindrical assumption method is no longer applicable. Only part of the tooth is under high pressure on the tooth with meshing point. While staying in transitional pressure zone, pressure acting on different tooth space varies with tooth space itself. Therefore, it is necessary to analyze the force acting on the single tooth space.

A precise evaluation of forces acting on the single tooth space should consider the characteristics of the involute gear (Fig. 4). Take a random point A on the tooth surface, on which the force acting is the normal direction. As is known from the characteristic of the involute, the normal direction is tangent to the base circle on point B. Assuming the involute angle of point A is \( \theta \), and pressure angle is \( \alpha_k \) with generating line being \( s_k \). The angle formed by line OB and y axis is \( \beta \).
Force acting on point A can be expressed as follows.

\[
\begin{align*}
    dF_x &= PR_b \left( \tan \alpha_k \right) \cos \left( \tan \alpha_k - \delta \right) \frac{1}{\cos^2 \alpha_k} \\
    dF_y &= PR_b \left( \tan \alpha_k \right) \sin \left( \tan \alpha_k - \delta \right) \frac{1}{\cos^2 \alpha_k}
\end{align*}
\]

The evaluation of forces acting on the ring gear can be done by numerical integration derived from Eq.5.

2.2 Oil Film Model

Oil film model deals with the supporting force, which consists of two parts: the high pressure chamber supporting force and the sealing area oil film supporting force.

Fig. 5 depicts sectional view of the case. As is shown, there are two high pressure chambers, in which locate two outlets. Therefore, the pressure in high pressure chamber equals to outlet pressure $P_o$. Each high pressure chamber is surrounded sealing area, and the sealing area is surrounded by low pressure area, whose pressure equals to inlet pressure $P_i$.

Unwrap one of the sealing areas and it is depicted in Fig. 6. It is a rectangular sealing area, with high pressure chamber in the middle and low pressure in the border. The transition area around the middle is sealing area. The pressure in the high pressure chamber $P_o$ diminishes to the inlet pressure $P_i$ along from the middle to the border.
Take two cross sections of the ring gear and the case at two axial positions M-N and M-N in Fig. 5 depicted in Fig.7.

At position M-M, both sealing area and high pressure chamber are included. Point ABCD means the beginning and end of the sealing area with arc AD being the sealing area. The supporting force consists of two parts, namely the pressure produced by the high pressure chamber in the middle and the supporting force by the oil film around the middle. Arc BC means the high pressure chamber, with its sector angle being $\beta$. The sector angle of sealing arc AB is $\alpha_1$, while the sector angle of sealing arc CD is $\alpha_2$.

At position N-N, only sealing area is included. The whole arc AD is the sealing area.

The tilting state of the ring gear in the case in an exaggerated way is represented in Fig. 8. Since the ring gear stays at a tilting state, the eccentric positions of both the front end and the back end have to be expressed, thus four eccentric parameters have to be induced.

The tilting state of the ring gear can be well defined by the eccentricities of the ring gear’s both ends relative to the axis of the case. In the front end plane, taking the center of the case $O_1$ as the origin, the coordinates...
$O_{12} (e_{11}, e_{12})$ of defines eccentric position of the front end. So is the back end.

Figure 8 Tilting state of the ring gear

Assuming that the sections of the ring gear and the case are both ideally round and the axes are both ideally straight. Point H is a random point in line $O_{12}O_{22}$ whose distance to point $O_{12}$ is $l_{az}$. Consequently, the oil film height can be well approximated.

$$h = r_e - r_r - \left( e_{11} - \frac{l_{ac}}{l_{e}^{\gamma}} (e_{11} - e_{21}) \right) \cos \phi - \left( e_{12} - \frac{l_{az}}{l_{e}^{\gamma}} (e_{12} - e_{22}) \right) \sin \phi$$

The pressure field of the sealing area needs precise calculation to integrate the supporting force. Since the oil film height is much less than the other two dimensions, the flow in the sealing area can be treated as laminar. Therefore, the Reynolds Equation to express the pressure field in differential form can be written as follows.

$$\frac{\partial}{\partial x} \left( \frac{\partial p h^3}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{\partial p h^3}{\partial y} \right) = 6(v_x \frac{\partial h}{\partial x} + v_y \frac{\partial h}{\partial y} + 2 \frac{\partial h}{\partial t})$$

Where, $v_x$ and $v_y$ are the oil film velocity caused by the ring gear’s velocity. Thus, it may be expressed as follows.

$$\begin{cases} v_x = 2\pi nr \\ v_y = 0 \end{cases}$$

Eq. (7) is a partial differential equation that can be solved numerically with well-known discretion method. The first two terms on the right hand side are relative to hydrodynamic effect of wedge-shaped oil film, while the last term is relative to squeezing effect, which is not taken into consideration here. As discussed above, since the oil film height is much less than the other dimension, the 2D finite volume method (FVM) is
Get double integral of Eq. (7) and the discretion form can be written in matrix from which help understanding and programming.

\[
\text{sum} \begin{bmatrix} \Delta y & 0 \\ 0 & \Delta x \end{bmatrix} \begin{bmatrix} \Delta y & 0 \\ 0 & \Delta x \end{bmatrix} = \text{tr} \begin{bmatrix} \Delta y & 0 \\ 0 & \Delta x \end{bmatrix} \begin{bmatrix} \Delta y & 0 \\ 0 & \Delta x \end{bmatrix} + AD
\]

(9)

Where, AD is an additional item.

\[
AD = -12\mu \pi r (h_{i+1} - h_{i+1}) \Delta y
\]

(10)

\( h \) refers to the oil film height at the particular point while \( \Delta x \) and \( \Delta y \) refer to the dimensional length of the grid.

With boundary conditions that \( p=p_0 \) in the middle and \( p=p_i \) at the border, the pressure field of the sealing area can be solved numerically from Eq. (9).

2.3 Forces Analysis

As discussed above, the gears meshing model deals with the pushing force that plays the role as pushing the ring gear close to the case. On the contrary, the oil film model deals with supporting force produced by the high pressure chamber and the oil film that lays the role as pushing the ring gear away from the case. As the requirement of sealing and lubrication, the pushing force has to be bigger than the supporting force so as the
ring gear will be attached to the case. Whatever, the resultant force of the two should stay small so as not to cause excessive wear.\[12\]

The coupling relationships between the forces acting on the ring gear and its position will be dealt in co-simulation model. Apparently the pushing force varies with time in one period under one particular working condition. Due to the radial micro-motion of the ring gear, the oil film height field varies, and so is the supporting force. Therefore, the resultant force, of which the direction is towards the case, varies with the micro-motion of the ring gear. Under the adjustment of the resultant force, the ring gear will stay at its best position so as the resultant force remains the minimum one.

The simulation model is implemented with MATLAB for its specialization in matrix operation. The simulation procedure is represented in Fig.10.

![Diagram of simulation procedure]

**Table 1. Main simulation parameters**

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed (rpm)</td>
<td>2500</td>
</tr>
<tr>
<td>Pressure $p$ (Mpa)</td>
<td>25</td>
</tr>
<tr>
<td>Modulus (mm)</td>
<td>3</td>
</tr>
<tr>
<td>Tooth number of shaft gear</td>
<td>13</td>
</tr>
<tr>
<td>Tooth number of ring gear</td>
<td>19</td>
</tr>
<tr>
<td>Exradius of ring gear $r$ (mm)</td>
<td>39.5</td>
</tr>
<tr>
<td>Length of ring gear $L$ (mm)</td>
<td>55</td>
</tr>
<tr>
<td>Gap between ring gear and case (μm)</td>
<td>40</td>
</tr>
<tr>
<td>Sector angle of sealing area $\alpha_1$ (°)</td>
<td>36.81</td>
</tr>
<tr>
<td>Sector angle of sealing area $\alpha_2$ (°)</td>
<td>23.96</td>
</tr>
<tr>
<td>Sector angle of high pressure chamber $\beta$ (°)</td>
<td>113.09</td>
</tr>
<tr>
<td>Angle between the beginning of sealing area and x positive axis $\varepsilon_1$ (°)</td>
<td>-31.58</td>
</tr>
<tr>
<td>Angle between the beginning of sealing area and x positive axis $\varepsilon_2$ (°)</td>
<td>81.51</td>
</tr>
</tbody>
</table>

*Figure 10 MATLAB simulation procedure*
3. SIMULATION ANALYSIS

3.1 Simulation parameters

Descriptions provided in Table 1 clarify the inputs of the simulation model. The operation parameters, such as rotational speed and working pressure, are included. So are the structural parameters of the ring gear and the case. The value of the gap between ring gear and case comes from empirical study.

3.2 Simulation results

The simulation results are illustrated in the following 4 figures after running the program with parameters in table 1.

Figure 11. Pushing force and Resultant force

Figure 12 Eccentricities of Ring Gear

Figure 13. Pressure field of the sealing area at 6 positions
Conclusions can be made after analyzing the above 4 figures attentively.

(1) Since the number of the gears in the gear shaft is 13, the period repeats itself when the gear shaft rotates at every 27.7 degree.

(2) The pushing force acting on the ring gear ranges from 65 kN to 68 kN. An undershot occurs when rotation angle is 10° and 20° because of the sudden decrease of high pressure sector angle. The reason to the former is one of the orifice in the floating plate is covered by gears while the latter is the changing position of the meshing point. Since the former sector angle does not change a lot, the pushing force does not change a lot, neither. However, the latter sector angle does change a lot, the pushing force change a lot, too.

(3) The resultant force remains at 14.5 kN, and do not vary a lot. An overshoot occurs when rotation angle is 20°. However, the pulse is not much, which gives not much influence on the fluctuations of the resultant force. In that case, the pressure coefficient, the quotient of the pushing force divided by the supporting force, remains between 1.2 and 1.25. This is important for reducing the wear condition and extending the pump’s working life.

(4) The ring gear stays at an eccentric position. The eccentricities of front end do not stay the same as those of back end, which means the ring gear remains at a tilting and eccentric position. The biggest eccentricity is 18 micron, which makes 2 micron away from the wall case bore. Given the surface roughness of ring gear and case is 0.6 micron, there is still no metallic contact between the ring gear and the case, which gives confidence to the model.

(5) There are mainly six eccentric positions of the ring gear. Eccentricities of the front end mainly stay at (-18, -8) and (-14.5, -13.833) and the back end mainly at (14.5, 13.833) and (18, 8). A bouncing motion occurs.
when rotation angle is 20°.

(6) The eccentricities of the ring gear determine the oil-film thickness, consequently the pressure field. As is shown above, the micro-motion of the ring gear includes not only the vertical component but also the horizontal component.

(7) The pressure shows a gradual descent from the middle high pressure chamber to the border, and the gradient is relative to the oil film height. The former sealing area does not vary a lot with different positions while the latter one varies more. That is because the latter sealing area oil film height is thinner than the former one, and the pressure gradient is relatively higher.

4. CONCLUSIONS

(1) A simulation model that can predict the radial micro motion of the ring gear is proposed in this paper. The model mainly consists of two parts: the gears meshing model and the oil film model. Implemented with MATLAB, it runs in co-simulation, which is useful for optimization of the ring gear and case friction pair.

(2) The forces acting on the ring gear are well discussed in this paper. Key points, such as meshing point and dividing point, are obtained after numerical modeling of the internal gear pair in MATLAB. Pressure acting on the tooth surface is integrated based on the characteristics of the involute.

(3) The tilting positions of the ring gear are discussed in this paper. Reynolds equation is applied to the sealing area to simulate the pressure field that is relative to oil film height filed caused by tilting positions.

(4) The oil film height filed is illustrated by solving the coupling relationship between the force acting on the ring gear and its position. Thus the eccentricities ring gear is positioned, and the law of the ring gear’s radial micro motion is presented.

REFERENCES


ABSTRACT

Legged robots have not yet demonstrated the desired versatility and higher mobility that would justify their more complicated design with respect to wheeled or tracked vehicles. To make these robots ready for real world applications -- for example as assistants to humans in dangerous areas -- important challenges must be solved first, such as dynamic locomotion over rough terrain, dynamic balancing after disturbances, structural robustness to falls, self-righting (to get back up on the feet after falling), active or passive compliance in the legs, state estimation, perception and optional dexterous manipulation. In this paper we will focus on the robustness, self-righting and manipulation aspects. We will give an overview of the design of two new hydraulic robots: HyQ2Max, an improved, robust version of our hydraulic quadruped HyQ, and HyQ2Centaur, a centaur-style robot that combines the HyQ2Max locomotion platform with a pair of new hydraulic manipulator arms. We will focus on the self-righting ability of the quadruped robot and present the results of rigid-body dynamics simulations. Next, we will focus on the mechanical design concept of the new compact hydraulic arms and discuss the hydraulic actuation system. To the authors’ best knowledge this is the first time the design of a fully hydraulically actuated centaur robot is presented.

KEYWORDS: hydraulic actuation, hydraulic centaur, quadruped, legged robot, mechanical design

1. INTRODUCTION

Research into legged robots is expected to result in vehicles that are able to navigate with agility on rough terrain, exceeding the mobility of wheeled and tracked vehicles. However, despite the efforts of several decades of research into legged robots, the current state of the art is still far from reaching this goal. Recently a class of medium-sized, hydraulically actuated and torque-controlled quadrupedal robots (e.g. Boston Dynamics’ LS3 and BigDog [1] and IIT’s HyQ [2]) have shown promising results of agile navigation over flat and rough terrain and in presence of lateral disturbances [3]. Such robots are expected to assist humans for practical applications such as search and rescue, fire-fighting, forestry and inspection/maintenance tasks in dangerous areas or where automation is required in unstructured environments. Fundamental capabilities that such robots will need to have, are the following: dynamic locomotion over rough terrain, dynamic balancing after disturbances, structural robustness to falls, self-righting (to get back up on the feet after falling), active or passive compliance in the legs, state estimation and perception.
The HyQ project of the Istituto Italiano di Tecnologia (IIT) started in 2007 and resulted in the first version of HyQ in 2011, a fully hydraulic, torque-controlled quadruped robot [2]. Since then, HyQ has demonstrated a wide repertoire of static and dynamic motions ranging from walking trot over flat, inclined and rough terrain (indoors and outdoors), balancing under disturbances [3], flying trot [4], squat jumps, step reflexes [5], perception-enhanced trotting and crawling [6][7], to an optimized crawl gait for walking on stairs and stepping stones [8]. A summary video of these results is available online [9].

![Figure 1. Pictures of the first version of HyQ robot and leg. Left: HyQ robot on outdoor test track (2013); Right: first prototype of HyQ leg on vertical slider test bench (2008).](image)

Based on our experiences with HyQ and earlier leg prototypes (Figure 1), we have been developing a second version of the robot that improves upon the weaknesses of HyQ. It is our goal to develop a versatile machine that can be used for real-world applications. We expect that there will be situations in which the robot loses balance and falls. To allow the robot to continue with its operation, it is fundamental that it is robust to such impacts and that it can self-right after a fall. We therefore entirely redesigned the legs and the torso to increase the robot's robustness, extend its joint range of motion and increase its joint torque. This resulted in a new robot called HyQ2Max. Furthermore, we believe that a versatile quadruped robot used for real-world applications needs to have the option to mount a pair of dexterous arms to allow it to perform manipulation tasks. Therefore, we have been developing compact hydraulic manipulator arms that can be mounted on HyQ2Max, turning the quadruped robot into a centaur-like machine called HyQ2Centaur. A quadruped locomotion platform with two arms combines the stability and agility of four legs with the dexterity and functionality of a two-arm system.

This paper gives an overview of the design and hydraulic system of HyQ2Max and HyQ2Centaur. We will focus on the self-righting ability of the new quadruped robot and present the results of rigid-body dynamics simulations. To the authors' best knowledge this is the first time the design of a fully hydraulically actuated centaur robot is presented.

This paper first discusses the state of the art in the field of hydraulic quadruped machines and centaur-style robots. Section 3 then introduces the new hydraulic quadruped robot HyQ2Max, presenting the concept of its mechanical design, and the results of a simulated self-righting motion. Section 4 gives an overview of the design concept of HyQ2Centaur, focussing on the design of the new hydraulic arms. The hydraulic system of the robots is presented in Section 5. Finally, Section 6 discusses open problems and concludes the paper with final remarks.

2. RELATED WORK

This section discusses the state of the art in the field of hydraulically actuated quadruped robots and centaur-style robots.
2.1. Hydraulically actuated quadruped robots

Robotics research has resulted in a big variety of quadruped robots, most of them actuated by electric motors. A much smaller number is powered by hydraulic actuators. This section presents the most important examples.

In the 1960s, General Electric developed a four-legged walking truck that weighed over 1300 kg. It was hydraulically actuated and controlled by a human operator. Each limb of the operator was controlling one of the robot’s four legs through an interface with force feedback. After about 20 hours of training an operator was able to control the machine to walk, climb a stack of railroad ties and push a jeep out of the mud [10].

In the 1980s, Marc Raibert and colleagues constructed several hydraulically actuated legged robots, among which also a quadruped robot. The robot had four prismatic legs with 3 hydraulic joints each and a pneumatic spring at the end of the leg. It was able to trot, pace and bound on flat ground [11][12]. More recently, Raibert and his team at Boston Dynamics constructed several other hydraulic quadruped robots: BigDog [1], LS3, cheetah and wildcat. These robots clearly raised the bar of what is possible. However, very little information on the robot hardware, hydraulics and control has been published.

Shigeo Hirose’s Titan XI is a large size hydraulically actuated quadruped robot. The 7000 kg robot is designed for construction work on slopes. [13]. Statically stable walking on flat and inclined terrain has been experimentally demonstrated.

IIT’s HyQ robot is an 80 kg hydraulic quadruped that was first presented in 2010 in Claudio Semini’s PhD thesis [14]. Since 2011 the robot is fully torque controlled and has demonstrated a wide repertoire of motions ranging from highly dynamic motions to carefully planned navigation over rough terrain. For a more detailed introduction on HyQ see Section 1.

A number of hydraulic quadruped robots have been developed in Korea and China in the last years. For example, the P2 robot [15] and Jinpoong developed by KITECH, SCalf by Shandong University [16] and BabyElephant by SJTU [17].

2.2. Centaur-style robots

While humanoid and quadruped robots are very popular among researchers in the field of robotics, a combination of the two has rarely been investigated. This section presents the state of the art in the field of centaur-style robots.

The first known centaur robot was developed by a Japanese consortium of industry and universities from 1984-1993 as part of the ART project. The project was focussing on the development of several types of nuclear inspection machines, including an electric centaur-style robot [18]. A few years later, KIST presented their centaur robot with hydraulic legs and electric upper body [19]. The robot stood 1.8 meters tall and weighed 150 kg. More recently, Tsuda et al. presented a few papers on a small centaur robot that is actuated by electric RC servomotors [20]. Several other centaur-style robots were constructed with wheels at the end of their legs (e.g. WorkPartner [21], NASA centaur 2 [22]). Even though not a full centaur, it is worth mentioning that in 2013 a video of BigDog with one manipulator arm throwing a cinder block was published online [23].

3. HyQ2Max ROBOT DESIGN

This section introduces the design concept of the HyQ2Max robot, shows the results of a study on self-righting and presents an overview of possible future application scenarios of this robot.
3.1. HyQ2Max Design Concept

The HyQ2Max robot (Figure 2) is an improved version of the hydraulic quadruped robot HyQ [2]. The main improvements are increased reliability and robustness of the robot’s hardware, larger joint range of motion and higher joint output torque, as explained next.

Reliability and robustness against impacts and dirt are fundamental requirements for a legged vehicle performing real-world tasks. HyQ2Max is designed to be robust against impacts and dirt. All sensitive parts like sensors, valves, actuators and electronics are protected inside the structure. The torso is constructed with a frame made of a strong aerospace-grade aluminium alloy (7000 series), tubular roll frames in the front and back, and light-weight glass fibre/Kevlar covers that protect the onboard computer and hydraulics. The four legs are built of the same aluminium alloy as the torso. The upper leg consists of two rugged halves forming a shell that acts as protection and structural element. The lower leg is made of a light-weight yet robust aluminium tube.

The joints’ range of motion of a legged robot determines the size of the workspace of its feet. The larger this workspace, the more versatile motions can be implemented on the robot. A large workspace is especially important for self-righting motions as explained in Section 3.2. Table 1 compares the joint range of motion of HyQ and HyQ2Max and Figure 3 confronts the two different workspaces in the leg’s X-Z plane.

<table>
<thead>
<tr>
<th>Description</th>
<th>HyQ</th>
<th>HyQ2Max</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of actuated joints</td>
<td>12</td>
<td>12</td>
</tr>
<tr>
<td>Joint range of motion (HAA, HFE, KFE)</td>
<td>90°, 120°, 120°</td>
<td>80°, 270°, 160°</td>
</tr>
<tr>
<td>Peak joint torque (HAA, HFE, KFE) @ 20MPa</td>
<td>120Nm, 181Nm, 181Nm</td>
<td>120Nm, 245Nm, 250Nm</td>
</tr>
<tr>
<td>Upper, lower leg segment lengths</td>
<td>0.35m, 0.35m</td>
<td>0.36m, 0.38m</td>
</tr>
<tr>
<td>Robot weight (offboard power supply)</td>
<td>80kg</td>
<td>80kg</td>
</tr>
</tbody>
</table>

It can be clearly seen, that HyQ2Max has a larger foot workspace than HyQ, leading to (1) faster running since the step length can be increased, (2) self-righting ability since the leg can be moved completely up above the robot’s center of mass (see Section 3.2), (3) a rest position of the robot by retracting the legs until the bottom of the torso touches the ground and (4) an increased number of footholds for climbing motions with foothold planning.
Table 1 also shows that the HFE and KFE joints of the new robot have a higher joint output torque. This is important for self-righting (see Section 3.2), carrying payload and for more agile motions. The HAA joint is actuated by a double-vane rotary actuator, the HFE joint by a single-vane rotary actuator and the KFE joint by a cylinder connected to a four-bar linkage.

3.2. Self-Righting study

As mentioned in the introduction, the self-righting capability is fundamental for a real-world legged robot, since it is unavoidable that the robot falls during its operation on challenging terrain. We therefore implemented a self-righting sequence (see Figure 4) and simulated it inside our rigid body dynamics simulator SL [24]. All kinematics and dynamics calculations are implemented with efficient C++ code, automatically generated by the robot code generator RobCoGen [25].

The joint angle and torque plots of this simulation are shown in Figure 5. The different steps of the self-righting sequence are illustrated with different colours. The thin black lines show the limits of joint angle and torques as specified in Table 1. Note that the torque limits of the KFE joint depends on the KFE joint angle since the four-bar linkage creates a nonlinear torque output profile. For a detailed discussion on such output profiles, refer to [14]. The figure shows that all values stay inside their limits during the entire motion.
Figure 5. Simulation results of self-righting motion showing joint angles and torques for the left front (LF) and right front (RF) leg. The different colours indicate the different steps of the self-righting sequence. The black dashed line shows the joint angle and torque limits. **Left:** joint angle vs. time plots of the hip flexion/extension (HFE) and knee flexion/extension (KFE) joints. **Right:** joint torques vs. time plots of the same joints.

3.3. Possible future application concepts

HyQ2Max is designed to be the light-weight, high-performance version of this new four-legged vehicle. In the future, the robot’s hardware and configuration can be customized to match the requirements of the desired application. Figure 6 shows the concept of the robot applied to a range of possible future tasks. Task-specific features range from radiation-hardened hardware (e.g. nuclear decommissioning) to specific onboard sensing (e.g. inspection) and manipulation capability (e.g. maintenance, decommissioning). The next section will discuss our current efforts to add manipulation capability to HyQ2Max.

![HyQ2Max application scenarios. From left to right: construction, fire and rescue, forestry industry, inspection and maintenance, nuclear decommissioning.](image)

4. HyQ2CENTAUR ROBOT DESIGN

Future quadruped robots operating in real-world applications will most likely need to manipulate objects in the environment at some point, e.g. through a pair of dexterous arms. A centaur-style robot consists of a quadruped locomotion platform and a pair of arms. It thus combines the advantages of a stable four-legged base with the dexterity of a two arm system.

This section presents the design of HyQ2Centaur, which is a combination of HyQ2Max and a pair of arms. We will first give an overview of the design of a pair of custom-built, light-weight hydraulic arms that can be mounted onto HyQ2Max. Then we will present the concept of the centaur robot design and possible future application scenarios of the centaur robot.
4.1. Hydraulic arm design

The most important requirements for a dual arms system mounted onto a quadruped robot are (1) low total weight of arms including controllers, (2) compactness, (3) torque controllability, and (4) high joint speed and torque. Commercially available solutions are either too bulky because of their heavy base and controller units (Barrett’s WAM arm, KUKA’s lightweight arm), not torque controlled (Universal Robots UR5) and/or too slow (HDT Robotics’ MK1).

Due to this lack of commercial solutions, we have developed a compact arm with 6 hydraulic, torque controllable joints (Figure 7, left). The arm including all electronics and valves weighs around 13kg. The 6 degrees of freedom (DOF) are constructed with a combination of light-weight cylinders and rotary vane actuators. Table 2 lists the actuator type and properties of the arm’s 6 joints, according to the definition shown in Figure 7 on the right.

![Figure 7. CAD rendering and kinematics of the new arms. Left: CAD rendering of the new pair of hydraulic 6-DOF arms. Right: kinematics of the arm with the names of the joints: shoulder abduction/adduction (SAA), shoulder flexion/extension (SFE), humerus rotation (HR), elbow flexion/extension (EFE), wrist rotation (WR) and wrist flexion/extension (WFE).](image)

<table>
<thead>
<tr>
<th>Joint Name</th>
<th>Actuator type</th>
<th>Joint range</th>
<th>max. Joint torque/force (at 20MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SAA</td>
<td>single vane rotary</td>
<td>210°</td>
<td>126Nm</td>
</tr>
<tr>
<td>SFE</td>
<td>double vane rotary</td>
<td>90°</td>
<td>120Nm</td>
</tr>
<tr>
<td>HR</td>
<td>double vane rotary</td>
<td>98°</td>
<td>120Nm</td>
</tr>
<tr>
<td>EFE</td>
<td>cylinder</td>
<td>130°</td>
<td>4kN</td>
</tr>
<tr>
<td>WR</td>
<td>single vane rotary</td>
<td>210°</td>
<td>60Nm</td>
</tr>
<tr>
<td>WFE</td>
<td>cylinder</td>
<td>120°</td>
<td>4kN</td>
</tr>
</tbody>
</table>

Each joint’s position is measured with high resolution absolute encoders (19Bit). While the rotary actuators’ torque output is measured with strain-gauge torque sensors, the cylinder force is obtained with load cells in series to the piston rod. All actuators are controlled by MOOG E024 servo valves. Distributed electronics on the arm read the sensors and create the output signal for the valve amplifiers. An EtherCAT bus connects the arm to the robot. For more detailed information on the arm design and components, see [26].

4.2. HyQ2Centaur Design Concept

HyQ2Centaur is a combination of HyQ2Max (see Section 3.1) and a pair of the new hydraulic arms (Section 4.1) as illustrated in Figure 8. The modular design of the arms allows easy mounting and removing from the robot’s torso. The hydraulic interface consists of two quick release couplings for the arm’s pressure and return lines. The communication interface is a single EtherCAT cable that also provides the electric power to the arm’s electronic boards and valve amplifiers.
The additional weight of the two arms is around 26kg. This payload is not located in an optimal position with respect to the locomotion stability. The most conservative stability criterion is static stability, where the projection of the robot's center of mass onto the ground needs to stay inside the support polygon (created by the feet in contact with the ground). Other stability criteria consider simplified dynamics of the robot to create stable locomotion (e.g. Zero Moment Point).

Since all joints of the robot can be controlled in torque [27], the robot's whole body dynamics model can be used to obtain joint torque profiles that optimise the force distribution of the four feet (and other contact points, e.g. with the arms). We have recently presented our first results with optimized joint torques that allowed the robot to climb inside a V-shaped groove [28]. The same approach allows the centaur robot in the future to optimise joint torques during manipulation tasks.

4.3. Possible Application Scenarios

Manipulation capability allows a legged robot to perform various tasks in real-world applications. Figure 9 illustrates a few of these tasks performed by HyQ2Centaur. During an inspection or rescue operation it might be necessary to open doors or to navigate over challenging terrain to open/close a valve. Other important tasks will be the remote handling of hazardous objects for example for nuclear decommissioning.

5. HYDRAULIC ACTUATION SYSTEM

This section discusses the hydraulic system of HyQ2Max and HyQ2Centaur. In the current configuration hydraulic power is supplied to the robot through two highly flexible hoses. An onboard power pack is currently under development.

Figure 10 shows the schematic of HyQ2Centaur’s hydraulic actuation system. For simplicity only the details of one leg are shown. The torso of the robot carries the following hydraulic system components: An accumulator to smooth out pressure ripples and provide extra flow during fast variations in hydraulic flow demand; a pressure relief valve to protect the system; a normally-open, solenoid-operated vent valve
connecting the pressure supply to tank in case of an emergency and two pressure transducers. The robot’s leg and arm joints are moved by cylinders and rotary vane actuators. Each joint is controlled by a servovalve.

Figure 10. Schematics of the hydraulic actuation circuit of the HyQ2Centaur robot. For simplicity only the details of one leg are shown. The two arms and other legs are built up with the same actuators and valves.

6. DISCUSSION AND CONCLUSION

Open challenges in the field of hydraulic legged robots are primarily the energy efficiency of the hydraulic actuation system, the hose routing and the large size of commercially available components. Energy efficiency is especially poor in torque controlled hydraulic robots because of the internal leakage of the high-bandwidth servovalves needed for proper torque control [27]. Digital hydraulics (see [29] for a recent review) and variable pressure systems are some of the possible solutions that are currently being investigated by the research community. A neat routing of hoses across moving joints is tricky especially if the joint range of motion is large. Slip rings, custom connectors and highly flexible hoses are possible solutions. Another challenge for a hydraulic robot designer is the generally large size of commercially available components. Since the market for small scale components is (still) small, custom-made parts or expensive niche products are often the only solution.

This paper presented the design concepts of the hydraulic quadruped robot HyQ2Max and the centaur-style robot HyQ2Centaur. HyQ2Max is an evolution of IIT’s hydraulic quadruped HyQ, a robot that since 2011 demonstrated various types of agile locomotion and carefully planned navigation over rough terrain. The second version has improved robustness, larger joint ranges and higher joint torques. We presented an overview of the robot’s design and demonstrated the robot’s self-righting ability with a rigid body dynamics simulation. Next, we showed the design of a pair of new light-weight hydraulic manipulator arms that can be mounted onto the HyQ2Max platform to turn the robot into the centaur-style machine HyQ2Centaur. Furthermore, we presented the robots’ hydraulic actuation systems and discussed open research problems in the field of hydraulic legged robots. To the authors’ best knowledge this is the first time the design of a fully hydraulically actuated centaur robot is presented.

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DEVELOPMENT OF A LIGHTWEIGHT ON-BOARD HYDRAULIC SYSTEM FOR A QUADRUPED ROBOT

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Abstract
This paper presents the development of an on-board power pack for a highly dynamic and lightweight hydraulic quadruped robot called MiniHyQ. It is a torque-controlled quadruped robot able to walk over rough terrain, jump and run. The compact power pack is designed to fit inside MiniHyQ’s torso section. The hydraulic power source is provided by an on-board miniature gear pump, which is driven by a high torque brushless motor. The selection of each appropriate component of power pack is discussed in detail. A step-by-step procedure is proposed which demonstrates how to design of the power pack for a quadruped robot in order to obtain the desired performance. A centralized compact and lightweight manifold design is also presented which works at 20 MPa operating pressure.

KEYWORDS: Hydraulic Quadruped robots, Miniature Hydraulics, Hydraulic Power Pack

1. INTRODUCTION

The hydraulic actuators are very favorable for the highly dynamic robots because of their higher torque and smaller size than the electric motors. Most of the current research platforms in legged robotics are hydraulically actuated. The hydraulic actuation provides them strength and fast motions to perform dynamic tasks. The majority of the successful platforms are hydraulically powered as opposed to electrically actuated, for example, WildCat and BigDog [1] from Boston Dynamics and HyQ [2] from IIT. Hydraulics has the significant advantage of being able to absorb high impact loads (no gears required), having a high power to weight ratio and allowing the implementation of high performance torque control [3]. For producing the desired hydraulic actuator’s joint torque, it is essential to have a hydraulic power pack which supplies needed pressure and flow rate. But having an on-board hydraulic power pack is a burden to a legged robot. It is not only increase a weight of a robot but also raise the joint torque requirements. To reduce the weight of the hydraulic power pack, we built a highly dynamic and lightweight hydraulic quadruped robot MiniHyQ. To the authors’ best knowledge, MiniHyQ is the lightest and smallest hydraulic quadruped robot that has been built so far. The weight of the hydraulic power pack is in proportion to the supply pressure and the supply flow. So the velocity of the hydraulic actuator should be reduced for reduction of a supply flow. MiniHyQ uses a special mechanism [6] for its knee joint which has tendency to produce high velocity when its leg is fully extended. This knee joint helps MiniHyQ to reduce its actuator velocity during the leg in swing phase and it also minimizes the flow rate consumption.

T. Kim et al. proposed and experimentally validated the algorithms [15] which uses the kinematic redundancy of each leg mechanism for a hydraulic quadruped robot to minimize the flow rate consumption. He
demonstrated that the reduction in actuator velocity by using leg kinematic redundancy reduces the weight of the hydraulic power pack up 34% [15].

In the case of other hydraulically actuated quadruped robots like BigDog [1] monopedal robot [10], and JINPOONG [12], they have a combustion engine to actuate the pump inside the torso. However, with combustion engine it is difficult to conduct experiments indoors, because of the noise and the exhaust fumes. Normally for indoor experiments, the external electric pump is used. It supplies hydraulic power to the robot by means of two hydraulic hoses. These hoses can negatively affect the dynamics of the robots causing unpredictable disturbances and restricting the working range of robot in a circumference around the pump. Therefore, we decided to develop on board hydraulic system with electric pump for MiniHyQ. By using the electric motor, the robot needs only electric wire which is more flexible and less affect motion of the robot. Although the robot can be power autonomous by putting a battery on the robot, we focused on wired system in consideration of weight of the battery. This kind of on board hydraulic system is already adopted to Atlas humanoid robot and Baby elephant [14].

The motivation for this work arose from the experience of our group (the Dynamic legged systems (DLS) lab) with the quadruped robot HyQ [2] shown in Figure 1 (left), which leads us to build MiniHyQ which can be seen in Figure 1 (right).

![Figure 1. The DLS lab robots (left) the HyQ Robot; (right) the MiniHyQ Robot](image)

**Contributions:**

The main contribution of this work is the development of an on-board power pack for a lightweight hydraulic quadruped robot-MiniHyQ. To the authors’ best knowledge, MiniHyQ is the lightest and smallest hydraulic quadruped robot that has been built so far. We demonstrated the design of the power pack which fits inside MiniHyQ’s torso section and a step-by-step procedure is proposed which demonstrates how to design the power pack for a quadruped robot in order to obtain the desired performance. In order to keep MiniHyQ legs as lightweight as possible, a centralized manifold is placed in torso rather than using distributed manifolds on each leg.

**Paper Outline:** The paper is structured as follows: first, an overview MiniHyQ design is discussed in Section 2 and it includes the description of the MiniHyQ’s control system; next, Selection of each component of the MiniHyQ’s hydraulic system is demonstrated in Section 3. Section 4 discusses the power pack and concludes the paper.
2. MINIHYQ ROBOT

This section presents an overview of MiniHyQ’s design [16] and its control system architecture. Table 1 shows the specification of MiniHyQ. Each leg is driven by 3 hydraulic actuators. We selected these actuators based on our scaling studies [4][5]. In these studies, we considered extreme tasks that push the actuators to their limits. This gives us a good estimation of the joint torques and velocities necessary to select the leg actuators. MiniHyQ has a 0.85m long torso as shown in Figure 4 (right). It is made of 2 mm thick folded aluminium sheet and contains the computing system, IMU (inertia measurement unit) sensor, hydraulic manifolds and compact power pack. Hip Flexion/Extension (HFE) and Knee Flexion/Extension (KFE) are the joints, which work in leg-sagittal plane. They are responsible for generating the main forward and upward motion of the robot. Most tasks like straight walking and running on flat terrain are accomplished by these joint. Rotary hydraulic actuators have wide range of motion and constant torque. However, they are heavier than linear actuator. For MiniHyQ’s HFE joint, we used a rotary actuator. But if we put rotary actuator inside the KFE joint, it would increase the inertia of leg significantly. Therefore, for KFE we used linear actuator with special knee mechanism [6], which does not only provide wider range of motion but also provides an optimized torque profile. The third joint named as hip Abduction/Adduction (HAA) is less involved in the creation of forward propulsion. Linear actuators are also used for the HAA Joint. CAD Model of MiniHyQ’s leg design is shown in Figure 2.

![Figure 2. CAD model of MiniHyQ Leg which consists of 3 active DoF i.e HAA, HFE and KFE joints.](image)

**Hip Abduction/adduction (HAA)** is important joint of a quadruped robot to support robot’s weight (mostly in case when robot legs are not parallel to its leg-sagittal plane). HAA joint always needs to react quickly to keep robot’s balance. It requires a reasonable joint torque and velocity. An asymmetric hydraulic cylinder is used, which has a bore diameter of 13mm and a rod diameter of 6mm with 69 mm stroke length. It weighs 0.11kg and one end of cylinder is attached on the top surface of Hip Flexion/Extension (HFE) joint motor and other end is attached to torso plate as shown in Figure 2.

**Hip Flexion/Extension (HFE)** joint is based on hydraulic rotary actuator. It has joint range of motion of 220°(-110° to 110°) and it provides constant joint torque 60 Nm at 20MPa.
Knee Flexion/Extension (KFE) is based on an isogram mechanism, we proposed in [6]. It has a changeable instantaneous center of rotation like a human knee joint. We optimized a set of design parameters to obtain a smoothly distributed torque profile that provides high torque in a retracted joint configuration (i.e. flexed leg) and high velocity (but lower torque) when approaching the fully extended configuration. Furthermore, a large knee joint range \( q \) from 0 to 180°.

Table 1. Specifications of MiniHyQ Robot

<table>
<thead>
<tr>
<th>Specifications</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dimensions (LxWxH)</td>
<td>0.85m x 0.35m x 0.7m</td>
</tr>
<tr>
<td>Weight (off-board/on-board Power Pack)</td>
<td>24kg, 35kg</td>
</tr>
<tr>
<td>Degrees of Freedom</td>
<td>12 (3 per leg (2-linear1-rotary hydraulic actuator))</td>
</tr>
<tr>
<td>Joint Torque/Range of motion</td>
<td>75 Nm, 90° HAA</td>
</tr>
<tr>
<td></td>
<td>60 Nm, 220° HFE</td>
</tr>
<tr>
<td></td>
<td>75 Nm, 180° KFE</td>
</tr>
<tr>
<td>Sensors per Leg</td>
<td>2 Load cells, 1 Torque sensor</td>
</tr>
<tr>
<td></td>
<td>3 Absolute encoders</td>
</tr>
<tr>
<td>Hydraulic Valves</td>
<td>12 High performance servo valves (Moog E024)</td>
</tr>
<tr>
<td>On-board Computing</td>
<td>1 computer (real time Linux)</td>
</tr>
<tr>
<td>Operating Pressure</td>
<td>20 MPa</td>
</tr>
<tr>
<td>Peak Flow Rate</td>
<td>13 l/min</td>
</tr>
</tbody>
</table>

The control system architecture of MiniHyQ is shown in Figure 3. It basically consists of a main unit and 4 leg units. In the main unit, control PC running Linux kernel patched with real-time Xenomai takes care of all low level control of servo valves via main I/O board and high level control such as leg trajectory. Leg unit collects input signal from 3 magnet encoders (19 bit, absolute type), 2 force sensors (±4448N) and 1 custom designed torque sensor, and sends these data to the main unit. For communication between each unit, EtherCAT bus.
is used and gives the system high speed real time communication. As a low level controller, each joint is fully torque controlled based on the HyQ’s torque controller [7]. Full torque control allows the robot to perform active compliance which is essential to cope with impacts during dynamic motions. Furthermore inverse dynamics can be used for improving control of locomotion [8].

Figure 4. The CAD model of MiniHyQ with exposed view of onboard power pack, magnetic encoder disks and EtherCat control PCB in upper leg link.

3. DESIGN OF THE HYDRAULIC SYSTEM FOR MINIHYQ

In this section, we explain the step by step design procedure of the hydraulic system of MiniHyQ. Table 2. shows the specification of the designed power pack and Figure 5. shows the schematic of hydraulic system of MiniHyQ. It consists of a miniature pump, an electric motor, manifolds, vent valve and relief valve for safety. This power pack is detachable and it can be replace by an external pump if it is available. Before designing hydraulic system, the maximum pressure of hydraulic system was decided by the actuators desired operating pressure, which is 20 MPa. In following design we use this value as maximum pressure.

Table 2. Specification of power pack MiniHyQ

<table>
<thead>
<tr>
<th>MiniHyQ Power Pack</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Size(L x W x H)</td>
<td>0.59 x 0.20 x 0.19 m</td>
</tr>
<tr>
<td>Weight</td>
<td>12 kg</td>
</tr>
<tr>
<td>Rated flow rate</td>
<td>10 L/min</td>
</tr>
<tr>
<td>Max System Pressure</td>
<td>20 MPa</td>
</tr>
<tr>
<td>Max Power Consumption</td>
<td>5.5 kW</td>
</tr>
</tbody>
</table>
3.1. Estimation of required flow rate

First of all we estimated the required flow rate for each actuator. In order to estimate the required flow rate, we used the experimental data of a previous quadruped robot HyQ[2]. Since MiniHyQ and HyQ have almost the same length of leg segments, we assumed that the required angular velocity of each joint is the same for MiniHyQ and HyQ. As a template motion which determines the maximum performance of the robot, 2m/s running trot and 0.2 m squat jump were chosen. As we explained in section 2, MiniHyQ has different leg mechanisms compared to HyQ. Required flow rate for each joint is calculated by multiplying required angular velocity and volumetric displacement of each actuator. In case of a rotary actuator for HFE, volumetric displacement is constant. However, in case of a linear actuator with linkage mechanism for HAA and KFE, volumetric displacement varies with each position and the volumetric displacement is calculated by inverse kinematics. Figure 6 shows the sum of the required flow rate of all of the actuators and the leakage flow of servo valves.

Figure 5. Schematic of Hydraulic system

Figure 6. Estimated flow rate, blue line shows the total required flow rate of the robot while Running trot, red dotted line shows the total required flow rate while Squat Jump, and black line shows the average flow rate while Running trot.
3.2. Selection of Servo Valve

In order to control the dynamically walking of a robot such as the MiniHyQ, high-bandwidth performance is the most important for torque control. In this case, we choose MOOG E024 nozzle flapper type servo valve whose dynamic performance is 250Hz. This servo valve is also used for the HyQ robot. In order to verify selected servo valve can control the flow and pressure while template motions, Figure 7 shows required flow rate versus load pressure of HFE and KFE joint actuator (HAA is excluded because it is not dominant for running trot and squat jump) which were calculated above and the flow-load characteristic of the Moog E024 servo valve at fully open. As shown in the graph, at all of the operating point, flow rate and load pressure are under the performance of the servo valve at fully open.

![Figure 7. Verification of servo valve performance](image)

3.3. Selection of a pump and an electric motor

The most important points of a power pack for a mobile robot is lightweight. In order to achieve lightweight, a high power weight ratio electric motor and lightweight pump is used which fulfill the required specification estimated above (average flow rate 10L/min, system pressure 20Mpa).

Main problem of selecting pair of an electric motor and a pump is mismatching of required speed and torque, even though motor output power and pump output power is same. Nowadays it is not difficult to find a high power weight ratio electric motor. However such high power is only obtained at high speed region and low torque region. On the other hand required output power of a pump is decided by its differential pressure, volumetric displacement and maximum input speed. Volumetric displacement can be choose, but maximum input speed is limited by a sealing. To solve this problem, two ways are considered. First one is introducing a gear reduction mechanism to change ratio of torque and velocity. Downside of this way is decreasing power weight ratio because of increasing of total mass and there is energy loss which indicated as efficiency. Thus big reduction ratio should be avoided. Second solution is selecting as much as small volumetric displacement pump to fulfill the required flow rate. As shown in below equation, required input torque is decreased proportionally to volumetric displacement. $V_g$ is volumetric displacement per rotation [cm$^3$], $\Delta P$ = differential pressure [MPa], $\eta$ = efficiency of a pump.

$$T_{input} = \frac{V_g \cdot \Delta P}{2\pi \cdot \eta} \quad (1)$$

Unlike introducing the gear reduction mechanism, volumetric displacement is basically does not change allowable input speed. Thus to keep its output power required input speed have to be increased inversely proportional to volumetric displacement.
Considering above 2 points we choose an electric motor and a pump from available consumer products. According to Figure 6, required average flow rate what a pump needs to supply is 10 L/min, and the maximum hydraulic pressure is 20 MPa. To achieve this specification with minimum volumetric displacement, we choose small size axial piston pump with constant volumetric displacement made by TAKAKO (maximum operating pressure 21MPa, maximum input speed 3000rpm, volumetric displacement 3.15 [cm³], weight 1.94 kg). Actually its rated flow rate from the specification is 9.15 L/min, but we consider required average flow rate 10 L/min is not always required and in short time operation the pump could rotate at 3300 rpm to achieve 10 L/min. Otherwise we have to choose another pump which has double volumetric displacement and it requires too much torque to the motor. As a high power weight ratio electric motor, brushless DC motor originally designed for a hobby air plane by Neu motor (maximum output power 10kW, weight 1.36kg) and 1 stage planetary gear box (reduction ratio 6.5, weight 0.43kg) was choose. Additionally the motor contains cooling fan internally to keep its working temperature. The chosen pump and motor are shown in Figure.8.

3.4. Accumulator

To absorb sudden change of flow rate and deviation of flow rate because of pump, accumulator is required. In this case, diaphragm accumulator was chosen to compensate 3 L/min which is difference between maximum flow rate 13 L/min and average flow rate 10 L/min. To select appropriate accumulator we assume adiabatic change, actuation time of accumulator is 0.2s and minimum operation pressure is 18MPa. If we want to sustain 3L/min, the accumulator needs to provide 0.1 L. Pre-charged pressure 14.4MPa is calculated by using is recommended compression ratio 0.8. From these values required pre-charged nitrogen gas volume is by calculated below equation. $V_i$ and $P_i$ means the volume and the pressure of nitrogen gas of each state. i=0 is pre-charged state, i=1 is minimum hydraulic pressure state and i=2 is maximum hydraulic pressure state. Efficiency of accumulator $\eta$ and is assumed 0.95.

$$V_0 = \frac{V_1 - V_2}{\left(\frac{P_0}{P_1}\right)^{\frac{1}{\eta}} - \left(\frac{P_0}{P_2}\right)^{\frac{1}{\eta}}} \times \frac{1}{\eta} = 0.17 L \quad (2)$$

We search accumulator which has gas volume more than 0.17 L and found HYDAC 0.32 L diaphragm accumulator as smallest and lightest one.
3.5. Filter and Oil cooler

The servo valves require NAS 3 or lower, which means contamination must be smaller than 3μm. We choose a line filter which is a lightweight filter among the one fulfill this requirement and its maximum operating pressure is 20 MPa. Estimated pressure drop is 0.15MPa at flow rate 13L/min, thus we considered this pressure drop is negligible. We choose the oil cooler which has aluminum honeycomb shape and its cooling capacity is estimated 2300W with 9.0m/s air flow. Since we use constant volumetric pump, if robot does not consume kinetic energy e.g. standing, most of energy will be turned into heat and it is difficult to cooling such amount of heat by the oil cooler and also a lot of heat dissipation decrease energy efficiency of the robot. In order to avoid these problems we will control the rotation speed of the pump depends on movement of the robot.

3.6. Reservoir

In case of walking robot whole body is always vibrating and the air can enter into the oil because of walking motion, although conventional reservoir is usually open to the atmosphere. In order to solve this problem some aircraft uses self-pressurizing reservoirs (bootstrap reservoirs). However commercial self-pressurizing reservoir is too bulky for mobile robot, thus we uses an accumulator as a reservoir.

Maximum system oil difference was estimated 0.19 L (linear actuator x8 = 0.06L, accumulator = 0.1 L, temperature variation = 0.03L). By following same procedure with section 3.4, if we assume maximum pressure = 0.5MPa, minimum pressure =0.2MPa and pre-charged pressure = 0.17MPa, required gas volume is calculated as 0.47L. Since this accumulator is only used under low pressure.

3.7. Manifold

MiniHyQ has 12 active joints and its each actuator is control by the high performance servo valve (Moog E024). Two separate centralized manifolds are used for front and hind legs. Both manifolds are identical and each has capacity of six valves. A centralized manifold shown in Figure 9 that is placed in torso rather than using distributed manifolds on each leg and this design is validated by FEM analyses.

![Figure 9: MiniHyQ's centralized manifold](image)

4. DISCUSSION & CONCLUSION

MiniHyQ is a pioneer, slightly smaller in size than HyQ [2] but the lightest among the existing hydraulically actuated quadruped robots [1][12][14]. The development of an on-board power pack for the MiniHyQ robot is a significant step forward in miniature hydraulics. We demonstrated the development of an on-board hydraulic system and thoroughly described how we designed on-board power pack for it.
**Future work:** Future experiments will be performed on MiniHyQ using an on-board power pack. The optimization of MiniHyQ's servo valves is also the part of future work.

**ACKNOWLEDGMENTS**

The authors would like to thank also the colleagues that collaborated for the success of this project: Michele Focchi, Jake Goldsmith, Bilal Ur Rehman, Ioannis Havoutis and our team of technicians. This research has been funded by the Fondazione IstitutoItaliano di Tecnologia.

**REFERENCES**


VEHICLE MASS ESTIMATION FOR HYDRAULIC DRIVE SYSTEM USING LONGITUDINAL MOTION MODEL

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ABSTRACT

The real-time mass estimation of the vehicle is applied for the machine with the hydraulic drive system. The mass estimation is based on the longitudinal drive model comprising the model of hydraulic drive transmission. The resistance forces of the longitudinal motion such as the air drag, rolling resistance and friction of the drive system are modelled. The actual mass of the vehicle is deduced from the measured hydraulic torque and from the force causing the acceleration. As the aerodynamic drag, rolling resistance, road grade load and transmission losses have a significant share from the total drive torque, the effects of these forces are taken account. Further, the estimated mass data is classified by recognising operation conditions where the mass estimation is accurate. After a short acceleration-deceleration period, the measured signals provide enough data for estimating the mass of the vehicle. The experimental tests are run with a middle-size wheel loader and with a typical work cycle resulting ±5% accuracy from the real mass. Furthermore, the proposed model and recognition of the operation conditions are applicable to estimate also other vehicle parameters such as friction force or road grade.

KEYWORDS: Longitudinal motion, mass estimation, hydraulic drive system, wheel loader

1. INTRODUCTION

The actual mass of a vehicle is an important parameter when tuning a drive system or operator assisted systems. For example, continuously variable transmission (CVT) and speed control is adapted in many tractors which are no more the throttle controlled [1], [2]. Taking to account the mass variations of a vehicle can improve the accuracy and usability of these control methods. Even small changes in vehicle mass have a large effect on the performance of a fixed gain controller and this can be compensated with parameter estimation [3]. Furthermore, operator assistance systems and data acquisition systems can require the information about actual mass. An overloaded truck may cause several adverse consequences such as truck instability, reduced braking capacity, reduced maneuverability or overheating of the tires [4]. To reduce accidents, a warning system of the overloading can be applied. In the agriculture, vehicle weight, position, travelsability and other machinery information connected with the field information will be one way to improve the food production [5].

Various approaches to an online estimation of vehicle mass, road friction and road grade are developed in automotive industry [6], [7], [8]. A widely adopted method for estimating vehicle mass is to capitalize on the relationship between the longitudinal force and the longitudinal acceleration. The existence of unknown road angles and other alternating resistance forces complicate the longitudinal motion based estimations. To
exclude the effect of unknown forces, the filtering techniques are applied [6]. Alternative solution is real-time estimation of unknown parameters such as a road grade [9], [10].

Further, the vehicle mass can be estimated from the vertical motion of the sprung mass by applying suspension models [11], [12]. There, the models apply the sprung and unsprung mass accelerations and filtering methods for estimating the mass of the vehicle. In addition, the lateral and yaw motion or composing the lateral, vertical and longitudinal dynamics can be applied for the defining vehicle mass [6], [8].

This research applies the longitudinal motion model of the hydraulic wheel loader which includes the model of the hydraulic drive system (Section 2). The longitudinal motion model has been developed earlier for the prediction of drive torque but the same model can be applied here [13]. To estimate the mass of the wheel loader, the operation conditions are recognised with predefined criterion. Arithmetic mean from the estimated mass is calculated in the conditions where the mass estimation is valid (Section 3). The experimental tests are run with a typical work cycle of the wheel loader (Section 4). After a short acceleration-deceleration period, the measured signals and vehicle model provide a data for the estimation algorithm. Finally, the paper presents a discussion and conclusion of these results (Section 5).

2. VEHICLE

The vehicle under study is a wheel loader, mass around 5500 kg and named as M12-machine. The drive unit is a 96-kW diesel engine connected to the drive pump and producing the traction force. As a whole, the hydraulic transmission is implemented with a coupled hydraulic pump and motor connected to a two-speed gearbox.

The wheel loader has been instrumented with various sensors producing measurements about pressures, temperatures, positions, speeds and accelerations. The diesel engine and all hydraulic actuators are electronically controlled which provides the opportunity to study various research objectives. To ensure effective and reliable control and measurements, the internal communication of the controllers employs CAN-technology (Controller Area Network). All sensor information and most of the control signals can be collected by reading the CAN-messages.

2.1. Longitudinal motion model

The longitudinal motion model of the vehicle is based on the modelled resistance forces. To simplify the model, only the longitudinal forces are taken account. That assumption sets lateral forces to zero and longitudinal forces are evenly distributed on the right and left wheels, so only longitudinal dynamics is affecting.

In general, the acceleration force of the vehicle is the effective traction force taken off the resistance forces. Here, the machine is treated as a mass point with a rigid body and the acceleration force can be derived from the forces effecting on the vehicle.

\[ F_{\text{acc}} = m \ddot{v} = F_{\text{Traction}} - F_{\text{Roll}} - F_{\text{Air}} - F_{\text{Slope}} \quad (1) \]

where \( F_{\text{acc}} \) is the acceleration force, \( m \) is the mass and \( \ddot{v} \) is the acceleration of the vehicle, \( F_{\text{Traction}} \) is the effective traction force of the vehicle, \( F_{\text{Roll}} \) is the rolling resistance, \( F_{\text{Air}} \) is the air drag and \( F_{\text{Slope}} \) is the force cause by the slope of the road. The rolling resistance, air drag and slope of the road are given by:

\[ F_{\text{Roll}} = f_{\text{roll}} mg \cos(\theta) \quad (2a) \]
\[ F_{\text{Air}} = 0.5 \rho_{\text{air}} C_{\text{air}} A \dot{v}^2 \quad (2b) \]
\[ F_{\text{Slope}} = mg \sin(\theta) \quad (2c) \]
where, \( f_{\text{roll}} \) is the rolling resistance coefficient of the vehicle which is assumed constant, \( m \) is the mass of the vehicle, \( g \) is the gravity, \( \rho_{\text{air}} \) is the density of the air, \( c_{\text{air}} \) is the aerodynamic drag coefficient, \( A \) is the frontal area of the vehicle, \( v \) is the velocity and \( \theta \) is the angle of the inclination to the horizontal (a slope angle).

The longitudinal forces affecting the body are described in Figure 1 where the positive directions of forces are observed against the direction of the movement. Because the vehicle and potential payload are considered as one mass point and forces are supposed to distribute evenly between tires, the tractive force and each resistance force can be observed as unity.

![Figure 1. Longitudinal forces of the machine.](image)

To define the mass of the vehicle, Equations (1) and (2) can be applied. The mass is assumed to be a mass point which contains the actual mass of the vehicle and payload. The rotational mass of the vehicle is not taken account here.

\[
m = \frac{F_{\text{Traction}} - F_{\text{Air}}}{\dot{v} + f_{\text{roll}} g \cos(\theta) + g \sin(\theta)} \tag{3}
\]

Furthermore, the tractive force is assumed to the same as the force produced by drive transmission. That simplification ignores the tire slip and excludes the power loss due the sliding.

2.2. Vehicle model

Commonly, the tractive force of the vehicle cannot be directly measured. Instead, the engine, transmission or hydraulic torque can be measured or estimated by utilizing instrumented sensors. In the wheel loader under this study, the hydraulic torque can be defined. Therefore, a vehicle model with a drive transmission is required to determine the effective tractive force and to estimate the actual mass of the vehicle.

Figure 2 presents the hydraulic transmission of the wheel loader. That has a hydrostatic circuit which is a coupled system with a variable displacement pump and variable displacement motor. The pump is directly connected to the diesel engine, so the rotational speeds of the engine and pump are the same. The hydraulic motor is connected to a two-speed gearbox which selects the active gear of the wheel loader. The rotational speed of the two-speed gearbox is then converted to the tires speeds in gear boxes of the front and rear body.
As Figure 2 presents, the transmission comprises several rotating speeds and inertias which generate internal torque while accelerating or decelerating the vehicle. Previously, the effective rotational inertia is ignored when calculating the mass of the vehicle. When the effective rotational mass is known, the effective mass can be separated to the actual mass and rotational mass:

\[ F_{\text{acc}} = m_{\text{eff}} \ddot{v} = (m + J_{\text{Eff}} \frac{\omega_{\text{Tire}}}{r_{\text{Tire}}}) \ddot{v} \]  

(4)

where \( m_{\text{eff}} \) is the effective mass of the vehicle, \( J_{\text{Eff}} \) is the effective rotational inertia proportional to rotational speed of tires, \( r_{\text{Tire}} \) is the effective radius of tire, and \( \ddot{v} \) is the acceleration of the vehicle. The acceleration force is positive during acceleration and negative while deceleration.

Further, the effective rotational inertia can be determined from the partially defined inertias applying the speed ratios from Figure 2. As the effective rotational inertia is proportional to the rotational speed of the tires, the partially defined inertias are converted to the proportional of the tire speeds.

\[ J_{\text{eff}} = 4J_{\text{Tire}} + J_{A1} + J_{A2} \alpha_1^2 + J_{M}(\alpha_1 \alpha_{\text{gear}})^2 \]  

(5)

where, \( J_{\text{Tire}} \), \( J_{A1} \), \( J_{A2} \) and \( J_{M} \) are the inertia of tire, shafts and hydraulic motor, \( \alpha_1 \) is the speed ratio of the front and rear gear box and \( \alpha_{\text{gear}} \) is the speed ratio of the gear box connected to the hydraulic motor.

As the traction force cannot be directly measured, the torque of the hydraulic motor is applied instead. The frictional loss of the transmission between the tires and the hydraulic motor reduces the produced traction force. The relation between the torque of the hydraulic motor and the traction force can be described as:

\[ F_{\text{Traction}} = \frac{T_m}{\alpha_{\text{final}}r_{\text{Tire}}} - F_{\text{fric}} \]  

(6)

where \( \alpha_{\text{final}} \) is the final gear ratio between the tires and the hydraulic motor (here \( \alpha_{\text{final}} = \alpha_1 \alpha_{\text{gear}} \)), \( F_{\text{fric}} \) is the friction force caused by the gearing and rotating shafts, and \( T_m \) is the torque of the hydraulic motor.

The delivery of the hydraulic pump produces the pressure difference of the hydraulic motor which generates the motor torque. The hydraulic torque of the motor is given by:
\[ T_m = \frac{\varepsilon_m V_m \Delta p \eta_{HM,m}}{2\pi} \]  
\( (7) \)

where \( \varepsilon_m, V_m \) and \( \eta_{HM,m} \) are the displacement ratio, maximum displacement and hydro-mechanical efficiency of the hydraulic motor, and \( \Delta p \) is the differential pressure of the hydraulic motor.

The actual mass of the wheel loader can be determined by applying Equation (3) and the defined model of the drive transmission.

\[ m = \frac{T_m}{\alpha_{final} r_{Tire}^2} - F_{fric} - \frac{J_{Eff}}{r_{Tire}^2} \dot{v} - F_{Air} \frac{\dot{v}}{v} + f_{roll} g \cos(\theta) + g \sin(\theta) \]  
\( (8) \)

When the mass of the wheel loader is estimated, the measured inputs for the motion model are the velocity, acceleration, angle of the slope, displacement ratio of the hydraulic motor and differential pressure of the hydraulic motor.

2.3. Parameters of the wheel loader

The longitudinal motion model requires defining the parameters of the wheel loader; these parameters are defined in the earlier study [13]. Table 1 presents the parameters of the wheel loader which remains constant during the measurements. The wheel loader applied here, is a prototype rebuild as the research platform.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rolling resistance coefficient</td>
<td>( f_{roll} )</td>
<td>0.025</td>
</tr>
<tr>
<td>Air drag coefficient</td>
<td>( C_{air} )</td>
<td>2.0</td>
</tr>
<tr>
<td>Density of air</td>
<td>( \rho_{air} )</td>
<td>1.225 kg/m(^3)</td>
</tr>
<tr>
<td>Gravity</td>
<td>( g )</td>
<td>9.81 m/s(^2)</td>
</tr>
<tr>
<td>Frontal area of vehicle</td>
<td>( A )</td>
<td>4.0 m(^2)</td>
</tr>
<tr>
<td>Effective rotational inertia</td>
<td>( J_{Eff} )</td>
<td>144 kgm(^2)</td>
</tr>
<tr>
<td>Tire radius</td>
<td>( r_{Tire} )</td>
<td>0.55 m</td>
</tr>
<tr>
<td>Maximum displacement of the motor</td>
<td>( V_m )</td>
<td>85e(^{-6}) m(^3)</td>
</tr>
<tr>
<td>Maximum displacement of the motor</td>
<td>( V_p )</td>
<td>85e(^{-6}) m(^3)</td>
</tr>
<tr>
<td>Final gear ratio</td>
<td>( \alpha_{final} )</td>
<td>0.3636</td>
</tr>
</tbody>
</table>

Table 1. Parameters of the vehicle model.

All parameters cannot be defined beforehand as these are depending on the operation conditions. Moreover, the variables which are not measured directly are defined as a function of other measurements. The next section defines how the frictional loss, hydrostatic efficiencies and displacement ratio of the motor are defined as a function of operation conditions.
2.3.1. Frictional loss of drive line

The torque loss of the drive line is estimated as simplified friction model which is parametrized in the earlier research [13]. The friction force is determined as the function of the velocity.

\[ F_{\text{fric}} = F_c \text{sign}(v) + F_v v \]  

where \( F_c \) is the coulomb frictions force, \( F_v \) is the viscous friction force, and \( v \) is the velocity of the vehicle. The sign function defines the direction of friction force and sets friction to zero with zero velocity.

2.3.2. Hydraulic efficiencies

The hydro-mechanical and volumetric efficiency of the hydraulic motor and the pump are measured before the components are installed in the machine. The measurements are done in steady-state conditions by varying the displacement ratio, rotational speed and differential pressure.

Based on the measurement data, the efficiency functions are defined for the pump and motor where the inputs are the rotational speed, pressure difference and displacement ratio. The functions apply a linear interpolation between measured data points and return the interpolated efficiency.

2.3.3. Displacement ratio of the hydraulic motor

The displacement of the hydraulic motor is controlled by the proportional electric control where an adjustment of the displacement ratio is proportional to the control current applied to the solenoid. As the direct measurement of the displacement ratio is not available, the current of the control solenoid can be measured instead. Nevertheless, it has been noticed that in some situations the actual displacement ratio does not follow the control current accurately. Especially, the dynamic of the control varies that generates unknown delay to the estimated displacement ratio.

In consequence, the displacement ratio of the hydraulic motor cannot be estimated precisely from the control current of the solenoid. Instead, determining the delivery of the hydraulic pump and measuring the rotational speed of the hydraulic motor, the displacement ratio of the hydraulic motor can be defined as:

\[ \varepsilon_m = \frac{n_p \varepsilon_p V_p \eta_{V,p} \eta_{V,m}}{V_m n_m} \]  

where \( n_p \) is the rotational speed of the pump, \( \varepsilon_p \) is the displacement ratio of the pump, \( V_p \) is the maximum displacement of the pump, \( n_m \) is the rotational speed of the hydraulic motor, and \( \eta_{V,p}, \eta_{V,m} \) are the volumetric efficiency of the pump and motor. With Equation (10), the displacement ratio of the hydraulic motor is estimated accurately for the mass estimation of the wheel loader.

3. MEASURES AND RECOGNITION OF THE OPERATION CONDITIONS

To provide the accurate estimation of the vehicle mass, the operation conditions where the mass estimation is valid and the estimation error small should be defined. These operation conditions depend on the applied sensors and the properties of the longitudinal motion model. Thus, the accuracy of the valid measurement points is evaluated against the estimation error of the mass.
3.1. Instrumented sensors

Velocity is measured with the rotational speed sensor which is integrated into the hydraulic motor. As the gear ratio and tire radius are known, the velocity of the vehicle can be defined. Here, the assumption is that differential lock is not sliding and the rotational speeds of the tires are the same.

Even the tire slip is ignored from the force calculations, it can effect on the measured acceleration. Furthermore, the effective tire radius can vary with the different velocities and payloads of the vehicle. To overcome these challenges, the acceleration of the vehicle is measured with the inertial measurement unit (IMU) instead differentiating the measured velocity from the hydraulic motor. Applying accelerometers and gyros, the IMU provides measurements for the longitudinal and lateral accelerations. Also the slope angle can be determined.

The hydraulic drive pump has integrated electronics which provides the direct measures of the displacement ratio and differential pressure. As the hoses between the pump and motor are short, the differential pressure of the pump can be assumed the same as the pressure difference of the motor. The rotational speed of the pump is measured from the diesel engine which has the same rotational speed.

3.2. Recognition of operation conditions

Due the model and measurement inaccuracies, the predefined parameters are not accurate in all operation conditions. Additionally, the small errors in the measured signals have a significant effect on the estimated mass especially when the acceleration of the wheel loader is small. Therefore, the operation conditions where the model is accurate and the mass estimation error is small will be defined.

The deceleration phase of the vehicle cannot be applied for the mass estimation because the brake force is not measured. Therefore, the mass is estimated only when the wheel loader is accelerating and the brake force is not effecting. Further, the proposed longitudinal motion model does not include the static friction model. Therefore, model output is not valid while the vehicle is not moving or moves slowly.

As the inertia of the mass is dominant variable when the actual mass is estimated, the estimation algorithm is valid in operation conditions where the acceleration of the inertia dominates the longitudinal forces. Therefore, the mass of the vehicle is estimated only when acceleration is relative high. As the assumption is that tire slip is not affecting, the tire acceleration is compared with the vehicle acceleration from the IMU. This ensures that tires are not slipping significantly. The defined preconditions of the mass estimation method are:

1. The vehicle is accelerating (acceleration > 0)
2. The velocity of the machine is higher than 0.8 m/s.
3. The acceleration is higher than 0.8 m/s².
4. The difference between tire acceleration and vehicle acceleration is not higher than 0.5 m/s².

The limits for the proposed preconditions are defined experimentally and are applied when the mass of the vehicle is estimated.

4. EXPERIMENTAL RESULTS

A typical load-haul-dump cycle without loading operations is defined for the wheel loader. This work cycle provides typical operation conditions for the wheel loader and demonstrates the applicability of the mass estimation algorithm. The experimental work cycle consists of several acceleration-deceleration sections where the mass of the wheel loader can be estimated.

In the tests, the loader moves on asphalt on relatively flat terrain. The starting point and endpoint of the cycle are the same. The work cycle is repeated three times in each test and totally travelled distance is about 350
Figure 3 presents measured signals from the test case 3 where the wheel loader is driving without external load.

During the work cycle, the driving contains forwarding and reversing sequences with the peak velocity around 15 km/h as presented on the left side of the second section on Figure 3. The third section presents the torque of the hydraulic motor and the acceleration of the vehicle. During the work cycle the acceleration of the mass dominates the longitudinal forces. That can be noticed by comparing the shape of the acceleration and torque curve where the torque of the hydraulic motor follows the acceleration of the vehicle.

The first section on Figure 3 presents the estimated mass which is calculated from the measurement signals with Equation (8). The red color means the sections where data is valid and the blue color means the sections where the data is ignored. The estimated mass oscillates significantly because of the disturbances in the longitudinal model. Furthermore, the delay and dynamics of the sensors cause a phase difference for measurements and generate the error in the estimated mass value. Also, the hydrostatic transmission induces the delay between the measured acceleration and torque of the hydraulic motor because of the transmission is not rigid as the mechanical transmission.

The left part of the second section on Figure 3 presents the filtered mass estimation which is calculated in the operation conditions that satisfy the predefined recognition criterion. The single values of the estimated mass (red dots) are still varying between 3 tons and 10 tons. Despite the large scale of the variations, the mean value of the estimated mass (blue line) saturates quickly close to the weighted mass of the wheel loader.

To validate the accuracy of the proposed mass estimation algorithm, two series of empirical tests are presented. In the first series, the wheel loader is driving without external payload and tests are repeated four times. The wheel loader is weighted alone on the scale and the actual mass of the loader and driver is 5500kg. In the second test series, the payload of 1500 kg is installed in the front loader so the total mass of the wheel loader is 7000kg. Table 2 presents the measurement results of the estimated mass.
Table 2. The actual mass of the wheel loader is 5500kg in tests 1-4 and 7000kg in tests 5-8.

<table>
<thead>
<tr>
<th>Test</th>
<th>Mean (kg)</th>
<th>Error (kg)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Series 1 (5500 kg)</td>
<td>1</td>
<td>5477</td>
<td>23</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>5503</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>5563</td>
<td>63</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>5688</td>
<td>188</td>
</tr>
<tr>
<td>Series 2 (7000 kg)</td>
<td>5</td>
<td>7116</td>
<td>116</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>6823</td>
<td>177</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>7179</td>
<td>179</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>6809</td>
<td>191</td>
</tr>
</tbody>
</table>

The errors of each test case are calculated from the mean value. All tests detect the actual mass of the wheel loader higher than ±5 % accuracy. Comparing the estimation results with and without payload, the significant difference in estimation error cannot be detected.

5. CONCLUSION AND DISCUSSION

This paper investigates the mass estimation algorithm for the hydraulic drive system which is based on the longitudinal dynamics of the vehicle. The literature presents many approaches for the mass estimation especially in the car industry. Compared with these, the proposed mass estimation algorithm extends the longitudinal motion model for the hydraulic drive systems. As the drive torque is usually measured from the engine or with an installed torque transducer, this paper provides the method for applying the measurements of hydraulic torque.

The experimental tests consist of two series of the measurements, with and without 1500 kg payload. After a couple of acceleration-deceleration periods, the detected mass saturates quickly close to the final estimated mass. The estimated mass is arithmetic mean and in each test provides the accuracy higher than ±5 % from the actual mass.

REFERENCES


NOVEL HAPTIC CONTROLLER FOR NON-ROAD MOBILE MACHINE TELEOPERATION

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ABSTRACT

There are multiple commercially available haptic devices, but very few are specifically designed for the purpose of teleoperation of non-road mobile machinery (NRMM). This paper presents a novel haptic controller for the teleoperation of NRMM. The manipulability and workspace of an earlier designed haptic controller was analysed with Matlab and Robotic Toolbox. The workspace of the haptic controller prototype is a sphere which is hollow inside. Analysis showed that the haptic controller is most usable when the operator is using the device at the same level to the armrests and that it can be placed on either side of the operator. Results of this analysis suggest the direction of further development of the haptic controller.

KEYWORDS: Haptics, non-road mobile machinery, force feedback, Virtual reality (VR), Real-time simulator, teleoperation

1. INTRODUCTION

Efficient use of any driven machine requires sufficient and correct feedback of information to all senses [1]. When considering teleoperation, special requirements should be taken into account. In context of this study teleoperation is considered to be the same as remote operation of machine outside the actual machine cabin, independent of the distance and data transmission methods. Things like speed of the machine’s motor can also be felt with haptic senses as forces and vibrations [1], [2]. When remotely operating a machine often the sensory information of forces and vibrations is missing. As a result, the efficiency of the operation can be low [3], [4]. In [3] it was proven that the efficiency of a locally human operated machine is higher compared to a remotely operated machine without force feedback. For force feedback, there are vibrators and motion platforms to provide information to the machine operator, but most joysticks are still without any force feedback features. On the other hand haptic devices are equipment specifically designed to produce force feedback to a user.

Mainly the haptic devices available are either for research purposes, medical industry, military or gaming purposes. The following devices (Geomagic, 2013), (Butterfly Haptics, 2013), (NovintFalcon, 2013) and (Force Dimension, 2013) are examples of commercially available research orientated devices. Figure 1 presents three of the most common types of these force feedback devices.
Multiple studies have been made related to the topic of novel haptic controllers and non-road mobile machines (NRMM) [5], [6], [7] [8] and [9]. In [5] a joystick with added force feedback was presented for rotary crane teleoperation. In [6] a control concept for a hydraulic mobile machine was presented using Phantom Omni [10]. This control concept is illustrated in Figure 2 and promises intuitive operation of the machine. Phantom Omni haptic device was used in this case due to geometric similarities with an excavator. This requirement of geometric similarity restricts the application to other machines.

In [8] a magnetically levitated joystick like the one in Figure 1b was used to control a Feller/Buncher shown in Figure 3.
In [11] is presented a control system for a gantry crane. It has a graphical interface showing a virtual model of a gantry crane and Phantom Omni [10] for force feedback and control of the crane. Unfortunately, other choices for the haptic device were not considered. In [12] a virtual reality (VR) user interface station with haptic device for prototyping of an excavator control systems was developed, but a generic commercial device was used instead of developing an alternative device. In [9] Phantom Premium 1.5 HF was used as a controller of a VR version mine loader. A small shovel was used as a symbolic tool. The natural motions of using the shovel were translated into mine loader control actions. Tests showed the unpracticality of the Phantom Premium for controlling of a NRMM. In [13] it was shown that the Phantom Premium has an unsuitable workspace and mechanical configuration for the purpose of NRMM teleoperation.

Due to observations during research and literature state-of-the-art, a decision to find an alternative mechanical solution for a generic haptic controller for teleoperation was made. One of the targets was a controller that can be used in any NRMM application. Following requirements of the proposed haptic controller for NRMM control are:

- Clear directions of control motions similar to joysticks (one joint per control direction).
- Wide variety of placement positions by the driver’s seat.
- No limitations inside workspace for any control direction.
- Use of symbolic tools as a tool handle.

In this study, the manipulability and workspace of an earlier designed haptic controller is investigated in order to find out if it fulfils the workspace related requirements.

In section 2, the mechanical structure of the novel haptic controller for NRMM teleoperation use is presented. Section 3 shows the results of the analysis of the mechanism workspace. Section 4 contains discussion of results. Section 5 provides the conclusions and recommendations for future research.

2. HAPTIC CONTROLLER

Fluent and natural motion of the user’s hand, in a wider workspace than in Phantom Premium 1.5 HF, suitable for use next to a seat and independency on the orientation or the location was realised in a novel design of a haptic controller. Figure 4 shows the prototype of the haptic controller. User is operating a machine with a handle. In this case the motions of the handle are to represent a control of a NRMM in a similar way as in [9] with Phantom Premium 1.5 HF.

![Figure 4. Prototype of haptic controller](image)
Following section presents the kinematic structure of the prototype of the haptic controller. Table 1 shows it's specifications.

Table 1. Haptic controller specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Prototype of the haptic controller</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal position resolution</td>
<td>0.23 mm</td>
</tr>
<tr>
<td>Maximum exertable force (nominal position)</td>
<td>-</td>
</tr>
<tr>
<td>Continuous exertable force (nominal position)</td>
<td>47.7 N</td>
</tr>
<tr>
<td>Force feedback</td>
<td>x, y, z</td>
</tr>
<tr>
<td>Position sensing</td>
<td>x, y, z (roll, pitch, yaw)</td>
</tr>
<tr>
<td>Interface</td>
<td>Ethernet/VRPN</td>
</tr>
</tbody>
</table>

Figure 5 shows the kinematic structure of the haptic controller. As a 6 DOF mechanism, joint 1 provides the backwards and forwards control direction; joint 2 is for the vertical motion and joint 3 is for sideways direction. Joints 4, 5 and 6 provide three rotational control directions. Tool handle is the point where user holds the controller by hand.

Figure 5. Kinematic structure of the haptic controller

The nominal position resolution of the prototype is much lower than that of Phantom's according to [13]. However, this design is made specifically for NRMM teleoperation control purposes, where resolution requirements are not as in medical applications. The force produced by the prototype is larger than that of Phantom according to [13].
3. ANALYSIS

Matlab 2013a, symbolic toolbox and Robotics Toolbox Release 9 [14] were used for analysing the mechanism and workspace of the haptic controller. Mechanism of the haptic controller prototype was implemented into Matlab by using Denavit-Hartenberg parameters. The Denavit–Hartenberg (DH) parameters describe the kinematic chain of a mechanism. Table 2 shows the Denavit-Hartenberg parameters of the controller.

The DH parameters in Table 2 were used with Matlab and Robotics Toolbox to calculate all possible tool handle locations inside the workspace of the haptic controller prototype. This allows to present workspace as 3D point clouds. In addition, the manipulability of the controller at each point was calculated using Yoshikawa's manipulability measure [15]. All calculations were realised by using the ready-made functions of the Robotics Toolbox [14].

Table 2. Denavit-Hartenberg parameters of the prototype

<table>
<thead>
<tr>
<th>i</th>
<th>$\alpha_i$</th>
<th>$a_i$</th>
<th>$d_i$</th>
<th>$\theta_i$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$-\pi/2$</td>
<td>0</td>
<td>0</td>
<td>$q_1$</td>
</tr>
<tr>
<td>2</td>
<td>$\pi$</td>
<td>0.163 m</td>
<td>0</td>
<td>$q_2$</td>
</tr>
<tr>
<td>3</td>
<td>$\pi/2$</td>
<td>0</td>
<td>0</td>
<td>$q_3$</td>
</tr>
<tr>
<td>4</td>
<td>$\pi/2$</td>
<td>0</td>
<td>0.052 m</td>
<td>$q_4$</td>
</tr>
<tr>
<td>5</td>
<td>$\pi/2$</td>
<td>0</td>
<td>0</td>
<td>$q_5$</td>
</tr>
<tr>
<td>6</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>$q_6$</td>
</tr>
</tbody>
</table>

Figure 6 shows a 3D view of the workspace of the haptic controller prototype. The workspace is hollow inside the sphere and this workspace is made possible by the theoretical ability to use 360 degrees of positions on all joints. The dimensions of the sphere are defined by the distances between the different joints.

Figure 6. 3D view of the workspace of the haptic controller prototype
In Figure 7 can be seen the workspace with values of manipulability according to Yoshikawa’s manipulability measure [15] represented in different colours. Colour bar in Figure 7 indicates the manipulability value: red corresponds to highest manipulability and blue is lowest. The manipulability value is higher when the controller is capable of equal motion in all directions and lower when it is close to a singularity. In singularity, the controller loses one or more of its degrees of freedom. In this case, blue colours show less and red colours more possibility of motions. In Figure 7 the top and bottom of the sphere shape has lower manipulability then the regions located in the equator of the workspace, which can be seen more clearly in Figures 8 and 9.

![Figure 7. 3D view of the full workspace and manipulability value of the haptic controller prototype. Colour bar indicates manipulability value: red corresponds to highest manipulability and blue is lowest.](image)

Figure 8 is taken as a vertical slice of the workspace thru the centre of the sphere in Figure 7. In this can be seen that the controller is most usable when the operator is using the device around the equator of the sphere, referring to the brown and reddish coloured areas of the torus in Figure 8. The not usable areas of workspace in blue have also been noticed in earlier practical tests with the prototype.

![Figure 8. Vertical slice of the haptic controller workspace. Colour bar indicates manipulability value: red corresponds to highest manipulability and blue is lowest.](image)
Figure 9 shows a horizontal slice of the workspace taken thru the middle of the sphere shown in Figure 7. This figure indicates that despite the upper and lower portions of the spherical workspace not being usable, the equatorial middle section on the other hand is usable thru the whole lateral 360 degree range of motion of the haptic controller. Only limitation is the thickness of the torus which in the endpoints of motion of joint 3 (Figure 5) shows lower manipulability as expected in the edges of the shape.

Figure 9. Horizontal slice of the workspace of the haptic controller prototype. Colour bar indicates manipulability value: red corresponds to highest manipulability and blue is lowest.

4. DISCUSSION

The workspace of this haptic controller was analysed using Matlab and Robotics Toolbox. This analysis confirmed the functionality of the haptic controller in the areas of workspace needed for the targeted purpose. The good manipulability around the equatorial area of the workspace allows placement of the device in the vicinity of machine operator’s seat parallel to seat rests, yet does not limit its location only to one side or direction from the user due to symmetrical workspace. When compared to Phantom Premium 1.5 HF, the workspace of the haptic controller is larger when the operator is using the device around the equator of the sphere. The not-usable areas of workspace in blue have also been noticed in earlier practical tests with the prototype.

Advantage of the novel haptic controller is that like joysticks, it can be used with any existing NRMM. And it does not have placement and mechanical limitations of the available haptic devices and freely can be adjusted to work with any existing NRMM. This flexibility of novel haptic controller is achieved with possibility to use up to 6 DOF in control and force feedback.

In this study, only workspace and manipulability were investigated. This analysis provides initial information for choosing the dimensions of the haptic controller based on workspace size needed in NRMM. Dimensions of the haptic controller affect how forces, friction, inertia and backlash are felt by the user and dimensions have to be decided before continuing further with the choice of actuators and sensors. Therefore, for future study closer look on forces, friction, inertia, and backlash of electric motors and used gears is required in details. Also needed resolution of position and force sensors should be researched.
5. CONCLUSION

Although multiple commercial haptic devices are available, they are not designed for the purpose of teleoperation of NRMM. A novel haptic controller for the teleoperation of NRMM was introduced. The workspace and manipulability of this haptic controller were analysed using Matlab and Robotics Toolbox. This analysis confirmed the functionality of the haptic controller in the targeted areas of workspace. The higher manipulability around the equatorial area of the workspace allows placement of the haptic controller next to operator’s seat. In this study, it was discussed that the same analysis can be used for determining the most suitable dimensions for future versions of haptic controller for different NRMM applications.

ACKNOWLEDGMENTS

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ABSTRACT

Autonomous mobile work machines need the capability of sensing and mapping the surrounding area. Machines can utilize several sensors such as laser scanners and cameras for this purpose. The challenge in their use is the relatively high price compared to the value of mobile work machines, and the sensitivity of sensors to harsh operating conditions. This paper presents a low-cost 3D LIDAR for the sensing and mapping of autonomous work machine, which is based on 2D laser scanner and electric motor drive that rotates the scanner. The 2D laser scanner provides range and intensity values from the measured plane and the controller of electric motor provides the rotation angle and rotation speed of the scanner. By combining these values together with navigation data of the machine, 3D point cloud of the surrounding area can be created. This paper presents the development of hardware and control system for the rotating of the 2D laser scanner. Their integration to autonomous mobile work machine and example of mapping results are also presented.

KEYWORDS: Autonomous mobile machine, Mobile robotics, 3D LIDAR, CANopen, Measurement data time synchronization

1. INTRODUCTION

Autonomous mobile work machines require accurate maps of their surroundings in order to be able to perform the required tasks efficiently and especially safely. The machines need first of all rough information of their position at the work site, where the accuracy of few meters is often acceptable. This can be provided for the machine with GPS-based navigation. However, they need in addition high resolution - few centimeters – information of their position in relation to the nearby objects and environment. As the machines are constantly moving, this information needs to be constantly updated.

There are some commercial solutions of 3D LIDARs on the market, for example from manufacturers such as SICK [6] and Velodyne [10], but in many cases they are quite expensive especially for outdoor applications. Therefore there is a need for a robust low cost 3D LIDAR, especially for the research use. This paper presents a solution for a low cost 3D LIDAR based on a 2D laser scanner, electric motor drive for rotating the 2D scanner and data fusion with navigation system of the mobile machine. The 2D laser scanner provides range and intensity data from measured plane. Controller of the electric motor drive provides the rotation angle and rotation speed of the laser scanner. By combining these measurement data with navigation data
one can create point clouds that can be used for sensing and mapping the environment of an autonomous mobile machine. In this paper the mechanics and control system structure of the servo are presented more detailed to ease up building similar type of system for interested reader. The same challenges has been addressed in reference [1].

The developed hardware described in this paper is made for the laser mapping of autonomous mobile machine used in the GIM project [2]. The rotating laser scanner scans the fore field of the machine. The laser map is used among other type of maps for path planning and obstacle avoidance purposes.

The following Chapter 2 presents the control system of the autonomous machine for which the laser scanner was developed. In Chapter 3 the 2D scanner and electric motor drives are presented. The developed hardware and software of the laser scanner and their integration to the autonomous machine are presented in Chapter 4. An example of using the scanner for environmental mapping is presented in Chapter 5.

2. CONTROL SYSTEM OF AUTONOMOUS MOBILE MACHINE

The work machine, for which the mapping system was developed, is a modified version of multipurpose wheel loader. The frame of machine is original, but the control system, electronics and hydraulics have been changed to enable the research of autonomous operations. The control system architecture of this autonomous machine is illustrated in Figure 1.

![Figure 1. Control system architecture of autonomous mobile machine.](image)

The hardware devices are located physically in two different locations. Visualization and operator computers are off-board computers and those are connected through network switch and over WLAN to on-board network switch. On-board computers and peripheral devices are also connected to the on-board switch. On-board computers and peripheral devices follows device architecture illustrated in Figure 1.

The Low level control device architecture consists a control of the actuators, such as hydrostatic drive pump, diesel engine and hydraulic valves of the machine. It also takes care of data logging of the Inertial Measurement Unit (IMU), central joint resolver, pressure and some other sensors. The Low level control is based on six EPEC embedded vehicle computers and they communicate with the middle level control through four CAN buses. The Implementation of CAN buses follow CANopen standardization, which enables fluent CAN network management and consistency handling.

The Middle level control, including navigation, path planning and network interfaces with Real-Time Kinematic Global Navigation Satellite System (RTK-GNSS) has been implemented on an industrial embedded PC using Matlab / xPC Target as an operating system. The control of the servo including electric motor drive and encoder measurement were also implemented in this level through CAN bus.
The High level control of device architecture is also implemented on an industrial embedded PC. This PC communicates with the Middle level using UDP protocol. This protocol is also used in data transfer from the 2D laser scanner to Middle level control to ensure real-time data transfer of the laser scanner data.

3. ELECTRIC MOTOR DRIVES AND 2D LASER SCANNER

3.1. Electric motors

In general, two different type of DC motor exists. Brushed direct current (DC) motors and brushless direct current (BLDC) motors are the most typical ones. BLDC motors are typically driven by three phase conductors and phase voltages are generated by braking voltage of the intermediate circuit. In accurate position and speed electric motor drives BLDC motors are more common. [4] Thus, BLDC motor is used in this case. A ceramic planetary gear was used for the reduction of output rotation speed and to increase output torque.

3.2. Feedback sensors

Accurate and not (or in practice minimum and constantly) delayed measurement of the rotation angle is essential to enable later data fusion with laser scanner range data for creating a 3D point cloud data. This is more important than for example an accuracy of the position or speed control of the servo, because in these cases small error or delay is not reflected to the 3D point cloud calculation.

DC motors uses typically built-in brushed commutator for commutation but the selected BLDC motor requires external sensor to sense angle and speed of rotor for commutation. The commutation type depends on sensor type and it can be sensorless commutation, six-step commutation or sinusoidal commutation. Sensorless commutation doesn’t use sensors at all, six-step commutation uses Hall sensor and sine commutation uses for example incremental encoder for commutation. Sine commutation is the most recommended commutation type if application requires constant torque generation over whole rotation speed range [7]. Thus an external incremental encoder was selected for measuring speed of electric motor for speed control purposes. Measurement of incremental encoder is relative measurement and it is referenced to a certain reference rotation angle of rotor and it can’t be directly used to rotation angle measurement. This is because calibration, i.e. finding the zero position, of the sensor is needed every time when the sensor is started. Therefore an external absolute encoder was selected for measuring rotation angle despite the fact that absolute encoders are typically more expensive than incremental encoders because the internal structure of absolute encoder is more complex. Because data fusion with laser scanner range data requires accurate measurement of laser scanner rotation angle, rotation angle measurement is made directly from the load side. Thus the inaccuracy coming from the backlash of the planetary gear can be avoided.

3.3. Control system of electric motor

In this paper control system of electric motor controller bases on two control loops as illustrated in Figure 2.
Two control loops were used because the system includes planetary gear that brings backlash to system under control. Furthermore, backlash causes delay in control system and it might have an effect to system stability. Planetary gear is needed because it is hard to estimate precisely torque that load requires.

Purpose of auxiliary control loop is to stabilize, define damping and dynamic behavior of the system. Sensor in auxiliary control loop is incremental encoder that is located back end of the electric motor. Primary control loop uses absolute encoder as a feedback sensor. Primary control loop controls load angle of rotation.

All controllers in control system are implemented to electric motor controller as discrete time controllers. Current regulator is PI controller with 10 kHz sampling time. Speed controller is implemented as a PI controller with speed and acceleration feed forward. Position controller is implemented as PID controller with speed and acceleration feed forward. Sampling times for both controllers are 1 kHz. In control system speed feed forward can compensate speed-dependent friction that is caused by bearing among other things. Acceleration feed forward provides more current in cases when one need high acceleration or load inertia is high. [22] Electric motor controller is delivered with ready-made software that can be used for controller tuning. It was found out that software makes similar experiments that Ziegler-Nichols experiments [5] that uses step response and oscillation methods in controller tuning, but exact methods for tuning and parameterization of controllers remained unknown.

3.4. 2D laser scanners

The purpose of electric motor drive is to rotate continuously a 2D laser scanner. The scanner type was predefined for this hardware by the end user and it is SICK LMS111. Table 1 presents the most relevant properties of the scanner.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Angle resolution</td>
<td>0.25° / 0.5°</td>
</tr>
<tr>
<td>Output frequency</td>
<td>25 Hz / 50 Hz</td>
</tr>
<tr>
<td>Operating range</td>
<td>max 20 m</td>
</tr>
<tr>
<td>Field of view (FOV)</td>
<td>max 270°</td>
</tr>
<tr>
<td>Systematic error</td>
<td>± 30 mm</td>
</tr>
<tr>
<td>Statistical error</td>
<td>Typical 12 mm</td>
</tr>
</tbody>
</table>
The laser scanner and its final installation position on the roof of autonomous machine are presented in Figure 3.

Figure 3. 2D laser scanner (SICK LMS111-10100) installed on autonomous mobile machine.

The operating principle of the 2D laser scanner is based on time-of-flight (TOF). 2D laser scanner gives range and intensity values as a measurement values from measured plane. Measured range values are relative to rotating mirror of the 2D laser scanner and for every measured point one can get also intensity value [6].

4. HARDWARE AND SOFTWARE OF THE 3D LASER SCANNER

System level requirements based on simulations were defined as follows: movement range 100 degrees, field of view 140 degrees at the whole movement range, maximum rotation speed 40 degrees/s. Total mass of the load of the electric motor was about 2.3 kg and calculation of the inertia of the load was based on this mass, dimensions of the laser scanner and assuming a homogeneous mass distribution. Based on this data a Maxon EC-max 30 was chosen for an electric motor drive and planetary gear. The electric motor has parameters as tabulated in Table 2.

Electric motor drive contains planetary gear, so electric motor controller must support two control loops. In rotation speed loop incremental encoder with 1000 counts per turn (CPT) and index channel was used. In position control loop single turn absolute encoder with 16 bit resolution was selected. Purpose of shunt regulator is to prevent current flowing back to power source. Returning current can cause power source output voltage to rise.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value / Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal voltage</td>
<td>24 V</td>
</tr>
<tr>
<td>No load speed</td>
<td>9250 rpm</td>
</tr>
<tr>
<td>No load current</td>
<td>123 mA</td>
</tr>
<tr>
<td>Nominal speed</td>
<td>7230 rpm</td>
</tr>
<tr>
<td>Nominal torque</td>
<td>33.6 mNm</td>
</tr>
<tr>
<td>Nominal current</td>
<td>1.48 A</td>
</tr>
<tr>
<td>Speed / torque gradient</td>
<td>59.1 rpm / mNm</td>
</tr>
<tr>
<td>Speed constant</td>
<td>393 rpm / V</td>
</tr>
<tr>
<td>Torque constant</td>
<td>24.3 mNm / A</td>
</tr>
</tbody>
</table>

Table 3 presents all components that were needed for the implementation of electric motor drive.
Table 3. Components of the electric motor drive.

<table>
<thead>
<tr>
<th>Component</th>
<th>Manufacturer</th>
<th>Component type</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric motor</td>
<td>Maxon Motor Ag</td>
<td>EPOS2 50/5</td>
<td>Support for dual loop control</td>
</tr>
<tr>
<td>控制器</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Incremental encoder</td>
<td>Maxon Motor Ag</td>
<td>Encoder MR Type ML</td>
<td>1000 CPT with index channel</td>
</tr>
<tr>
<td>光栅编码器</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Absolute encoder</td>
<td>Scancon Industrial</td>
<td>SAG-S101G-0016-C06S-PAL</td>
<td>Resolution 16 bit, SSI interface to motor controller</td>
</tr>
<tr>
<td>Encoder</td>
<td>Encoders</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electric motor</td>
<td>Maxon Motor Ag</td>
<td>EC-max 30</td>
<td>Build-in Hall sensors</td>
</tr>
<tr>
<td>电动机</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Planetary gear</td>
<td>Maxon Motor Ag</td>
<td>Ceramic Planetary</td>
<td>Gear ratio 1181:1</td>
</tr>
<tr>
<td>齿轮</td>
<td>Gearhead GP 32 C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shunt regulator</td>
<td>Maxon Motor Ag</td>
<td>DSR 50/5</td>
<td>-</td>
</tr>
<tr>
<td>分流器</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Choke Module</td>
<td>Maxon Motor Ag</td>
<td>Choke module</td>
<td>-</td>
</tr>
</tbody>
</table>

Mechanics of hardware consists of two different units. Control unit contains all components that are closely related to electric motor controller and rotation unit contains all components that are closely related to electric motor and sensors. Between these two units exists only wiring for electric motor and sensors. Case of electric motor controller was not enough for demanding environment. Thus, case was improved additional aluminum case. This aluminum case is easy to move and install to any mobile machine. This control unit with panel connectors is illustrated in Figure 4.

Figure 4. Control unit of electric motor (left) and the schematics and a photo of the 2D laser scanner rotation unit without casing (right).

2D laser scanner is supported from both sides with slide bearing. These slide bearings are in yellow in Figure 4. Slide bearings support mechanical structure so that axial movement of the 2D laser scanner is prohibited. Other purpose of slide bearings is to tolerate radial load. Slide bearing material is brass alloy and axle material against this brass alloy is Hydax 15 steel.

Mechanical structure and manufacturing technique might cause axial misalignment, so component that fit axial misalignment is needed. Mechanical coupling on both side of the 2D laser scanner make this axial
misalignment fitting. In addition, mechanical coupling forward rotational motion and torque generated by the electric motor and planetary gear. Mechanical coupling fit axis of planetary gear and axis of mounting patch on left side in Figure 4. Furthermore, the other mechanical coupling fit axis of mounting patch and axis of the absolute encoder on the right side.

The control system of the mobile machine runs on Middle level control. The electric motor drive and 2D laser scanner are controlled by smaller control unit seen in Figure 4. That was integrated to existing control system. Execution of the developed software follows flow chart as illustrated in Figure 5.

![Flow chart of developed software.](image)

In initialization phase the electric motor controller and 2D laser scanner are parametrized, devices are started and whole system goes to waiting state. In waiting state hardware wait command from the control system. In initialization phase of the 2D laser scanner FOV and angle resolution are set. During initialization phase of the electric motor controller state machines are set to right state and periodic SYNC is set to CAN bus. In addition, software is set to receive rotation angle and speed data from the electric motor controller. Heartbeat receiving was also implemented.

Control system of the hardware implements two operating modes. “Drive to position” drives the 2D laser scanner to certain angle according to given parameters and “Sweep between angles” rotates the 2D laser scanner between certain angles with given parameters. Both operating modes uses electric motor controller internal operating mode “Profile Position Mode” [7].

Figure 6 presents basic verification measurements of the electric motor drive.
The measurements in Figure 6 illustrate how position (left) and angular speed (right) behave over one movement cycle. One can clearly see that profiles are quite smooth over time and only small angular speed ripple exists.

Implementation of networks on the mobile machine bases on using CAN and Ethernet networks. Developed hardware uses both networks for transferring measurement data. Thus, measurement data must be synchronized in time domain some way. One method for time synchronization is time stamping. Various devices can add time stamp to measurement data to be send. For example, 2D laser scanner and electric motor controller can add time stamp to measured data when devices send measured data to control system. Based on these time stamps measurement data can be time synchronized. This requires that clocks of both devices will be synchronized frequently enough to prevent time drifting.

Some devices don’t provide time stamp to measurement data or it is not usable for some reason. In this case one must arrange time synchronization some other way. Thus, deeper understanding behavior of networks and protocols are needed. Time stamping can be arranged such way that receiving instance of the measurement data add time stamp to the measurement data. This instance in this context is the Middle level control. This Time of Arrival (TOA) is useful when there is not much data transfer lag and small time synchronization error is allowed. With this hardware time stamping is done following way. 2D laser scanner sends measured data with constant time interval which is 25 Hz or 50 Hz. Measured data is transferred from 2D laser scanner to Middle level control through the Ethernet. Middle level control will then send SYNC request to CAN bus. Electric motor controller responds to this SYNC request with data that contains rotation angle and rotation speed.

According to datasheet, output of the 2D laser scanner is real time output [6]. Thus, this is assumed to work so that the 2D laser scanner measures range data and sends measured values to Middle level control through UDP with low network latency right after measurement. This latency time in data transfer in Ethernet network is assumed to be negligible. We estimate that this latency time is less than one millisecond.

If UDP packet send by the 2D laser scanner arrives to Middle level control at time instant txPC then worst case scenario is that SYNC request is send to CAN bus from Middle level control at time instant txPC + 1 ms. Electric motor controller responds to SYNC request quite fast. Based on CAN analyzer measurement time that goes from SYNC request to response from the electric motor controller is less than 500 µs. In this context we assume that the electric motor controller works as specified in CANopen standards so within this measured time happens reaction to SYNC request, so sensors are read, data is packed to PDO and PDO is send back to Middle level control. If SYNC request is send at time instant tSYNC then response is available during time instant tSYNC + 1 ms. When one takes into account all synchronization error sources, total synchronization error is around 2 ms and it is caused by data transfer delay, program execution step size and reading sensor to Middle level control.

Internal operation of the 2D laser scanner affect also to synchronization error. Internal rotating mirror of the 2D laser scanner rotates with 25 Hz or 50 Hz so one cycle takes 40 ms or 20 ms. UDP packet that contains measured data is send at time instance tS. This time instance is in area where 2D laser scanner is not making measurement. This region is in blue in Figure X. One can’t know this time instance exactly. Last measured point is measured at time instance tM and this time instance differs fundamentally from time instance tS. Time difference between these two time instances is around 5-10 ms when measurement frequency is 50 Hz and FOV is 140 degrees. Figure X illustrates region in red that the 2D laser scanner uses in this application. Time instance tM illustrated to Figure X is the place where last measurement is done. When one takes into account all mentioned error sources above between last measured point and rotation angle of the 2D laser scanner exist 15 ms time synchronization error at maximum.

Time difference between measured range values and rotation angle causes time synchronization problem that affect to laser map quality because time synchronization problem causes motion synchronization problem. In this context motion synchronization problem means that there is error between position of the rotating mirror and rotation angle of the servo of the 2D laser scanner. Thus, measured range values are not
from there where rotation angle measured express them to be. This error is related the angular speed of the servo, bigger angular speed means larger error.

Figure 7. 2D FOV of the laser scanner used in the mapping (red) and region when laser scanner does the UDP packet sending (blue).

Operating with this hardware requires coordinate frame transform. Coordinate frame transform is needed from rotating mirror of the 2D laser scanner to axle of the electric motor drive. The related frame transformations are calculated in [8] where also the motion synchronization is taken into account by issuing uncertainty for the frame transformation. Developed hardware requires calibration because mechanical installation affect pose of the developed hardware, for example. Methods for calibration are not included in this paper but one must aware of this issue. Calibration method in our case is described in [8].

5. MAPPING

The presented low-cost 3D LIDAR has been used for mapping and path planning for autonomous mobile machines. The mapping method uses servo angle and localization data to transform laser range data into 3D point cloud in global frame. Each point is also assigned with covariance data. The covariance for each point is calculated from the uncertainty data that is related to each coordinate transformation. The uncertainties propagate from each transformation that is required to transform laser range measurement to 3D point.

Point cloud is then processed to height map and further into obstacle map. Mapping method and experiments are presented in [8]. Example point cloud and related obstacle map are shown in Figure 8. The point cloud was created with presented low-cost 3D LIDAR system installed on GIM-machine that is modified multipurpose loader presented in Figure 8. The path driven with GIM machine is presented as red line in point cloud data. One can see that the red area in point cloud data is recognized as occupied area in obstacle map. The produced obstacle map can then be used for path planning as presented in [9].
6. CONCLUSIONS

In this paper a development of the electric motor drive for the 2D laser scanner is addressed. The main focus in this research work was to build a low cost solution using standard components available in the market. This paper discusses the control system architecture, hardware and software solutions of the build 3D LIDAR system.

The build 3D LIDAR system was successfully used in the experimental mobile work machine to create obstacle map of the surrounding area for path planning purposes. Some experiments of the mapping were presented in the paper.

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REFERENCES


MODELLING STUDY OF AN OPTIMUM ELECTRIC MOTOR FOR DIRECTLY DRIVEN HYDRAULIC PUMP EMULATOR IN REAL-TIME HIL-SIMULATION

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         tatiana.minav@aalto.fi

ABSTRACT

Hardware-in-the-loop (HIL) setup is used in research, since it allows fast, accurate and easily repeatable testing of hybrid topologies of non-road mobile machinery under realistic conditions. In a previous study, it was shown that a virtual model of a hydraulic machine in a HIL setup can be used for testing by emulating a directly driven hydraulic pump with an electric motor. As the used setup did not completely fulfil the requirements, criterion for choosing a new electric motor was suggested. Therefore, in this research set of initial solution motor parameters to fulfil a minimum settling time of $T_s = 0.001$ seconds for hydraulic pump emulator electric motor torque response was made based on analytical design of a coreless motor. A Matlab/Simulink model of the optimum electric motor for directly driven hydraulic pump emulation was made. According to the simulation, the capability of the designed electric motor in following a 1 kHz torque reference signal was proven.

KEYWORDS: Modelling, Hardware-in-the-loop, hydraulics, direct-drive, electric drive, emulation, Non-road mobile machinery.

1. INTRODUCTION

With continuation of tightening of CO₂ emission rules, new tools for the rapid testing and development of hybrid drive systems in non-road mobile machinery (NRMM) are required [1]. Practical tests of hybrid drive systems generally require a high level of complexity in combining hydraulic and electric equipment. However, such conditions can easily be avoided by using Hardware-in-Loop (HIL) simulations. HIL-simulation is a technique that is used in the development and test of complex real-time systems in comprehensive, cost-effective and a repeatable way.

Various approaches using HIL and Software-in-loop simulations have been discussed in the literature. For instance, HIL-simulations are widely used in airplane applications [2, 3], development of vehicles [4,5], power steering [6,7] and control systems [8,9]. It is observed that real hydraulic components are used as a part of simulated systems [10-13]. Therefore, the idea of a flexible HIL-setup without real hydraulic and mechanical components was developed in [14, 15] with use of a real-time software [16], which allows easy and rapid changes to the model and test setup. According to [17], a virtual model of a hydraulic machine in a HIL
setup can be used for testing by emulating the hydraulic pump with an electric motor. The components of this kind of test setup need to be chosen carefully: optimized for data transfer speed, small delays and suitable refresh rates.

Figure 1 illustrates the suggested HIL-setup [14, 15] containing a real-time simulator, electric drive (motor and frequency converter) as a hydraulic pump emulator and an electric drive under test for a hybrid NRMM drive. The real-time simulator (RTS) is used to simulate the hydraulic and mechanical components of the NRMM via control signals connected directly to the hydraulic pump emulator. As a result, the hydraulic pump emulator is seen as a load by the tested electric drive. The tested electric drive consists of an electric motor and/or a frequency converter. The electric drive being tested can be controlled either directly by the operator (via human machine interface (HMI)) of the NRMM or by a signals from the RTS.

![Figure 1. HIL setup with an electric motor for emulating the behaviour of a hydraulic pump. a) Computer with real-time simulator and HMI, Hydraulic pump emulator: b) frequency converter and c) electric motor used for emulating pump dynamics. Electric drive under test d) electric motor and e) frequency converter under test.](image)

Figure 2 illustrates example of a direct driven electro-hydraulic circuit of a test setup realised in real-time software, where dark grey area is the electric motor under test (Figure 1d), light grey area is the real-time simulator (Figure 1a) and the dotted area is the hydraulic pump emulator (Figure 1b and 1c).

![Figure 2. An example of a direct driven electro-hydraulic circuit realised partly in real-time software: Dark grey area-electric motor under test, light grey area- real-time simulator, and dotted area- hydraulic pump emulator [14].](image)

Previous study showed that the electric motor that was used was not sufficient in all aspects for this emulation purpose. Also a search of suitable electric motor on the market proved the need for a totally new design. Therefore, this research concentrates on finding sufficient parameters for the initial design of the
hydraulic-pump-emulator electric motor. Initial boundary conditions and requirements of motor characteristics used are obtained from a previous study [19].

In the following sections the refresh rate/settling time requirement of the hydraulic pump emulator is discussed and boundary conditions are created. Initial analytical design of electric machine for hydraulic pump emulator is performed to fulfil the boundary conditions and initial requirements. A Simulink model based on the motor analysis is made. Final sections contain discussion of results and conclusions.

2. INITIAL REQUIREMENTS

According to [14, 15], one of the main criterions for choosing an electric motor to be an hydraulic pump emulator in a HIL-setup is the refresh rate in torque control of electrical machine. In this research, refresh rate is the frequency of updates of a piece of data from an external source. In [15] refresh rates of the different components of the HIL-setup were obtained from datasheets, measurements, by calculation and estimation.

In [14, 15] the real-time simulator has the highest refresh rate of 1 kHz of its external connections. Therefore, the refresh rate of the hydraulic pump emulator should be as close as possible to that of the real-time simulator. Target for the hydraulic pump emulator is set to be \( f_{\text{refresh}} \geq 1 \text{ kHz} \).

The electric machine refresh rate estimation is based on the step response settling time of open-loop transfer function of a simplified force controlled DC-motor [18]. Transfer function is based on equivalent electrical circuit for a DC-motor. Following simplifications are made: Since the shaft compliance is evident, the mechanical time constants can be neglected in torque controlled mode and the dynamic properties of the electric motor are fully dominated by the electrical time constant. Therefore, open-loop transfer function of a simplified force controlled DC-motor is:

\[
W(s) = \frac{K_m}{R_a (\tau_e s + 1)},
\]

where \( L_{\text{coil}} \) is coil inductance, \( R_a \) is coil resistance, \( \tau_e \) is electrical time constant and \( K_m \) is torque constant

\[
\tau_e = \frac{L_{\text{coil}}}{R_a},
\]

Step response analysis was used to calculate the torque refresh rate based on settling time (Equation 3). Therefore, target for hydraulic pump emulator is to meet settling time of \( T_s \geq 0.001 \text{ seconds} \).

\[
f_{\text{refresh}} = \frac{1}{T_s},
\]

where \( f_{\text{refresh}} \) is the maximum digital signal frequency, \( T_s \) is settling time.

To find a range of motor parameter values \((L_{\text{coil}}, R_a, K_m)\) that fulfil the initial requirements of settling time \( T_s \geq 0.001 \text{ seconds} \) [19], a Matlab code was made. Purpose of this code was to test if a sensible range of the three parameters \((L_{\text{coil}}, R_a, K_m)\) to realise the settling time could be found.

Figure 3 shows results of Matlab calculations. It represents all possible solutions (variation of phase resistance \( R_a \), coil inductance \( L_{\text{coil}} \) and torque constant \( K_m \)) that meet the requirements of the hydraulic-pump-emulator electric motor (Figure 1c) step response settling time.
According to [19] and Figure 3 a low inductance value suggests using design of electrical machine based on permanent magnets and coreless stator structure and axial flux topology. Coreless design allows to obtain a low inductance value, and reduced ripple torque. Central stator and double rotor axial flux machine permits an adequate air gap flux density, to reach the high torque density needed. A realistic narrower area of boundary conditions were chosen from Figure 3 that also fit the minimum settling time of $T_s = 0.001$ seconds for further investigation.

The chosen boundaries of motor parameter values ($L_{\text{coil}}$, $R_a$, $K_m$) to fit minimum settling time of $T_s = 0.001$ seconds are presented in Table 1:

**Table 1. Chosen boundary conditions for the hydraulic-pump-emulator electric motor**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Min Value, [Units]</th>
<th>Maximum Value, [Units]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{\text{coil}}$</td>
<td>0.00086 [H]</td>
<td>0.0086 [H]</td>
</tr>
<tr>
<td>$R_a$</td>
<td>1.3528 [Ohm]</td>
<td>13.528 [Ohm]</td>
</tr>
<tr>
<td>$K_m$</td>
<td>2.2053 [Nm/A]</td>
<td>6.6159 [Nm/A]</td>
</tr>
</tbody>
</table>

Other initial requirements (Table 2) for the hydraulic-pump-emulator were chosen based on the electric motor used in the laboratory tests [14, 15], assuming use in a boom application similar to the one used in earlier research.

**Table 2. Chosen initial requirements for hydraulic-pump-emulator electric motor**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value, [Units]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>10, [kW]</td>
</tr>
<tr>
<td>Motor rated torque</td>
<td>30, [Nm]</td>
</tr>
<tr>
<td>Maximum speed</td>
<td>3000, [rpm]</td>
</tr>
<tr>
<td>Voltage source</td>
<td>500, [V]</td>
</tr>
<tr>
<td>Number of pole pairs</td>
<td>4</td>
</tr>
<tr>
<td>Peak air gap flux density</td>
<td>0.5, [T]</td>
</tr>
<tr>
<td>Winding factor</td>
<td>1</td>
</tr>
<tr>
<td>Stator number of phases</td>
<td>3</td>
</tr>
</tbody>
</table>
Following section contains analytical study of motor design to find set of parameters to fulfill all requirements and boundary conditions from Table 1 and 2.

3. ANALYTICAL ELECTRIC MOTOR STUDY

As was stated early, design of electrical machine based on permanent magnets and coreless stator structure and axial flux topology will be used. Based on following Equations, a set of electrical machines are designed in order to meet the boundary conditions established on Table 1 and initial requirements on Table 2.

Mutual flux linkages produced by the stator current distribution are represented by the magnetizing inductance. This parameter can be calculated for axial flux machine configuration [20] taking into account infinity iron permeability

\[ L_{\text{coil}} = \frac{m \cdot D_s}{4 \cdot \pi \cdot p^2 \cdot \delta_{id}} (1 - k^2)(k_w \cdot N)^2 \]  

where \( \delta_{id} \) represents the ideal equivalent airgap length considering the physical airgap length, \( h_s \) is stator depth and the magnets height, \( p \) is number of pole pairs, \( k_w \) is winding factor, \( N \) is number of turns per phase, \( m \) is stator number of phases, \( k \) is diameter ratio between \( D_s \) stator inner and \( D_e \) outer diameter.

And torque expression for this motor topology is:

\[ T = K_m I, \]  

where \( I \) is stator current. \( K_m \) torque constant results

\[ K_m = \frac{\sqrt{2}}{8} m(k_w N)D_s^2(1 - k^2)B_g, \]

where \( B_g \) is peak airgap flux density.

Figure 4 shows results of Matlab algorithm based on Equations 4, 5 and 6. Figure 4 represents all possible solutions (variation of phase resistance \( R_a \), coil inductance \( L_{\text{coil}} \) and torque constant \( K_m \)) of coreless electrical machine design. Only a few of designed electrical machines are feasible considering the voltage source and cooling requirements. On Figure 4, the region where these constraints are fulfilled (less than 500 V line to line voltage and RMS current density 5 A/mm² from Table 2), is highlighted with blue. The electrical machine parameters and dimensions computed based on Equations 4, 5 and 6 are listed on Table 3. These numbers fulfill all boundary conditions from Table 1 and 2, which include settling time and torque constant.

Figure 4. 3D graph of electric motor parameter solutions for Ts <= 0.001 seconds.
Table 3. Initial design electrical machine parameters for hydraulic-pump-emulator

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Induced voltage</td>
<td>277, [V]</td>
</tr>
<tr>
<td>Coil inductance</td>
<td>1.3, [mH]</td>
</tr>
<tr>
<td>Torque constant</td>
<td>2.5, [Nm/A]</td>
</tr>
<tr>
<td>RMS stator current</td>
<td>12, [A]</td>
</tr>
<tr>
<td>Phase resistance</td>
<td>0.6, [Ohm]</td>
</tr>
<tr>
<td>RMS current density</td>
<td>4.1, [A/mm^2]</td>
</tr>
</tbody>
</table>

It can be seen that phase resistance is not large enough and does not meet the initial requirements; therefore, external resistance must be included.

\[ R_{\text{ext}} = \frac{L_{\text{coil}}}{T_s} - R_a \]  

Using the values of the Table 3 in to Eq. 7, \( R_{\text{ext}} = 0.7 \) Ohm is the minimal external resistance required.

4. MODELLING AND RESULTS

Figure 5 illustrates Matlab/Simulink model of an hydraulic pump emulator electric motor and drive created with Matlab/Simulink R2014a using Simscape toolbox.

In the model a permanent magnet synchronous motor (PMSM) is connected to a rigid wall with a flexible shaft as described in details in [15]. The motor is controlled with a drive using torque control mode. The flexible shaft includes small inertias and velocity and torque sensors in both motor and wall end.

For performing simulation model analysis, designed motor parameters were used from section 3. Parameters of the drive were modified in order for the controller to be able to keep the output torque stable.

As a reference torque value input to the drive a sampled Gaussian noise was generated. Figure 6 illustrates this random input signal which was used as the torque reference signal. The signal values changes...
randomly with a frequency of 1 kHz. This signal is a representation of a reference signal that can be obtained from a real-time simulator like [16] to be followed with the newly designed motor.

Figure 6. Input signal

Figure 7 shows the torque output from the PMSM shaft imposed on the reference signal.

Figure 7. Torque response

From the Figure 7 it can be seen that the torque follows the reference signal with the same frequency of changing value though with a minimal delay. In the beginning of the signal the motor starts from steady state of 0 as reference value and is able to start following the signal at approximately 0.0005 seconds. It can be seen that the motor with the chosen parameters can fulfil the requirement of being able to follow a 1 kHz torque reference signal.

5. DISCUSSION

According to the simulation results the electric motor was able to follow the 1 kHz torque reference signal. The chosen parameters were confirmed to be acceptable. The total performance of the system was heavily dependent also on the controller and drive performance. The controller, including the frequency converter should be designed and tested to match the performance requirements of the load emulation in a real-time simulator. A more accurate design and modelling of the electric motor should also be made to find out more about its mechanical and thermal characteristics.

6. CONCLUSION

In this study, the initial parameters of this hydraulic pump emulator electric motor for HIL setup was chosen based on refresh rate of the electric motor’s torque control model. Simulink modelling of a hydraulic pump emulator using optimum electric motor parameters based on analytical design was made. Simulation proved
that the designed electric motor can follow a 1 kHz torque reference signal. For future research, the design of the frequency converter and controller should also be taken into consideration as they had significant impact on the system performance.

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MODELLING OF A SECONDARY CONTROLLED SIX-WHEEL PENDULUM ARM FORWARDER

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ABSTRACT

One of the major concerns in the forest industry is the impact on the soil caused by the forest machines during harvesting, where damage can have a negative impact on growth at replanting for example. Another concern is the working environment of the operator. Both these issues have a negative impact on productivity. A new six-wheel pendulum arm forwarder is being developed within a collaborative research project. The new forwarder aims to reduce soil damage by means of an even pressure distribution and smooth torque control. This paper presents the first step in the development of the driveline, where a secondary control approach is chosen for its ability to control the motion of each wheel individually. Simulation models of both vehicle and driveline have been constructed developed, partly for the development of the control strategy, and partly for evaluation. A speed control concept and a torque control concept have both been evaluated for different scenarios with regard to their ability to reduce wheel slip. Results have shown that a velocity control approach is more sensitive to kinematic model accuracy while wheel slip is handled automatically. A torque control approach is more robust towards model accuracy while the reduction of slip is dependent on an accurate model.

KEYWORDS: Secondary control, forest machinery, anti-slip control, traction control

1 INTRODUCTION

Compared to a traditional forwarder, a forwarder with pendulum arms and individual control of each wheel will have clear benefits when it comes to the impact on the soil caused by the machines during harvesting. Within a collaborative research project between Skogforsk (the Forestry Research Institute of Sweden), Bosch Rexroth, Extractor AB (machine owner), Linköping University and The Royal Institute of Technology (KTH) in Stockholm, such a forwarder with six wheels is being developed, where the overall objective is to minimise soil damage and improve the working environment for the operator’s. The project initially began as a student project, followed by a Master thesis. More information on these can be found in [1] and [2]. One of the main reasons behind soil damage is wheel slip. Studies have been conducted to register the impact different forwarders have on the soil and can be found in [3] and [4].

While the traditional forwarder uses one pump, one motor and a mechanical linkage to drive the forwarder through sequential control of the hydraulic machines, this new forwarder comprises two hydraulic pumps and six hydraulic motors. Each hydraulic motor is attached to each wheel, allowing individual control of the motion of each wheel. Since each wheel is also attached to a pendulum arm, the height of each wheel can be freely adjusted with respect to the chassis, and each other, allowing for a levelling of the chassis and an even pressure
distribution between the wheels and the ground. These two properties should help to minimise damage to the soil. The complete chassis can also be utilized to reduce movements of the cab, thereby improving the working environment operator’s. A comparable machine is the el-forest, [5]. El-forest has some similarities to the forwarder presented in this paper, but has an electric driveline instead of an hydraulic driveline.

This paper deals with the development of the control strategy for the driveline. The driveline must be efficient while also minimising the impact on the soil. The paper looks into two different approaches for how to avoid wheel slip. Both concepts are based on a secondary control strategy due to its ability to control the torque or velocity on each wheel independently, without any throttling losses. Compared to the sequence-controlled transmission, the secondary control transmission can cope with wheel slip more easily since no additional hardware is needed, e.g. a differential lock. The first controller proposed is torque control; this concept has similarities to a differential. A differential brake can be implemented in software with known reference wheel speed. The second concept is speed control and can be seen as a differential lock with known speed on each wheel. The simulation results show the performance of the concepts for different cases, where the purpose is to show the suitability of a secondary control strategy for such a vehicle.

2 SYSTEM DESCRIPTION

The forwarder being developed has a total maximum weight of 31 tonnes, including a maximum load of 14 tonnes and is shown in figure 1(a). It consists of three bodies with two actuated steering joints and six wheels, each attached to a pendulum arm, as illustrated in figures 2(a) and 2(b). The transmission consists of two closed hydraulic circuits with one Bosch Rexroth A4 variable pump and three Bosch Rexroth A6 variable motors for each circuit, as shown in figure 1(b). The two pumps are powered by the same 6 cylinder, 360 bhp diesel engine. System parameters are stated in table 1. The pumps can go over centre. A gear is installed between each hydraulic motor and wheel to increase the output torque. Rotational velocity sensors are attached to each wheel and four pressure sensors measure the high and low pressure of each hydraulic circuit. In addition, the exact position of each pendulum arm can be calculated from the pendulum arms’ position sensors.

![Figure 1: The forwarder and the driveline simulated in the paper.](image-url)
(a) The two steering joints between the three bodies.  
(b) The wheel attached to the pendulum arm.

Figure 2: Kinematics of the forwarder’s three turning axes and the pendulum arm.

Table 1: System parameters for the simulated forwarder.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel motor power</td>
<td>$P = 268$ kW</td>
</tr>
<tr>
<td>Pump rotational speed</td>
<td>$n_p = 2000$ rpm</td>
</tr>
<tr>
<td>Pump displacement</td>
<td>$D_p = 140$ cm$^3$/rev</td>
</tr>
<tr>
<td>Secondary machine displacement</td>
<td>$D_m = 107$ cm$^3$/rev</td>
</tr>
<tr>
<td>Mass of the vehicle</td>
<td>$m = 31 000$ kg</td>
</tr>
<tr>
<td>System pressure, high pressure side</td>
<td>$p = 32$ MPa</td>
</tr>
<tr>
<td>System pressure, low pressure side</td>
<td>$p = 3$ MPa</td>
</tr>
<tr>
<td>Gear ratio</td>
<td>$U = 48.3$</td>
</tr>
<tr>
<td>Wheel radius</td>
<td>$r = 0.74$ m</td>
</tr>
</tbody>
</table>

3 CONTROL

Secondary control is a technique known for high dynamics and the possibility to recuperate energy and avoiding throttling losses. It was first developed in 1977 [6] and was investigated in [7] for example. Historically, one of the drawbacks has been the necessity for advanced electronic controllers, where a purely hydromechanical solution is not possible. However, this can not be seen as a drawback with the computational technology available today. A control strategy for a secondary controlled hydrostatic transmission was developed in [8].

The basic principle of secondary control is shown if figure 3(a). The primary and secondary machines are controlled separately in the sense that the pump is controlled to keep the system pressure constant, while the secondary machine directly controls the output, hence the name secondary control. The basic schematic figure in figure 3(a) shows an open circuit. The closed circuit architecture used in this paper is shown in figure 3(b). The systems work according to the same principle. However, with the closed circuit, the secondary machine does not need to swivel over zero to brake the rotational speed as in open circuit architecture. At braking the high pressure side is changed and the secondary machine begins to work as a pump, thus braking the vehicle.

To recover energy in the open circuit, the secondary machine needs to swivel over zero. In a closed circuit, however, energy can be recuperated even if the secondary machine does not swivel over zero through an arrangement of valves, similar to [9] for example. With the secondary control architecture, several drives can be added to the same pressure line with no load interference. This is the most important property for this work, while energy recuperation is not within the scope of this paper.

The simulated transmission has two separate hydraulic systems, each with one pump and three motors. Two different approaches for secondary control are evaluated. A torque controller adjusts the output torque on each wheel in order to maintain a certain vehicle speed. The wheel's rotational speed is defined by the
Figure 3: The secondary control in an open circuit and a closed circuit respectively. The pump controller controls the system pressure and the controller for the secondary machine controls the transmission output.

torque equilibrium of each wheel. The differential speed of the inner and outer wheel during cornering is therefore solved automatically. This concept requires separate handling of wheel slip, i.e. when a wheel loses grip. The second approach is a speed controller, which controls the rotational speed of each wheel. The differential velocity of the inner and outer wheel needs to be controlled in a proper manner, while the slip is solved automatically in this case.

Addition to the secondary controller, the source pump needs another control strategy to maintain a certain pressure level. The controllers are kept simple to be able to compare the systems in respect of the controllers themselves. The controllers are explained in the following sections.

3.1 Pressure control

Pressure controlled pumps were investigated by Palmberg et al. [10], where the pump with its hydro-mechanical controller was described by an inductance and a break frequency for the pump, together with the break frequency of the system volume, made up the complete control system. In this work, however, the pump is not of the pressure control type, but has an integrated electro-hydraulic displacement controller. A first order lag is assumed to represent the displacement control, with a time constant of 20 rad/s. The pressure control loop is then implemented in the software based on the measurement of the pressure. A schematic of the pump controller is shown in figure 4. The low pass filter represents the internal pump controller. A PI-controller is used to control the pressure. The pressure level is maintained at a constant 32 MPa. The maximum pressure fluctuation in the simulations is 1%.

Figure 4: Schematic of the pump controller.
3.2 Torque control concept

Figure 5 shows a schematic of the implemented torque controller. The outer loop and the controller, $C_D$, can be either a software implemented controller maintaining a certain vehicle speed or a driver controlling the vehicle speed. The velocity control error, equation (1), is to be minimised by controller $C_D$.

$$e(v_c) = v_{ref} - v_{current}$$ (1)

In this work, the outer controller is a PI-controller that calculates the drive torque needed to follow the reference velocity, $v_{ref}$. Initially, the total torque is divided equally over all six motors. The corresponding motor displacement setting for each motor is calculated in block $G_T$ according to equation (2),

$$\epsilon_i = \frac{1}{6} \frac{2 \pi T_{tot}}{D_m \Delta p} \quad \text{where} \quad i = 1 \ldots 6$$ (2)

where $D_m$ is the motor displacement and $\Delta p$ the system pressure. The system pressure, controlled by the pump controller, fluctuates in the simulation by a maximum of $\approx 1\%$, which means that $\epsilon_i$ is approximately proportional to $T_{tot}$.

The slip is controlled by an inner controller by comparing the actual rotational speed of each wheel with the calculated reference rotational speed. In the case of one wheel slipping, the rotational speed of that wheel will increase. If the difference against the model calculated speed is too large, the displacement setting of the corresponding motor will be decreased. The control rule for the motors’ displacement setting $\epsilon_{m,i}$ from the inner control loop is stated as:

$$\epsilon_{m,i} = \epsilon_i - \epsilon_s (\omega_i - \omega_{ref}) \quad \text{where} \quad i = 1 \ldots 6$$ (3)

where $\omega_i$ is the measured rotational velocity of the hydraulic motor and $\omega_{ref}$ the reference rotational speed. The calculation of the desired rotational speed of each wheel is based on a kinematic model of the vehicle and the reference vehicle velocity, see section 4.1. The velocity can be derived in different ways. In [11], the wheel slip is estimated using GPS signals and a similar estimation is needed for this vehicle. The estimation of the vehicle speed is not within the scope of this paper and the vehicle speed is therefore derived from a vehicle model. $\epsilon_s$ is a tuneable parameter, dependent on the amount of slip that is allowed. The choice of parameter value is also subject to expected model error.

As with the pump controller, a first order lag filter is implemented to represent the displacement control. The time constant is set to 20 Hz.

![Figure 5: Torque control of the transmission.](image-url)
3.3 Speed control concept

Figure 6 shows a schematic of the implemented speed controller. The outer loop can be seen as the driver, similar to the torque controller but the outcome is the reference speed sent to the kinematic model. The kinematic model calculates the reference velocity of each hydraulic motor. The inner control loop is a PI-controller.

The control of the slip is also handled by the inner control loop. The proportional gain and the integrating part can assume rather large values. As long as the change in wheel speed is included in the kinematic model, the reference speed will be correct. If there is a mismatch between the kinematic model and the actual vehicle kinematics, there will be a control issue when using large proportional gains and the integration part. This is shown in the paper by simulating a road bump, explained in detail in section 4.2. There is a trade-off between minimising slip and handling the model error in the vehicle kinematics. If the proportional gain is chosen to be small, a larger modelling error is allowed while at the same time more slip is allowed. Conversely, a large gain will minimise slip but require a more accurate model of the vehicle.

Note that this control strategy does not rely on measurements of the system pressure, as is the case with the torque controlled concept.

4 MODELLING

Developing models of the forwarder and subsystems under study facilitates the development of control strategies as well as giving a first indication of the performance of the vehicle and the suitability of the control concepts. A fairly simple vehicle model is used, but it covers the important aspects when studying the driveline. The driveline itself consists of the hydraulic machines. The losses in the hydraulic motors are minor leakage and a viscous friction coefficient. At this point, the diesel engine is not modelled and delivers a constant rotational speed. A slip model is implemented to investigate how the controllers behave and how they handle loss of traction in a wheel.

4.1 Vehicle model

For this paper’s analysis, the kinematic model is simplified. The vehicle has two articulated steering joints, see figure 2(a). The first body turns with angle \( \alpha \) and the last body’s steering angle \( \beta \) will follow the first body in such a way that no lateral slip occurs on each wheel. This means that the radii \( R_F \), \( R_M \) and \( R_R \), coincide at the same centre point during turning. The wheels perceive different speeds during turning and to prevent slip of
neither inner nor outer wheel the hydraulic motors’ controller has to react. The vehicle has different distances between the steering joints and the wheel axes, which makes the steering radius individual for each wagon to prevent slip. Note, this is only one possible steering condition. The driving transmission is the main focus in the paper and this simplified driving mode is satisfactory. Note that the speed controlled concept is able to steer the vehicle by actively controlling the differential velocity of the inner and outer wheel during cornering. There is no need for a steering mechanism. This is not the case in the torque controlled concept.

The resistive forces considered in the vehicle model are rolling resistance, \( F_r \), slope resistance, \( F_g \) and a disturbance force from a simulated bump, \( F_b \). The bump model is explained in the next section. The rolling resistance is calculated as in equation (4),
\[
F_r = C_r F_N
\]
where \( F_N \) is the normal force. The resistance is independent of the torque and it can be justified due to the low torque. The slope resistance \( F_g \) is calculated as:
\[
F_g = mg \sin(\gamma)
\]
The vehicle acceleration can now be calculated as:
\[
m\ddot{v} = \sum_{i=1}^{6} F_{m,i} - \sum_{i=1}^{6} F_{r,i} - F_g - F_b
\]
where \( F_{m,i} \) is the driving force for each hydraulic motor.

The model only considers longitudinal acceleration since only low speed manoeuvring is considered. The model is used both for the vehicle itself and the reference model. The vehicle model uses the torque from the hydraulic motors to calculate the driving force and with the kinematic constraints it calculates the velocity of the vehicle, \( v_v \). The kinematic model is used when estimating the hydraulic motor’s rotational speed in the speed controller. In this case, the kinematic constraints are used to calculate the wheel’s rotational speed, hence the reference rotational speed of the hydraulic motors, \( \omega_{ref,i} \).

### 4.2 Bump model

A bump is implemented in the simulation model to show how the two controllers behave under disturbance. The bump can be seen as a rock or a stump for example. The bump includes both a velocity change and a sudden change in torque demand. The wheel velocity change due to the road bump is implemented according to equation (7),
\[
\omega_{bump} = \frac{1}{\cos\left(\arctan\left(\frac{\pi H}{L} \sin\left(\frac{\delta L}{2} \pi\right)\right)\right)} \quad \text{where} \quad 0 \leq \delta \leq S
\]
\( H \) and \( L \) are the height and the length of the bump, respectively. The radius of the bump is never smaller than the wheel radius. The height of the bump is half of the vehicle’s levelling distance, i.e. 0.25 meters. Figure 7 shows the bump’s curvature.

### 4.3 Slip model

The force transferred from the vehicle to the ground depends on the slip ratio, i.e. the velocity difference between the tyres and the vehicle. The slip ratio, \( \lambda \), is calculated as equation (8),
\[
\lambda_i = \frac{v_v - \omega_i R}{\max(v_v, \omega_i R)}
\]
\( \lambda \) can take on values between -1 and 1. The equation is undefined when both the vehicle velocity and the wheel velocity are zero. For simulation purposes the slip ratio is set to zero at this point.
Figure 7: The simulated bump’s curvature and its radius. The height is 0.25 meters and the radius is minimum 0.8 meters.

Pacejka’s model [13] is commonly used for modelling the tyre friction, equation (9). The parameters $D, C, B$ and $E$ are set differently depending on surface and conditions. In this paper the exact behaviour of the contact between ground and tyre is not important, but the model should capture the loss of grip in the forest. The main parameter is $D$ and defines the maximum friction coefficient.

$$\mu(\lambda_i) = D \sin(C \arctan(B\lambda_i - E(B\lambda_i - \arctan(B\lambda_i)))) \quad i = 1, 2, \ldots, 6$$

Figure 8 shows the friction characteristics for $D$ values equal to 0.5 and 0.04. The two values are used here to show the difference in principle between the control strategies.

Figure 8: Tire friction characteristics as function of slip ratio between ground and tire. Solid line is the characteristic for good traction with parameter $D = 0.5$. The dashed line represents the case with parameter $D$ lowered to 0.04.

The force transferred between the tire and ground is calculated as in equation (10). Only forward movement is considered and hence all the possible transferable force is used to accelerate the vehicle.

$$F_f = F_N \mu(\lambda_i)$$
Fictitious driving scenarios have been used to test the two control strategies and highlight the differences between them.

**Scenario 1 - Speed change and slip**  This scenario tests the two control strategies when traction is reduced on the right rear wheel. Figure 9 shows the vehicle speed during scenario 1. At 3 seconds, the vehicle accelerates in approximately 2 seconds from 0.5 m/s to 1.5 m/s and at 10 seconds the vehicle decelerates to 0.5 m/s again. The vehicle has no brakes, which means that the displacement will be reduced during deceleration. The machines is not made for negative displacement settings, but, as stated earlier, the system can be modified with valves to give a solution to this case.

**Scenario 2 - Bump**  The right front wheel experiences a bump between 7 and 9 seconds, as shown in figure 7. The vehicle speed is 1.5 m/s when the bump is simulated. This scenario tests the control concepts at a torque disturbance. The scenario also tests the required accuracy of the kinematic model.

**Scenario 3 - Turn**  The turning angle is the same throughout the simulation and the first wagon’s steering angle is set to 10°. The third wagon has a smaller turning angle due to the kinematics of the vehicle. The scenario tests the control strategy dependencies of a kinematic model.

The results of the speed and torque control strategy are presented in section 5.1 and 5.2 respectively.

### 5.1 Speed control concept

The speed controller is dependent on the accuracy of the kinematic model of the vehicle when a speed difference between the wheels occurs. The gain $C_\omega$ is relevant for how the slip protection works but with a high gain the kinematic model has to be accurate, as explained earlier.

Figure 10 shows how the motors’ displacement setting $\varepsilon$ and the corresponding slip ratio change when one wheel has lower traction than all other wheels, scenario 1. The solid line shows the reference conditions when all wheels have equal traction. The motor’s displacement setting is reduced on the wheel which experience reduced traction to reduce the slip. When the gain is reduced, the controller can not maintain the reference speed and hence the wheel has increased slip. Acceleration and deceleration also increase the slip due to the increased torque demand, i.e. the characteristics of the slip model.

Results for scenario 2 are shown in figure 11. The figure shows how the slip and displacement settings change when the right front wheel experiences a bump. Two different gains in $C_\omega$ are also shown for these simulations.
Figure 10: Scenario 1 with speed change and right rear wheel with less traction. The solid line is the reference curve when all wheels have the same traction. Dashed lines show the wheel with traction while dashed/dotted lines show the wheel with reduced traction.

Figure 11: Scenario 2 with a bump on the right front wheel. The solid lines show results when the bump is included in the kinematic model; the highest values of the solid lines are for the wheel which experiences the bump. The other lines show the results when the bump is not included in the kinematic model. The dashed lines are the wheel which experiences a bump and dashed/dotted line is one of the other wheels.
If the kinematic model used to calculate reference speeds matches the actual kinematics of the vehicle, the gain can be chosen to be a greater value than if the bump is not considered. Figure 12 shows the controller’s sensitivity to error in the bump expectation in the kinematic model. The gain in the results is the value of the higher proportional gain in previous figures. Scenario 3, when the vehicle is making a turn, shows similar results to the bump, i.e. the slip ratio becomes smaller if the kinematic model is correct and higher gain can be used.

5.2 Torque control concept

The torque controller will apply the same displacement setting on all motors unless the slip controller reduces the displacement settings. Results for scenario 1, i.e. reduced traction on one wheel, are shown in figure 13. The displacement setting is reduced for the slipping wheel.

In the torque control, the torque, and hence the slip, will be the same for each wheel and independent of any speed changes if slip does not occur. Figure 14 shows the rotational speed and slip ratio when the right
front wheel experiences a bump. The increased torque results in an increased displacement setting and thus increased slip. If the vehicle is turning, the wheel speed is automatically adapted to the kinematics of the vehicle.

![Graph](image1.png)

Figure 14: Scenario 2 with a bump on the right front wheel. The solid lines show result for the right wheel and the dashed lines for the left wheel.

6 DISCUSSION

The torque control does not need any kinematic model of the forwarder in differential operations, i.e. cornering. The wheel speed will automatically be controlled. If one wheel slips, because the traction is reduced, some kind of kinematic model is needed to calculate the slip and reduce it. The simplest model would just assume that all wheels should rotate at the same speed. This corresponds to a locked differential if the gain is large. However, much improved behaviour is obtained if a slightly more advanced model is used. A very simple controller is used in this paper to control the slip. The controller works satisfactorily until the vehicle velocity can not be maintained. The driver then has to react or a more advanced controller has to be used.

The speed control strategy needs a kinematic model to work satisfactorily under differential operation conditions. However, the slip will be controlled automatically. As in the torque control case the control algorithm works satisfactorily until the vehicle velocity is not maintained.

Both controllers need some kind of kinematic description of the vehicle. The speed control concept is probably more dependent on a good model description. The proportional gain in the speed controller is also dependent on the accuracy of the kinematic description. With a more accurate description of the kinematics, the proportional gain can be increased and the slip control will thus be better.

The kinematic model depends on the geometry of the vehicle. Other dependent parameters include the tyre pressure and the weight of the vehicle. The speed of each wheel will also depend on the soil condition. The latter parameters are harder to include in the kinematic description. There are many available measured signals that can be used for the kinematic model. The orientation of the vehicle body can be measured with a gyroscope and angle sensors. The wheel height can be measured with position sensors and angle sensors at the pendulum arm structure. The ground pressure can be measured by pressure transducers for each pendulum cylinder. All these signals can be used to create an accurate state of the kinematic model. A state estimator can be implemented where the signals are used to update the kinematic model online.

Vehicles, especially off-road vehicles, will have wheel slip more or less all the time. The two control proposals in this paper have to be tested in real environments to verify their performance.
7 CONCLUSIONS

In this paper, two control proposals for a six wheel drive forwarder are analysed. The speed controller uses a kinematic model of the forwarder to calculate the speed of each wheel. The torque controller estimates the torque needed and sets the displacement setting equal on each motor. To reduce slip with the torque control concept, an additional controller is needed which reduces the displacement settings when slip occurs. Both controllers work satisfactorily in the tested scenarios: slip, acceleration, bump and turn. The speed controller probably needs a more accurate kinematic model of the forwarder than the torque controller.

Future work includes implementing the two control concepts to verify how the control concepts work in a real environment.

8 ACKNOWLEDGEMENT

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REFERENCES


ANOMALY DETECTION AND DIAGNOSTICS OF A WHEEL LOADER USING DYNAMIC MATHEMATICAL MODEL AND JOINT PROBABILITY DISTRIBUTIONS

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ABSTRACT

In this paper, we present anomaly detection and diagnostics for articulated frame steered hydraulic wheel loader. The presented methodology is based on the analysis and comparison of the responses of a dynamic mathematical model and a real wheel loader using a joint probability distribution of correlation coefficients of multiple variables. The behaviour of an undamaged machine is modelled by probability density functions of the correlation coefficients using histograms and test how well the future behaviour fits the model. First, the time series data of multiple variables are segmented into segments of the same length. Correlation coefficients are then calculated for each segment and the distributions of the correlation coefficients are estimated by computing probability density functions using histograms. Finally, the joint probabilities that the correlations in the data segments of the time series data are observed are calculated using the already computed histograms. The diagnostics is based on the combination of static threshold and threshold based on mean value of joint probabilities. The dynamic mathematical model of the wheel loader is presented with verification results. A jammed flushing valve of the hydrostatic transmission was used as an anomaly to study the changes in the joint probability values. Finally, the efficiency of the presented method is presented with good results regarding detection of anomalies and diagnostics of the wheel loader.

KEYWORDS: Diagnostics, time series, anomaly detection, joint probability, correlation coefficients, simulation, dynamic mathematical model, wheel loader, hydraulics

1. INTRODUCTION

Diagnostics and fault detection of machine systems have been studied widely and significant amount of literature exist of it. See for example the following surveys [1], [2], [3]. Diagnostic techniques can be generally classified into two approaches, depending on whether the diagnostics assessment is based on deterministic or on stochastic information (e.g. historical, statistical parameters) [4], [5]. Segmentation and feature extraction are two major components of time series analysis employed in diagnostics. Segmentation [6], [7] is a method which allows the dividing of time series data into smaller groups of data sets which describe the patterns of the measured variables. Feature extraction involves extracting relevant and discriminating information, and in so doing, reducing data dimensionality: see [8], [9]. The extracted features
are then used to derive the status and the condition of the system using for example classification methods [9], [10], [11].

The use of simulation models in the development of highly automated machines is becoming a necessity [12], [13], [14]. Models are typically created during the early development phase of a machine. However, these simulation models are not effectively utilized in the later phases of product lifecycle. In this paper, previously developed methodology [15], [16], originally developed in simulator environment [14], for using a dynamic simulation model of the machine system for diagnostics purposes is used and the results are verified in real machine environment. In this methodology, the responses of a real undamaged machine and a dynamic mathematical model of this machine are analysed and compared from a stochastic point of view based on probabilities. This means, a statistical model using certain drive sequences of a real undamaged (i.e. healthy) machine and simulated undamaged machine is build and tested how well the future behaviour fits this model.

In the following sections, we present an approach to the anomaly detection and diagnostics of a wheel loader. Section 2 introduces our studied machine, a wheel loader and its dynamic mathematical model. Section 3 describes the methodology to analyse time series data. In section 4 experiments to acquire the data and analysis results are presented, followed by the conclusion in section 5.

2. STUDIED WHEEL LOADER AND DYNAMIC MATHEMETICAL MODEL

In this chapter, the studied wheel loader and the developed dynamic mathematical model are presented with verification results. The wheel loader was engineered at the Department of Intelligent Hydraulics and Automation at Tampere University of Technology [17]. The wheel loader is shown in Fig. 1.

![Figure 1. Studied wheel loader [17].](image)

2.1. Analysed sub-system – hydrostatic drive (HSD)

An overview of the hydraulic systems of the wheel loader, which are related to the analysis performed in this study, is given here. To be precise, the analysed sub-system is hydrostatic drive (HSD). More details about the machine are presented in [17]. Fig. 2 shows a simplified hydraulic circuit of the closed loop HSD of the machine.
The main source of power is a 100 kW four-cylinder diesel engine. The HSD pump has a displacement of 110 cm³/r and contains various integrated hydraulic components, sensors and electronics to implement the closed-loop control of the swivel angle and the data communication.

Both the diesel engine and the pump are connected to the control system of the machine via the CAN bus. They also have separate control units, which are connected to the CAN bus, offering data from integrated sensors. Therefore, via the CAN bus several parameters related to the operation of these components, e.g. rotational speed of the diesel engine and displacement of the HSD pump, can be monitored and recorded for later analysis.

Every wheel of the machine is equipped with a slow speed hydraulic hub motor, with a displacement of 470 cm³/r. The displacement of the motors can also be reduced into 235 cm³/r when the displacement is bisected. Each motor has a pressure controlled holding brake and an integrated sensor for measuring rotational speed. A separate hydraulic gear pump provides the power needed by the steering system. The steering of the machine can be controlled using proportional flow control valve and two symmetrically placed hydraulic cylinders.

The valve connected to the ports A and B of the pump is called a flushing valve. A certain amount of flow is always circulated through this valve to the tank from the lower pressure side of the pump. This reduces the temperature of the fluid and the amount of impurities in it. In this research, this valve has been taken out to simulate a jammed flushing valve of the hydrostatic drive transmission which is used as an anomaly to study the changes in the joint probability values and to verify the results of anomaly detection and diagnostics. This component was selected because jamming of the valve cannot be directly recognized by the operator during normal use of the machine.

### 2.2. Dynamic mathematical model of wheel loader

The diesel engine of the studied machine has a common rail injection system. The most significant part affecting to the engine dynamics is the torque generation in the combustion chamber. This is modelled using a first order system, whose time constant depends on the rotational speed of the engine. The dynamics of fuel injection system is very fast and it is modelled using a 1st order system with a small time constant and a PID-controller. The dynamics of the engine model is verified with separate laboratory measurement data. A detailed description of the engine model is presented in [18] and [19].

The model of the HSD variable displacement pump presents the linear dependency leakage flow and loss torque as a function of port pressures and rotational speed. The equations for the hydraulic pump, and hydraulic fluid volumes are presented in [20] and they are based on [21], [22], [23], [24]. In the simulation model a viscosity of the hydraulic fluid is constant, pressures in the control chambers are distributed evenly.
and the pressure of the oil tank is constant. The dynamics of the displacement of the HSD pump are modelled with a delay and a first order transfer function, which is different for increasing and decreasing displacement. The HSD pump displacement controller includes a rate limiter and a cut off pressure functions, which are also modelled. The HSD pump model is verified based on separate step and ramp response measurements. A constant displacement auxiliary hydraulic pump feeding a hydraulic fluid from the tank to the HSD system through the check valves is included to the hydraulic model.

The leakage model of the HSD hub motors with two different configurations for the displacement is formulated the same way as the HSD pump. Mechanical efficiency of the hub motors is modelled with a bristle type dynamic friction torque model presented in [20], based on [25]. Parameters of the HSD motors are based on the measurement data provided by the manufacturer. The dynamics of the displacement change of all the four HSD motors is verified using the measurement data of the field tests instead of laboratory test bench data.

The flushing valve contains one flow control spool and one pressure relief valve after the spool, see Fig 2. Opening of the spool depends of the pressure difference of the A and B port pressures and a certain pressure difference, in this case 16 bar, is needed to open the valve against to the spring. The pressure – volume flow characteristics of the spool is modelled using look-up table data and dynamics of the spool using a first order transfer function. This parameter data was taken from the manufacturer catalogue. All pressure relief valves including one in the flushing valve are modelled with a semi-empirical model using catalogue data of the manufacturer and is described in more detail in [20].

The mathematical model of the mechanics of the studied machine includes models of the machine body and a tyre-road interaction model; see Fig. 3 for schematics of the mechanical model. The tyre-road model is presented in [18], [20]. It is based on [26] and the parameter data was got from the manufacturer. The body of the machine is assumed to be rigid and Matlab/SimMechanics toolbox is used to calculate 6DOF dynamics of the machine. The boom masses were taken from manufacturer data and the axle weights of the machine were measured. Machine body masses and inertia parameters are estimated based on this data.

![Figure 3. Schematics of machine mechanical model.](image)

### 2.3. Verification of simulation model

The dynamic mathematical model described in the previous section was verified with the data of over 50 different acceleration/deceleration tests. 13 different tests of these 50 were conducted both with functional flushing valve and without it, and four of them were repeated twice for testing the repeatability. These data were utilized in tuning the parameters of the simulation model. In this paper, three different cases for verification are shown.

The reference signals of the actuators (HSD pump and motor displacements, rotational speed of the diesel engine) of HSD were generated with a computer, but the starting time of acceleration and deceleration phase were defined manually by the user. The measurement data was recorded with 1 kHz sampling frequency to the hard drive of the machine and downloaded after the test to a storage drive. The tests were
done at flat asphalt terrain. Afterwards, the recorded references are used as inputs also to the simulation model. The measured variables for the verification were as shown in Table 1.

<table>
<thead>
<tr>
<th>Signal</th>
<th>Range</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel engine rotational speed</td>
<td>0...2200</td>
<td>rpm</td>
</tr>
<tr>
<td>HSD pump displacement</td>
<td>-100...100</td>
<td>%</td>
</tr>
<tr>
<td>Machine velocity</td>
<td>0...20</td>
<td>km/h</td>
</tr>
<tr>
<td>Consumption of diesel engine</td>
<td>0...20</td>
<td>kg/h</td>
</tr>
<tr>
<td>Pressure at port A</td>
<td>0...400</td>
<td>bar</td>
</tr>
<tr>
<td>Pressure at port B</td>
<td>0...400</td>
<td>bar</td>
</tr>
</tbody>
</table>

Figure 4 presents a verification test case 1 in which the machine is accelerated with a predefined 2 s ramp reference change of the HSD pump from 0 to 70% and keeping the diesel rotation speed reference at 1000 rpm. After reaching steady-state the diesel rotation reference was risen stepwise to from 1000 rpm to 1900. Finally, the machine was stopped driving the displacement of the HSD pump to 0 with a 0.7 s ramp. Displacement of the hydraulic motors was held constant (100%) during the test. The temperature of the measured hydraulic fluid was 38 degrees of Celsius in this test case.

In the verification case 2 presented in Fig. 5, the machine was driven with constant diesel engine rotation speed 1300 rpm and rising the displacement of the HSD pump stepwise from 0 to 60%. In the measurement of the pump displacement there can be seen slight overshoot. Because of the first order dynamics of the HSD pump model this phenomenon cannot be seen at the simulation curve. However, this does not give a big error to the operation of the machine, especially because the pump very rarely is controlled with stepwise control signal, and there is no need to use a second order model. At the time 8 s, the hydraulic motors were set to the half displacement and the velocity is increased from about 8 m/s to 15 m/s. In this measurement the temperature of the hydraulic fluid was only 23 degrees of Celsius and because of this the velocity of the machine is about 0.6...0.8 m/s greater than the simulated velocity in the steady state. In all simulations the viscosity of the hydraulic fluid was kept as a constant.

Figure 6 presents a verification test in which the flushing valve is inactive. The machine was accelerated with a predefined 2 s ramp reference change of the HSD pump (from 0 displacement to 50 %) and stopped with a 0.5 s ramp. In this test case, the hydraulic motors were set to half displacement. In this case, the verification results are also satisfactory, as they were also in all other test cases not discussed in this paper. It can be concluded that the model is suitable to be used in anomaly detection and diagnostics study presented in Sections 3 and 4, as well as in control system development presented in [27]. It should be highlighted, that the direct comparison between the measurements with active and inactive flushing valve for fault detection purposes is impossible and some other approach is needed.

Random (short time, high amplitude) error values were detected in some wheel speed measurements, which were done using Hall sensors and a gear ring installed in the wheel axles. Because of this the velocity measurement of the machine was not used at the time series data analysis discussed in Section 4.
Figure 4. Verification case 1. With flushing valve.

Figure 5. Verification case 2. With flushing valve.

Figure 6. Verification case 3. Without flushing valve.
3. ANALYSIS OF TIME SERIES DATA

Analysing and comparing the responses of a real work machine and a corresponding dynamic mathematical model can reveal deteriorated conditions and evolving failures. In this study, a statistical method called a joint probability distribution [16] is utilized for this purpose. In this method, the main idea is to model the behaviour of the system with probability density functions of the correlation coefficients for segmented multivariate time series data using histograms and test how well the future behaviour fits the model. The modelled histograms are then used to derive the joint probability values and furthermore the distributions of these values. Comparison of the joint probability values of multiple variables of individual segments or whole drive sequence enables us to detect effectively different anomalies [16].

When the correlation coefficients of the segmented multivariate data belong to sections of histograms where the probability is very low, then it is treated as rare occasion and the probability of an anomaly is high. Again, if the correlation coefficients belong to sections where the probability is high, it is treated as normal behaviour. On the basis of this information further actions can be targeted on essential subsystems and specific work movements. The main goal is not to try to allocate reasons for possible differences to a specific part of a component or even to a component. The output from the analysis is to verify whether all machine functions correspond to the requirements and to reveal response anomalies and other characteristics in machine operation. The complete procedure for the analysis of time series data will be explained in the following sections.

3.1. Data selection and segmentation

In the analysis, the data are first selected and parsed from the measurements to acquire specifically the data from the acceleration and deceleration phases. This way we can reduce the data dimensionality, synchronize and capture the most relevant data in regards to the analysis [9].

After data selection, the data sets are segmented (i.e. divided) into equal lengths where the segments are overlapping. Segmentation allows the dividing of time series data into smaller groups of data sets which describe the patterns of the measured variables. Segmentation enables allocation of the segments in the time series data that generates anomalies. The length of the segments was chosen long enough to capture transient periods which contain the most relevant data in regards to anomaly detection and diagnostics, but not too long, because otherwise the contribution of the important phenomena in the correlation coefficients decreases.

The measured signals after data selection are denoted $x_i$ for $i = 1, \ldots, m$, where $m$ is the number of signals of interest. We will assume $N$ data segments and let data segment $j$ contain $n$ data samples sampled at time instances $t_1^j, \ldots, t_n^j$ for $j = 1, \ldots, N$. Our earlier assumption on overlapping segments thus requires that $t_1^j < t_1^{j+1} < t_n^j$.

3.2. Estimated distribution of correlation coefficients and observed joint probability distributions

In different sub-systems of a machine, the data signals are mostly dependent to each other and in case of an anomaly these dependencies change in certain way. We will build statistical models of these dependencies and use these models to discriminate the faulty machines by detecting when the correlations deviate from the model of the undamaged machine.

When two sets of data are strongly linked together they have a high correlation. To compare extracted segments of multiple variables, correlations are calculated for each variable pair. The correlation coefficient, denoted by $r$, tells how closely data in a scatterplot fall along a straight line. The closer the absolute value of $r$ is to one, the better the data are described by a linear equation. The value $r = 1$ means a perfect positive correlation, and the value $r = -1$ means a perfect negative correlation. Data with values of $r$ close to zero
show little to no straight-line relationship. Pearson’s correlation coefficient [28] is defined in Eq. 1. For segment $j$, correlation between $x_i$ and $x_k$ is given by

$$r_{i,k}(j) = \frac{\sum_{p=1}^{n}(x_i(t_p^j) - \bar{x}_{i,j})(x_k(t_p^j) - \bar{x}_{k,j})}{\sqrt{\sum_{p=1}^{n}(x_i(t_p^j) - \bar{x}_{i,j})^2 \sum_{p=1}^{n}(x_k(t_p^j) - \bar{x}_{k,j})^2}}$$

(1)

where

$$\bar{x}_{i,j} = \frac{1}{n} \sum_{p=1}^{n} x_i(t_p^j)$$

(2)

is the mean value of segment $j$ of signal $x_i$. Other correlation coefficients and mean values are defined in similar manner.

The behaviour of an undamaged machine is then modelled by probability density functions of these correlation coefficients using histograms and test how well the future behaviour fits the model. In histogram calculation, interval $[-1,1]$ is divided to $M$ bins. Then, to calculate probability distribution $p_{i,k}$, the number of times that $r_{i,k}(j); j = 1, ..., N$ falls in each bin is counted and normalized such that sum of $p_{i,k}$ over all bins equals 1. Logarithmic scale is used to present $p_{i,k}$. Therefore, small positive values are added to zero probabilities before normalization to avoid singularity at zero. Notice that $r_{i,i} = 1$ and that correlation is a symmetric function, that is, $r_{i,k} = r_{k,i}$. Therefore, we will only calculate $p_{i,k}$ for $i, k \in \Pi$, where $\Pi = \{(i,k): i,k = 1, ..., m \text{ and } i < k\}$.

After the the statistical models (i.e. probability distributions of correlation coefficients) are built, those can be evaluated using the measured signals of the test machine. This means, for every extracted segment, correlation coefficients $r_{i,k}$ for $i, k \in \Pi$ are calculated as described in the previous section. Now, probability of outcome $r_{i,k}$ can be evaluated using $p_{i,k}(r_{i,k})$ function, that is, the value of $p_{i,k}$ at the bin where $r_{i,k}$ falls. Further, we will now calculate how probable this segment is using joint probability of all the correlation coefficients given by

$$P = \sum_{(i,k)\in\Pi} p_{i,k}(r_{i,k})$$

(3)

When most of the correlation coefficients are highly probable, value of $P$ is large, and probability of an anomaly is really low (i.e. anomaly is not detected). Otherwise, when large portion of correlation coefficients have low probabilities, $P$ becomes small and anomaly is detected. Threshold values are used for $P$ under which anomalies are detected.

4. EXPERIMENTS AND ANALYSIS RESULTS

Both, the wheel loader and its dynamic mathematical model (i.e. simulation model) were used to generating the analysed data. A jammed flush valve of the hydrostatic transmission was used as a fault case (an anomaly). To a healthy machine we refer as undamaged and to a machine with the fault as damaged. We used the similar type of control inputs to drive the real undamaged and damaged machines, and the dynamic mathematical model to generate comparable time sequences both for statistical modelling and tests.

First, we drove the undamaged machine according the defined test case. The control signals were recorded to be used for the mathematical model. The used test drive case was the following: acceleration (driving straight) – deceleration – stopping. The reference control signals for acceleration and deceleration are predefined for each experiment but starting and stopping the test drive is done manually. All the test drives were carried out without any additional load. We measured 17 sets of data in case undamaged and
simulated machine and seven sets of data in case of damaged machine as shown in Table 2. So altogether 41 test drives were completed. Undamaged and damaged data sets are actually part of the verification measurements which were chosen for analysis: see verification measurements in Section 2.3.

### Table 2. Number of measured test drives and their use in analysis.

<table>
<thead>
<tr>
<th>Machine</th>
<th>Training</th>
<th>Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Undamaged</td>
<td>10</td>
<td>7</td>
</tr>
<tr>
<td>Simulated</td>
<td>10</td>
<td>7</td>
</tr>
<tr>
<td>Damaged</td>
<td>-</td>
<td>7</td>
</tr>
</tbody>
</table>

Four variables were measured in each experiment as shown in Table 3. The sampling frequency was 100 Hz (resampled from verification measurements). 20 test drives from these 41 data sets were used in the training phase (statistical model generation) and 21 in the testing phase of the analysis: see table 2. Although, the driven test drives are quite similar, there are still differences and it was the purpose of our research to show that this methodology works even when the tested data sets are driven with different but still similar type of control inputs than used in the statistical model generation.

### Table 3. Analysed variables of HSD.

<table>
<thead>
<tr>
<th>Signal</th>
<th>Range</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$s_1$ Diesel engine rotational speed</td>
<td>0…2200</td>
<td>rpm</td>
</tr>
<tr>
<td>$s_2$ HSD pump displacement</td>
<td>-100…100</td>
<td>%</td>
</tr>
<tr>
<td>$s_3$ Pressure at port A</td>
<td>0…400</td>
<td>bar</td>
</tr>
<tr>
<td>$s_4$ Pressure at port B</td>
<td>0…400</td>
<td>bar</td>
</tr>
</tbody>
</table>

Data selection is based on the HSD pump displacement signal. The analysed signals are combined of three parts: 1) $a = 400$ data points after the displacement starts to rise (i.e. is higher than zero), 2) $b = 200$ data points before the displacement decreases to zero and 3) $c = 200$ data points after the displacement reaches zero displacement at the end of the test drive: see Fig. 7. This same procedure is done to all the analysed signals using the information (i.e. specific points a,b and c in time series data) of data selection acquired from the corresponding displacement signal.

![Figure 7. Data selection based on HSD pump displacement data: a) raw HSD pump displacement data, b) example of data selection.](image)

Measured variables from all the test drives used in the analysis in the case of undamaged, simulated and damaged machines are shown in Fig. 8. The data shown in Fig. 8 is already selected according the procedure described previously.
Figure 8. Analysed data in case of real undamaged, simulated and real damaged machines: a) diesel engine rotational speed, b) HSD pump displacement, c) pressure A and d) pressure B.

After data selection, each data set contained 800 data points for each measured variable. The measurements were segmented into parts of the same length. The length of the segment was $n = 100$ with 50 overlapping data points. Thus the number of segments equal $N = 15$ for each data set.

In modelling of probability density functions, every extracted segment included $m = 8$ variables, that is, $x_i; i = 1, \ldots, 8$, where $x_i; i = 1, \ldots, 4$ are signals $s_j; i = 1, \ldots, 4$ (defined in Table 3) of the undamaged machine, and $x_i; i = 5, \ldots, 8$ are signals $s_j; i = 1, \ldots, 4$ of the mathematical model. This means altogether 28 correlation coefficients. Probability density functions for these correlation coefficients were computed using histograms. The histogram, interval [-1, 1] was divided into $M = 21$ bins.

In testing phase, the correlation coefficients $r_{i,k}$ for $i, k \in \Pi$ are calculated in the same way as in the training using the undamaged or the damaged machine ($x_i; i = 1, \ldots, 4$) against the mathematical model. The mutual correlations of the simulated machine ($r_{1,2}, r_{1,3}, r_{1,4}, r_{2,3}, r_{2,4}, r_{3,4}$) were omitted. Based on earlier research, those correlation coefficients do not contain information which would help in anomaly detection [16]. Therefore, we used only the remaining 22 correlation coefficients.

After that, we calculate $P$ for each segment to obtain $P_n$ which is the sequence of the joint probabilities $P$ for $n = 1, \ldots, N$. Fig. 9 shows joint probability distributions of training (statistical model generation) and testing
with calculated mean values of the distributions and defined static threshold for anomaly detection. The differences between the undamaged and the damaged machines can be clearly seen.

The diagnostics is finally based on thresholds. Using a static threshold for every segment, which is a little smaller than the smallest $P$ in the case of the undamaged machine, we can detect the presented fault. So if $P$ in a segment is smaller than the threshold, then this segment is treated as anomaly. This enables also the combining of the anomalies with the operating states of the machine. From Fig. 9 can be seen that there are segments which have very low $P$ value. These segments are very rare or they do not appear at all in the training phase. It indicates a high probability of an anomaly. When the static threshold is defined to be smaller than the smallest value of the statistical model, here -70, we can detect anomalies that have smaller $P$. Because the data sets in the testing were slightly different than in the model generation, it can also be noticed that some of the segments in the testing in the case of the undamaged machine gets a $P$ which is lower than the defined threshold. Therefore, the threshold value could be lowered a little to get rid of this phenomenon. Table 4 shows the number of segments under the static threshold using the value -70.

<table>
<thead>
<tr>
<th>Undamaged train</th>
<th>Undamaged test</th>
<th>Damaged test</th>
</tr>
</thead>
<tbody>
<tr>
<td>0/111 (0 %)</td>
<td>5/111 (4.5 %)</td>
<td>14/111 (12.6 %)</td>
</tr>
</tbody>
</table>

In addition to the static threshold, a threshold based on the arithmetic mean of $P_n$ from one test drive or longer period containing several test drives can be used. In this way, both the single segments with low probability values, which indicate high probability of an anomaly, and also changing trends of the system can be detected. The mean value of the damaged machine is clearly lower than in the case of the undamaged machine, see Fig. 9. In testing, the difference between the mean value of the damaged machine (red) and the mean value of the statistical model of the undamaged machine (blue) is over five times higher (529%) compared to the undamaged machine (green).

5. CONCLUSIONS

Anomaly detection and diagnostics of a wheel loader was studied using a real machine (wheel loader), a dynamic mathematical model of this machine and joint probability distributions. The dynamic mathematical model of the wheel loader and its sub-components were verified with several different laboratory and field measurements. The verification results of three different cases were presented in the paper. Based on the
verification results can be concluded that the model is suitable to be used in anomaly detection and diagnostics study.  

The HSD of the wheel loader was selected as analysed sub-system of the machine. A jammed flush valve of the hydrostatic transmission was used as an anomaly to study the changes in the joint probability values and to verify the results of the anomaly detection and the diagnostics. Altogether, 41 test drives, 24 with the real machine and 17 with the dynamic mathematical model, were carried out to obtain measurement data for analysis purposes. The measured data comprised of four variables describing the behaviour of the HSD.  

Experimental results of anomaly detection and diagnostics were presented using a combination of a static threshold and a threshold based on the arithmetic mean of the joint probability distribution. This enables detection of both single segments with low probability values indicating anomalies and also changing trends of the system. The central point of the joint probability distribution method is that the probabilities of the multiple correlation coefficients of the variable pairs are compared, instead of comparing the responses or even the calculated correlations directly. Combining these probabilities enables the detection of anomalies, rare situations with low probabilities, from which one can conclude there is something wrong in the system or the subsystem. Joint probability distributions were calculated for the test drives using the computed histograms of the correlation coefficients. The experimental results show clearly lower probabilities for test drives where fault is present compared to ones without faults.  

The analysing methodology which is used in this study enables the detection of sudden critical faults as well as slowly evolving faults. The simultaneous examination of several variables enables also a more generic approach of detecting several different anomalies and applying it to different machine types.  

ACKNOWLEDGEMENT  

The support of the Finnish Funding Agency for Technology and Innovation (TEKES) in the research project Computational Methods in Mechanical Engineering Product Development (SIMPRO) is gratefully acknowledged (Grant No. 40204/12).  

NOMENCLATURE

<table>
<thead>
<tr>
<th>Designation</th>
<th>Denotation</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>cons meas</td>
<td>Measured fuel consumption</td>
<td>[kg/h]</td>
</tr>
<tr>
<td>cons sim</td>
<td>Simulated fuel consumption</td>
<td>[kg/h]</td>
</tr>
<tr>
<td>( \bar{e} )</td>
<td>Arithmetic mean of error</td>
<td>[-]</td>
</tr>
<tr>
<td>em meas</td>
<td>Measured displacement of hydraulic motor</td>
<td>[%]</td>
</tr>
<tr>
<td>em sim</td>
<td>Simulated displacement of hydraulic motor</td>
<td>[%]</td>
</tr>
<tr>
<td>ep meas</td>
<td>Measured displacement of pump</td>
<td>[%]</td>
</tr>
<tr>
<td>ep sim</td>
<td>Simulated displacement of pump</td>
<td>[%]</td>
</tr>
<tr>
<td>k</td>
<td>Number of variables</td>
<td>[-]</td>
</tr>
<tr>
<td>M</td>
<td>Number of bins</td>
<td>[-]</td>
</tr>
<tr>
<td>N</td>
<td>Number of segments</td>
<td>[-]</td>
</tr>
<tr>
<td>m</td>
<td>Number of data signals</td>
<td>[-]</td>
</tr>
<tr>
<td>n</td>
<td>Number of data points</td>
<td>[-]</td>
</tr>
<tr>
<td>( \eta ) meas</td>
<td>Measured rotational speed of diesel engine</td>
<td>[rpm]</td>
</tr>
<tr>
<td>( \eta ) sim</td>
<td>Simulated rotational speed of diesel engine</td>
<td>[rpm]</td>
</tr>
<tr>
<td>P</td>
<td>Joint probability value</td>
<td>[-]</td>
</tr>
<tr>
<td>( p_A ) meas</td>
<td>Measured pressure A</td>
<td>[bar]</td>
</tr>
<tr>
<td>( p_B ) meas</td>
<td>Measured pressure B</td>
<td>[bar]</td>
</tr>
</tbody>
</table>
Simulated pressure A [bar]
Simulated pressure B [bar]
Probability of \( r_{i,k} \) [-]
Minimum joint probability value [-]
Joint probability distribution [-]
Mean of joint probability [-]
Correlation coefficient [-]
Correlation between \( x_i \) and \( x_k \) for segment \( j \) [-]
Variance [-]
Analysed signals [-]
Time instances on which \( n \) data samples in data segment \( j \) are sampled [-]
Measured velocity of machine [km/h]
Simulated velocity of machine [km/h]
Data vector [-]
Mean value of segment \( j \) of signal \( x_i \) [-]
Data vector [-]
Mean value of segment \( j \) of signal \( x_k \) [-]
Accumulated error of \( P_n \) [-]

REFERENCES


ABSTRACT

The four-equation model consists of two equations for the fluid and the other two for the pipe, describing the axial vibration of liquid-pipes. Fluid-structure interaction (FSI) is significantly considered in this model, typically solved with the method of characteristics (MOC) which introduces the numerical error inevitably. When friction and damping effects are neglected, the model can be solved with an exact solution without interpolation errors. The only weakness of this solution is high time cost due to its recursion approach. An improved method based on this exact solution is proposed in this paper, using the time interpolation rather than the recursive method, to speed the calculation. The present solution improves the efficiency dramatically while retains the level of accuracy.

KEYWORDS: Fluid-structure interaction, exact solution, liquid-pipe vibration

1. INTRODUCTION

The fluid-structure interaction (FSI) effect in liquid-filled pipes is significantly considered in the areas of hydraulic systems, nuclear industry, and water supply systems. The FSI model contains fluid equations (extended from classic water hammer equations) and structure equations (based on the beam theory). Transients (also known as water hammer) considering the influence of the pipe wall was studied by Skalak [1],[2] and Thorley [3], and the FSI four-equation model was proposed. This model was developed for describing the axial vibration of straight pipes, and its validity was fully demonstrated [5]-[8].

The Method of Characteristics (MOC) is an approach to solve linear hyperbolic partial differential equations (PDEs), such as beam structure [9]. The principle is to transform PDEs into ordinary differential equations (ODEs) along characteristic lines. The FSI model of liquid-filled pipes is a hyperbolic PDE system and can be solved with MOC. The computation process is to mesh the distance-time plane, and time march from initial conditions. The MOC has been proved to be the most popular and efficient time-domain approach for fluid transients and FSI problems by many researchers [6],[10]-[13]. An investigation showed that most commercially available water hammer software packages are based on the MOC [14]. To retain the stability of MOC calculation, the time step $\Delta t$, space interval $\Delta z$ and wave speed $c$ should satisfy the Courant condition (Courant number $C_r = c \Delta t / \Delta z \leq 1$), although it is not always easy in practice. Interpolation technique [15] and wave speed adjustment [16],[17] are employed to address this problem, however, bringing numerical damping and phase error.
Some researchers [4],[10],[18] had made efforts on the exact time-domain solution of FSI, but these existing methods had limitations [19]. An exact solution without any interpolations and adjustments was proposed by Tijsseling [19]-[21], which can be implemented by means of a simple recursion. The wave front was tracked backwards to the initial conditions in time, and no computational grid was needed. The only weakness of this method was the exponential calculation time when the event’s duration increased.

The method presented in this paper is based on Tijsseling’s exact solution, and the primary improvement is using time-line interpolation instead of recursions, which will significantly speed the calculation while retain the accuracy level. The benchmark problem of water hammer is studied to validate the effectiveness of the new approach.

2. METHODOLOGY

The principle of the exact solution and its improvement developed by authors are presented in this section. The analytical derivation follows Ref.[8] and [19].

2.1. Mathematic model

The four equations describing the axial vibration of a liquid-pipe section are

\[
\frac{\partial V}{\partial t} + \frac{1}{\rho_f} \frac{\partial P}{\partial z} = 0, \quad (1)
\]

\[
\frac{\partial V}{\partial z} + \frac{1}{K^*} \frac{\partial P}{\partial t} - 2\nu \frac{\partial \hat{u}_z}{\partial z} + \frac{1}{K} = \frac{1}{K} + (1 - \nu^2) \frac{2r}{Ee}, \quad (2)
\]

\[
\frac{\partial f_z}{\partial z} - A_p \rho_p \frac{\partial \hat{u}_z}{\partial t} = 0, \quad (3)
\]

\[
\frac{\partial f_z}{\partial t} - A_p \nu \frac{\partial P}{e \partial t} - EA_p \frac{\partial \hat{u}_z}{\partial z} = 0, \quad (4)
\]

where \( V \) is fluid velocity, \( P \) is fluid pressure, \( \hat{u}_z \) is axial pipe velocity, and \( f_z \) is axial pipe force. This model is identical to the one in Ref.[19], and valid for thin-walled, linearly elastic, circular cross-section, straight pipes filled with inviscid liquid.

The PDEs can be written as

\[
A(\partial/\partial t)\phi(z,t) + B(\partial/\partial z)\phi(z,t) + C\phi(z,t) = 0 \quad (5)
\]

where \( \phi \) denotes the vector of system variables, \( A \), \( B \) and \( C \) are matrices of constant coefficients. Herein, \( C = O \) and \( \phi(z,t) = \left( V \quad P \quad \hat{u}_z \quad f_z \right)^T \).

A new set of dependent variables is introduced through

\[
\eta(z,t) = S^{-1}\phi(z,t) \text{ or } \phi(z,t) = S \eta(z,t), \quad (6)
\]

where \( S \) consists of the eigenvectors of \( A^\dagger B \). And Eq.(5) can be decoupled as

\[
\frac{d\eta_i(z,t)}{dt} = 0, \quad i = 1, 2, 3, 4 \quad (7)
\]

along characteristic lines.
\[ \frac{dz}{dt} = \lambda_i, \quad i = 1, 2, 3, 4 \]  

in the \( z - t \) plane, herein \( \lambda_i \) is the eigenvalue of \( A^{-1}B \).

2.2. Exact solution

The conventional solution (MOC) transforms Eq.(7) into difference equation

\[ \eta_i(z, t) = \eta_i(z - \lambda_i \Delta t, t - \Delta t), \quad i = 1, 2, 3, 4, \]  

and meshes the distance-time plane with computational grid.

Tijsseling [19][20] found that the variable \( \eta_i \) would not change along relevant characteristic line, therefore it can be tracked back to its initial value or boundary value (see Fig.2, Ref.[19]). The initial vector value can be obtained with the initial condition

\[ \eta(z, 0) = S^{-1}\phi(z, 0). \]  

The vector \( \eta(0, t) \) in grid point P on the left boundary (see Fig.3, Ref.[19]) can be obtained from boundary equations (Eq.(19), Ref.[19]) partly \( \eta_1, \eta_3 \), and from element \( \eta_2 \) of \( \eta(L, t + L / \lambda_2) \) and \( \eta_4 \) of \( \eta(L, t + L / \lambda_4) \) on the right boundary. And the vector \( \eta(L, t) \) on the right boundary can be obtained in the analogous process. So the characteristic line seems reflect backwards every time at the boundary, until reaching the initial conditions.

Consequently, the dependent vector \( \eta \) at any position can be calculated from the invariants \( \eta_i \) from initial conditions and boundary conditions, without any interpolations. But the approach has to track back to the initial conditions through a reflecting path in every calculation step, which results in the exponential computational time.

2.3. Improved method

The basic idea of the improved method presented here is using time-line interpolation rather than the recursion approach. This means \( \eta \) can be obtained by interpolating invariant values in boundary points, rather than reflecting characteristic lines backwards in time.

In Fig.1, the dots along the time-line refer to calculation points. The vector \( \eta \) in interior point P1 can get invariant elements from point \( A_i (i = 1, 2, 3, 4) \) which can be interpolated with invariants in adjacent points.
As for the point on the boundary (P2 in Fig.1), two invariant elements are calculated from boundary equations and the other two are from initial condition $A_2$ and $A_4$. The approach is shown in Fig.2, where point P0 reflects the time $L^t \eta_i$ on the left boundary. If the time is before P0, such as P1 in Fig.2, the element $\eta_4$ is fed by $A_4^i$ which has an initial value. And when the time after P0, such as P2, $\eta_4$ can be obtained from $A_4^{i+1}$ which has an interpolated value on the time-line of right boundary. So there are an exact zone and interpolated zone along the time-line. It is noted that the density computational points are essential in this approach, which means values in all points on the boundary need to be stored.

The process of new approach is explained in words here (also see the Appendix). At time $t$, the vector $\eta(0, t)$ and $\eta(L, t)$ will be calculated firstly and then be stored. The vector $\eta$ in interior point will be calculated from $A_i(i = 1, 2, 3, 4)$ which is the interpolation of boundary vectors in adjacent time-line nodes.

Using time-line interpolation, reflections of characteristic lines can be avoided, which means that there is no need to track back to the initial conditions at every calculation step. Therefore, the time-cost can be reduced dramatically. But as an interpolation method, the calculation result might not be as accurate as that of the exact solution. One can get a higher level of accuracy by means of increasing point density on the time-line on boundaries.
3. VALIDATION

The present method and previous exact solution [19] are compared in a simulation example. The Delft Hydraulics Benchmark Problems A, describing a reservoir-pipe-valve (RPV) system, is employed in this section to validate the efficiency and accuracy of the improved method. This case is just a numerical test only, for experiments have not been carried out.

![Reservoir-pipe-valve system of Benchmark Problem A](image)

The parameters of this system are defined in Table 1. The initial fluid velocity is 1 m/s and pressure is 0 Pa. The duration of valve closure is instantaneous, and the valve can be either free or structurally fixed. As the valve closes instantaneously, the water hammer will be induced significantly, and FSI will be considered here.

<table>
<thead>
<tr>
<th>Pipe (Steel)</th>
<th>Liquid (Water)</th>
<th>Geometry</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density 7900 kg m(^{-3})</td>
<td>Density 1000 kg m(^{-3})</td>
<td>Pipe length 20 m</td>
</tr>
<tr>
<td>Young’s modulus 210 GPa</td>
<td>Bulk modulus 2.1 GPa</td>
<td></td>
</tr>
<tr>
<td>Shear modulus 80.77 GPa</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Poisson’s ratio 0.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Radius 398.5 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pipe wall thickness 8 mm</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

If the valve is unrestrained and can move freely, its axial vibration will demonstrate a strong FSI process. The boundary condition at valve is \( \dot{V} = \dot{u}_z \) and \( f_z = A_f P \). Simulation results are depicted in Fig.4, containing the exact solution (solid line) and present method (dot line). Fluid pressure at the position of valve is calculated. The exact solution is calculated with 500 points of the whole computational duration, and the new method uses 2000 points. As the magnified detail shown in Fig.4, the present method can reach a very high level of accuracy compared with the exact solution. Importantly, the calculation time is reduced from 44.9 sec to 0.33 sec dramatically, with 2.5 GHz two-core CPU and 4GB memory.

![Pressure at valve for Benchmark Problem A with free valve: exact solution (solid) and present method (dot)](image)
If the valve is fixed, the FSI mechanism is Poisson coupling [22], which is not as significant as the FSI in the case of free valve. The boundary condition at valve would be $\dot{u}_z = 0$ for the fixed constraint, and $V = \ddot{u}_z$ for the closed end. The calculation results are shown in Fig.5, and indicate extremely good agreement between exact solution and present method, using 500 and 2000 calculation points respectively as well. The time-cost is reduced from 44.3 sec to 0.38 sec.

The calculation time-cost mentioned in this section is just for the applied computer, and it will change when more tasks run simultaneously or on different computers. The status of the applied computer’s system is consistent for calculations in this section, accordingly the purpose of indicating the effectiveness of the proposed method is achieved.

It can also be found that the oscillated pressure during water hammer process is lower when valve is restrained. And the wave shape resembles the classic water hammer more than that in the case of free valve. As Eq.(7) can only be obtained when $C = O$, the distributed friction or damping are not contained in either exact solution or present method. But one can use lumped friction concentrated on boundaries [20].

4. CONCLUSION

The exact solution for 4-equation model of FSI axial vibration is improved here, by means of using time-line interpolation technique. There is no need to mesh the distance-time plane or make wave front track back to the initial conditions. The improved method is tested in a benchmark problem of water hammer in RPV system, indicating a high level of accuracy and a big progress on the efficiency. The calculation speed is proved to be raised dramatically. The primary contribution of this work is the improvement on the calculation efficiency while achieving the high accuracy.

As the interpolation on time-line is absorbed, time-dependant boundary can also be contained, which means complex boundary conditions (elastic supports, various excitations) can be described. The future work includes applications on coupled pipes, as well as adding friction and complex boundary conditions.

ACKNOWLEDGEMENT

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NOMENCLATURE

Uppercase Letters

\( A \) cross-sectional area
\( E \) Young’s modulus
\( G \) shear modulus
\( K \) fluid bulk modulus
\( L \) length of pipe
\( P \) fluid pressure
\( V \) fluid velocity

Lowercase Letters

\( c \) wave speed
\( e \) thickness of pipe wall
\( f \) force in cross-section
\( r \) radii of pipe cross-section
\( u \) pipe displacement
\( z \) distance along the pipe
\( \rho \) density
\( \nu \) Poisson’s ratio

Subscripts

\( f \) fluid
\( p \) pipe
\( z \) axial coordinate

Superscripts

\( T \) transposed

Matrices and Vectors

\( A, B, C \) coefficient matrix of FSI model
\( D \) boundary matrix
\( Q \) excitation vector
\( \phi \) state vector of 14 variables

REFERENCES

APPENDIX

The input of the new approach is: the eigenvalue \( \lambda_i \) (i = 1, 2, 3, 4) and eigenmatrix \( \mathbf{S} \), the boundary matrix \( \mathbf{D} \), the excitation vector \( \mathbf{Q} \), and the initial value \( \eta_0 \). The output is the state vector \( \phi = \mathbf{S} \eta \).

1) Subroutine INTERP (input: \( z, t \); output: \( \eta \)) calculates the interpolated value of two stored \( \eta \) in adjacent points along the time-line on boundaries.
2) Subroutine BOUNDARY calculates $\eta$ in the boundary points. The process of the method is written in pseudo-code as below.

BOUNDARY (input: $t$; output: $\eta$)

if ($t \leq 0$) then
  $\eta = \eta_0$
else
  % when $z = 0$
  if ($t - L/\text{abs}(\lambda_2) < 0$) then
    CALL BOUNDARY ($L$, $t - L/\text{abs}(\lambda_2)$; $\eta$)
    $\eta_2 = \eta_2$
  else
    INTERP ($L$, $t - L/\text{abs}(\lambda_2)$; $\eta$)
    $\eta_2 = \eta_2$
  end
  if ($t - L/\text{abs}(\lambda_4) < 0$) then
    CALL BOUNDARY ($L$, $t - L/\text{abs}(\lambda_4)$; $\eta$)
    $\eta_4 = \eta_4$
  else
    INTERP ($L$, $t - L/\text{abs}(\lambda_4)$; $\eta$)
    $\eta_4 = \eta_4$
  end
  $\eta_1 = \alpha_{11}q_1 + \alpha_{12}q_2 - \beta_{11}\eta_2 - \beta_{12}\eta_4$
  $\eta_2 = \eta_2$
  $\eta_3 = \alpha_{21}q_1 + \alpha_{22}q_2 - \beta_{21}\eta_2 - \beta_{22}\eta_4$
  $\eta_4 = \eta_4$

(Herein, $\alpha$ is the coefficient of $\begin{bmatrix} \text{DS}_{1,1} & \text{DS}_{1,3} \\ \text{DS}_{2,1} & \text{DS}_{2,3} \end{bmatrix}^{-1}$, $\beta$ is the coefficient of $\begin{bmatrix} \text{DS}_{1,2} & \text{DS}_{1,4} \\ \text{DS}_{2,2} & \text{DS}_{2,4} \end{bmatrix}$, and $\text{DS}$ is the production of matrix $D$ and $S$.)

% when $z = L$

if ($t - L/\text{abs}(\lambda_4) < 0$) then
  CALL BOUNDARY ($0$, $t - L/\text{abs}(\lambda_4)$; $\eta$)
  $\eta_1 = \eta_1$
else
  INTERP ($0$, $t - L/\text{abs}(\lambda_4)$; $\eta$)
  $\eta_1 = \eta_1$
end
if \((t - L / \text{abs}(\lambda_3) < 0)\) then

\[
\text{CALL BOUNDARY (0, } t - L / \text{abs}(\lambda_3); \eta) \]
\[
\eta^3 = \eta_3
\]
else

\[
\text{INTERP (0, } t - L / \text{abs}(\lambda_3); \eta) \]
\[
\eta^3 = \eta_3
\]
end

\[
\eta_1 = \eta^1
\]
\[
\eta_2 = \alpha_1 q_3 + \alpha_{12} q_4 - \beta_1 \eta^1 - \beta_1 \eta^3
\]
\[
\eta_3 = \eta^3
\]
\[
\eta_4 = \alpha_2 q_3 + \alpha_{22} q_4 - \beta_2 \eta^1 - \beta_2 \eta^3
\]

(Herein, \(\alpha\) is the coefficient of \[
\begin{pmatrix}
\text{DS}_{3,2} & \text{DS}_{3,4} \\
\text{DS}_{4,2} & \text{DS}_{4,4}
\end{pmatrix}^{-1}
\]
, \(\beta\) is the coefficient of \[
\begin{pmatrix}
\text{DS}_{3,2} & \text{DS}_{3,4} \\
\text{DS}_{4,2} & \text{DS}_{4,4}
\end{pmatrix}^{-1}
\begin{pmatrix}
\text{DS}_{3,3} & \text{DS}_{3,3} \\
\text{DS}_{4,3} & \text{DS}_{4,3}
\end{pmatrix}
\])

End

3) Subroutine INTERIOR (input: \(z, t\); output: \(\eta\)) calculates \(\eta\) in the interior points, which is the same as that in Ref.[19].
INFLUENCE OF CLEARANCE ON THE SUPERFLUOUS TORQUE IN THE ELECTRO-HYDRAULIC DYNAMIC LOADING SYSTEM

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ABSTRACT:

The superfluous torque existed in the electro-hydraulic dynamic loading system would make the torque loading precision down and reduce the bandwidth of the loading system, the engineering practical experience shows that: the clearance in the mechanical connections of loading system transmission chain makes superfluous torque more complex and difficult to compensate, and torque loading precision even worse. Aiming at these problems, with the method of power bond graph, the electro-hydraulic dynamic loading system mathematical model including clearance nonlinearity is established, which abstracts from the dynamic loading process of semi physical simulation experiment in a large scale wind turbine's variable pitch-controlled system. When the value of the clearances are 0, 0.1, 0.2, 0.4, 0.8 and 0.9 mm respectively, the results of simulation with 20-sim software show that: when the position system (the bearing object of electro-hydraulic load simulator) is in starting and reversing, the clearance has the strengthening effect on superfluous torque; with the value of clearance increasing from 0 mm to 0.8 mm, the superfluous torque peak rises from 3.7 Nm to 164.2 Nm, and the torque loading precision changes from 1.85% to 82.1%; when the clearance reaches up to 0.9 mm, the phenomenon of limit cycle oscillations would occur in the loading system, and the system cannot work steadily at present.

KEYWORDS: Electro-hydraulic dynamic loading, clearance, superfluous torque, loading precision
1. INTRODUCTION

Electro-hydraulic dynamic loading system is the key equipment used for simulating the variety of dynamic loads on the bearing object in the semi physical simulation, which is also known as the electro-hydraulic load simulator. The bearing object is generally a position system, such as fin stabilizers, aircraft rudder, wind turbine’s variable pitch-controlled system and so on. In the process of dynamic loading, the loading system would produce superfluous force (torque) for the strong disturbance of the position system. Many scholars did a lot of work on this and obtained the experimental or theoretical achievements on certain, Jingfu Wang effectively restrained the superfluous torque with the P-Q servo valve[1], Biao Zhang proposed the way of combining the adjustable throttle and back-stepping adaptive control to suppress the superfluous torque[2], Iketani Hikarue restrained the superfluous force with the traditional PID compensation controller and the model reference adaptive compensation controller[3], etc.

An electro-hydraulic dynamic loading system’s actuator is normally a valve-controlled cylinder or motor, and the clearance inevitably exists in the mechanical joints between the actuator of loading system and the bearing object. Liejiang Wei studied the influence of clearance on simple pendulum load electro-hydraulic servo system and proposed the full compensation of clearance in electro-hydraulic position servo system by means of double-cylinder linkage [4, 5]. In essence, the clearance is a kind of nonlinear characteristic, which makes superfluous torque in the electro-hydraulic dynamic loading system more complex and difficult to compensate, torque loading precision even worse, system bandwidth reduced and loading control strategy too complex to realize. Therefore, it is very important for superfluous torque compensation and accurate loading of the electro-hydraulic dynamic loading system to study on the effects of the clearance on the superfluous torque.

In this study, the mathematical model of electro-hydraulic dynamic loading system including clearance nonlinearity is established, which abstracts from the dynamic loading process of semi physical simulation experiment in a large scale wind turbine’s variable pitch-controlled system, and this paper mainly aims at the effects of the clearance on the superfluous torque when the position system is in starting and reversing.

2. ELECTRO-HYDRAULIC DYNAMIC LOADING PROCESS

The large scale wind turbine’s variable pitch-controlled system is the mechanism to change the wind energy absorption efficiency of blades for adapting the change of wind speed, which overcomes all kinds of resistance on the blades to control the blades to a specified angle. It costs too much to test the variable pitch control strategy on the real wind turbine because of the complexity of the loads of wind turbine’s blades, so the loads should be simulated under laboratory conditions to study the impact on the performance of the variable pitch-controlled system, and to provide the basis for high-performance variable pitch control strategy.
Schematic diagram of the dynamic loading process of the large scale wind turbine's variable pitch-controlled system shown in Fig 1, consists of the variable pitch-controlled system and the torque loading system. For the variable pitch-controlled system, servo valve 1 controls the cylinder 2 as the actuator to drive the slider-crank mechanism, a feedback closed loop system is formed with angular displacement sensor 3, which is the real part on the wind turbine used to control the pitch angle. While for the torque loading system, servo valve 9 controls the rotary actuator 8 to load for the variable pitch-controlled system, a feedback closed loop system is formed with torque sensor 7, which is the part used to simulate the loads on the real blades under laboratory conditions. The torque loading system would produce superfluous torque for the strong position disturbance of the variable pitch-controlled system. There are many clearances in the transmission chain of the loading system: the connection between coupling 6, 8 and shaft 5, the connection between wind turbine blade 4 and shaft 5, etc. For convenience, just assume the following: there is no clearance existed in the transmission chain of the loading system except the clearance between wind turbine blade 4 and shaft 5.

2  MODELING AND ANALYSIS

2.1  The Bond Graph Model of Clearance

Generally, the connection between wind turbine blades and shaft is rectangle spline, hub of the blade is internal spline, the shaft is external spline, and the mechanical clearances \( b \) between the internal spline and external spline as shown in Fig 2.
When the spline tooth is moving in the clearance $b$, without contacting to internal spline, there is no energy and torque being transferred, the internal spline and external spline move in their own way, so the ideal converter in the bond graph theory can be used to model such clearance [6], which exerts 0 effort ($S_w$) to the 0 junction now. Once the spline tooth passes through the clearance $b$ and contacts to the internal spline, the ideal converter exerts the actual torque ($S_w$) to the 0 junction. The bond graph model of clearance in rectangle spline is shown in Fig 3, where $J_2$ and $J_3$ are the rotational inertia of hub and spline shaft respectively; $F$ and $v$ are the output force and velocity of cylinder, $T$ and $\dot{\theta}$ are the loading torque and speed of rotary actuator.

\[ \begin{align*}
F & \xrightarrow{J_2} 0 \\
v & \xrightarrow{J_3} T \\
S_w & \xrightarrow{\dot{\theta}}
\end{align*} \]

Fig 3. The bond graph model of clearance in rectangle spline

2.2 The bond graph model of load system

Based on the theory of bond graph, constant effort source $P_s$ stands for source of oil and constant effort source $P_t$ stands for return oil in the hydraulic system. The dynamic characteristics of servo valve are described with second-order oscillation module, flow characteristics of servo valve’s orifice are described with element $R_1$, $R_2$, $R_3$ and $R_4$ in the theory of bond graph, input is the input current of the servo valve, $A$ and $B$ are the high and low pressure oil conveyed to the actuator, the bond graph model of servo valve is shown in Fig 4.
Fig 4. The bond graph model of servo valve

At every moment, in the condition of ignoring the mass and friction of each component of the slider-crank mechanism, there will be $F \times v = T \times \dot{\theta}$, this relationship can be described by the adjustable 2-ports transformer MTF. With all the bond graph models of nonlinear factors considered, the structure invariance principle and the compensatory way of flow applied, the signal of angular speed feedback played a role in loading servo valve, the bond graph model of electro-hydraulic dynamic loading system including clearance nonlinearity is established with the 20-sim software, as shown in Fig 5, where $R_1$ and $R_2$ are the inner leakage of the hydraulic cylinder and the rotary actuator, $R_p$ stands for the frictional force of hydraulic cylinder piston and $R_f$ is the frictional torque of the rotary actuator, $m$ means the equivalent mass converted on the piston rod of hydraulic cylinder, $K_1$ and $K_2$ are the controller gains of the variable pitch-controlled system and the torque loading system respectively, $K_{a1}$ and $K_{a2}$ are the servo amplifier gains, $K_{at}$ and $K_{a2}$ are feedback gains of sensors, and $K_q$ is the flow compensation feedback gain.

Fig 5. The bond graph model of electro-hydraulic dynamic loading system including clearance nonlinearity
2.3 Simulation analysis

For the electro-hydraulic dynamic loading process of semi physical simulation experiment in a large scale wind turbine’s variable pitch-controlled system, setting the angle input as a sine signal which amplitude is 30° and frequency is 2Hz, the torque input as a step which amplitude is 200N·m, and the hydraulic parameters are: $P_s=21$ MPa, $P_t=0$ MPa. The step response curves of actual load torque due to different clearance value are shown in Fig 6.
Fig 6. The step response curves of actual load torque due to different clearance value

According to the given signals above, the variable pitch-controlled system would drive the blade reciprocating in the range of ±30°, the torque loading system exerts 200 N·m to the blade at the same time. If the influence of clearance is not considered, that is to say the clearance \( b \) is 0 mm, superfluous torque has been suppressed effectively for the reasons of the structure invariance principle and the compensatory way of flow, as shown in Fig 6(a). When the clearance exists in the transmission chain, and the variable pitch-controlled system is in starting, the spline tooth moves in the clearance \( b \) quickly and then runs into the internal spline, which causes the overshoot of actual loading torque; once the angle of blade reaches to +30° and reversing, the actual torque output would appear the zero state response for the clearance, and the actual torque output decreases to 0 N·m sharply, but did not reach 0 N·m because of the friction in the system; when the spline tooth gets through the clearance and runs into the internal spline, the actual torque output produces peak.

In a word, when the position system is in starting and reversing, the clearance existed in the transmission chain of the electro-hydraulic dynamic loading system strengthens the value and complexity of the superfluous torque, and the bigger the value of clearance is, the greater effect will be. As shown in Fig 6 (b-e), while the clearance are 0.2, 0.4, 0.6 and 0.8 mm respectively, the overshoot of actual torque output are 70.26%, 88.09%, 90.72% and 103.91%; when the clearance is 0.9 mm, the phenomenon of limit cycle oscillations would occur in the loading system, and the system cannot work steadily, as shown in Fig 6 (f).

Because of the clearance, the superfluous torque grows irregularly when the position system is in starting and reversing, which leads to the serious distortions of actual torque output. The loading precision is defined with \( \Delta_d \) as follow:

\[
\Delta_d = \frac{\max|T_d|}{\max|T_f|} \times 100\%
\]

Where: \( T_d \) is the superfluous torque, N·m; \( T_f \) is the actual torque output, N·m.
So, the trend graph of superfluous torque and loading precision due to clearance value is shown in Fig 7, while the value of clearance are 0, 0.2, 0.4, 0.6 and 0.8 mm respectively, the corresponding superfluous torque peak are 3.7, 18.2, 47.7, 110.9 and 164.2 N·m, and the loading precision are 1.9%, 9.1%, 23.9%, 55.5% and 82.1%.

![Fig 7. The trend graph of superfluous torque and loading precision due to clearance value](image)

### 3 CONCLUSION

This paper has established the electro-hydraulic dynamic loading system mathematical model including clearance nonlinearity with the method of power bond graph, which abstracts from the dynamic loading process of semi physical simulation experiment in a large scale wind turbine’s variable pitch-controlled system. When the values of clearance are 0, 0.2, 0.4, 0.6, 0.8 and 0.9 mm respectively, the effects of the clearance on the superfluous torque are studied with 20-sim software, and the conclusions are as follows:

- The clearance existed in the transmission chain of the electro-hydraulic dynamic loading system, strengthens the value and complexity of the superfluous torque at the moment that the position system is in starting and reversing, which leads to reduction of loading precision and dynamic performance.
- Only at the very moment the position system is in starting and reversing, the clearance has the strengthening effects on superfluous torque, and no effects at other moments.
- The moment the position system is in starting and reversing, the bigger the value of clearance is, the greater the superfluous torque will be, when the value of clearance are 0, 0.2, 0.4, 0.6 and 0.8 mm respectively, the corresponding superfluous torque peak are 3.7, 18.2, 47.7, 110.9 and 164.2 N·m respectively.
- The phenomenon of limit cycle oscillations would occur in the loading system when the clearance reaches up to 0.9 mm.
REFERENCES


THE USE OF INHERENT SENSOR EFFECTS IN HYDRAULIC VALVES –
A SIMPLE INSTRUMENTALITY APPROACH WITH MAGNETIC HYSTERESIS

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ABSTRACT

Key aspects such as machine safety, error detection and operator comfort in the sector of mobile machines become more and more important. Because of the increasing complexity of the machine and more extensive specifications triggered by cost pressure and competitors, new technologies are always required.

This article deals with the aspect of the inherent sensor function in magnetic actuators, which are used in hydraulic valves. The attempt is to generate information about the state of the solenoid actuator based on the electrical quantities current and voltage. It is known that a change in the air gap respectively the position of the armature will vary the magnetic resistances for the flux in the magnetic circuit. This is accompanied by a change in the inductance. The resulting current in the winding depends on this inductance. Thus, there exists a relationship between the electrical parameters and the position of the armature. If the position of the armature is known, the position of the spool can be concluded at least for a directly controlled hydraulic valve, and potentially even the related flow rate preconditioned the pressure difference is known.

This paper presents a way of sensorless position detection with simple instrumentality and with very high system restrictions. It investigates how to make the most of this effect without additional hardware compared to contemporary systems. Furthermore, the paper discusses the influence of magnetic hysteresis as detected by test bench measurements.

KEYWORDS: Inherent Sensor Effects, Sensorless Position Detection

1. INTRODUCTION

The availability of the inherent sensor effect provides additional information about the state of the valve for the control use [1] or malfunction can be detected early [2]. According to the state of the art the outcome for systems with an external transducer leads to savings in cost and space. The main focus for all investigations is the applicability in contemporary machines. Detailed studies on this topic were already shown in [3] and [4]. This presented approach is intended for applications from the field of mobile machinery and is supposed
to operate under very simple technical conditions. Furthermore the objective is to implement the inherent sensor effect on existing systems without an additional hardware effort and to avoid increasing system costs. The electronic control unit (ECU) on the machine also has to cope with the requirements for the evaluation of the electrical signals. Due to a predictable distance between the actuator and the ECU, it is expected that the transmission of oscillating signals with high frequency as described in [5] is not possible in this application. One more boundary condition is given by the power supply of these actuators. In general, there is a 12V electrical board system whereof the electrically operated consumers are driven by pulse-width-modulation. However, the requirements on the dynamics compared to industrial, fast-switching valves with a few milliseconds opening time are considerably lower. For the control of today’s proportional valves the use of a so called dither signal [6] is considered state of the art. The dither signal is an alternating superimposed input added to the main control signal. This leads to micro movements of the armature and the valve body in a permanent, fractional dithering. Thus, a significant improvement in controllability is reached by reducing the stick-slip effects. In these investigations exactly this alternating dither signal part is used to determine various electrical parameters of the magnetic circuit. Based on these variables the plunger position is determined. For the following examples only very low plunger speeds are considered. The influence of induced voltage in the cause of motion can be neglected at first and simplifies the analysis significantly.

2. FUNCTIONALITY AND PRINCIPLE OF MEASUREMENT

The principle of the valve body position detection used for these studies is only based on an analysis of the electrical signals voltage $u(t)$ and the current $i(t)$ inside the winding. From these signals the parameter impedance, differential inductance or the phase shift is determined and saved in a previously measured characteristic field with the information of current and valve body position. During valve operation the position can be read out of the inverted map by using the identified parameter and the actual current value. This basic process is shown in Figure 1. The main task of a hydraulic valve is the control a certain mass flow. If the valve geometry is known, it is possible to determine the oil flow preconditioned by the knowledge of the pressure ratio and the plunger position. As the position is not the controlled variable in a later application, but the mass flow, it is more practical to replace the position in the characteristic map by the flow. In this case a constant pressure ratio over the control edge is required. To simplify matters the following tests and measurements are carried out oil-free. Therefore the valve body position remains as the target variable.

![Figure 1. The procedure of sensorless position detection](image)

With such a method, the position of the valve body is not determined - only the position of the armature in the solenoid. Therefore it is important to ensure a geometric coupling between the armature and the sliding part of the valve with its control edges. In general, a force is generated on the plunger by the magnetic flux which then moves the valve body via a coupling bar. The counterforce is realized by a spring. The armature is pushed back to its starting position. A change in the armature position consequently leads to a change to the spool position and vice versa. Figure 2 shows a cross sectional view of such a directly operating hydraulic valve. Also shown is the attached external sensor which should be eliminated by this method.
Figure 2. Design of a direct operated valve

Today’s control of electro-hydraulic valves, or magnetic actuators in general, is usually realized by a pulse-width modulated (PWM) voltage signal. In proportional solenoids, an oscillating signal, the so called dither, is added to the constant voltage part which causes the main deflection of the valve body. $U_{\text{dither}}$, the added part, is in this case a sinusoidal signal with a constant amplitude of 1V and a fixed 100Hz frequency. The dithering of the moving elements reduces stick-slip effects and thus, mechanical hysteresis [6]. As shown in Figure 3, the PWM-voltage $U_{\text{PWM}}$ is the input signal to the magnetic circuit. The circuit’s current response is then adjusted by the alternating part and is led back as the control variable. In the next step the control deviation is calculated and introduced to the current controller.

Figure 3. Signal flow diagramm

In Figure 4 the exemplary waveform of the voltage and current signals are illustrated over a dither period. The small-signal behavior resulting from the PWM-excitation is not displayed here. When a voltage signal with an alternating component (blue line) is inserted, a phase-shifted current signal arises due to the inductive behavior (red line). The current profile depends on among other things, on the angular frequency $\omega$ and the armature position within the coil. This is due to the fact that a change in the air gap will influence the magnetic reluctance and, consequently, the inductance.
During these investigations the dither signal is used to determine various parameters. Based on those parameters a conclusion on the plunger position can be made. Hence, in addition to the phase shift $\Delta \phi$, the extreme values of each dither period are extracted out of the shown signal sequence in Figure 4. The absolute value of the impedance

$$|Z_{\text{stat}}| = Z_{\text{stat}} = \frac{\hat{U}_{\text{stat}}}{\hat{I}_{\text{stat}}}$$

is calculated from the ratio between the corresponding values for voltage and current. The dynamic impedance is similar determined from the electrical signals corrected by the mean value over one dither period:

$$|Z_{\text{dyn}}| = Z_{\text{dyn}} = \frac{\hat{U}_{\text{dyn}}}{\hat{I}_{\text{dyn}}}.$$  

(3)

The transformation from $Z_{\text{dyn}}$ and $\Delta \phi$ into the complex section leads to an apportionment by the resistive component $R_{\text{dyn}}$ and the inductive component $\omega L_{\text{dyn}}$ dependent on the dither frequency $\omega$:

$$Z_{\text{dyn}} = R_{\text{dyn}} + j\omega L_{\text{dyn}}.$$  

(2)

All the analyzed variables are afflicted by considerable noise. Filtering is definitely necessary due to the low sampling rate. A recursively-least-square estimator [7] is applied for the dynamic variables $\omega L_{\text{dyn}}$ and $R_{\text{dyn}}$. The advantage of this estimator is the flexible adjustment of the filter intensity by readjusting the oblivion factor. For the static measurements, whereas the armature velocity is zero, this factor is adjusted to a high value. In the following $\omega L_{\text{dyn}}$ is the prioritized value to determine the position. Nevertheless it is possible to utilize $R_{\text{dyn}}$ or the phase shift as proposed in [8].

3. PROTOTYPE MEASUREMENT

This section starts with the description of the investigated magnetic circuit and experimental equipment which is arranged in a test bench. The method to create the characteristic maps is explained and difficulties on this method are discussed. A comparative measurement is performed between the sensorless procedure and a reference signal. In addition, several influences of hysteresis effects are documented.
3.1. Test Bench and Magnetic Circuit

The magnetic actuator is controlled by a maximum input voltage of 24V. It is possible to position the armature over the full travel length of 7.5mm. For the map recording, the previously explained magnetic circuit shown in Figure 2 is coupled to a test bench. Figure 5 depicts the schematic structure of this test bench. The armature position in the solenoid can be adjusted arbitrarily with an external linear actuator. To measure the resulting force between the plunger and the adjustment unit, a force sensor is applied. Furthermore, the position is recorded as a reference signal by a second transducer. Another measured variable of the test bed is the valve current, which is determined by the voltage drop over a shunt. The control, the processing of the measured signals and the analysis up to the sensorless position signal are realized on a dSpace Autobox. It must be pointed out that in the coupled arrangement a dithering of the armature is largely prevented. However, the difference between the characteristic field and the free operating valve is to be taken into account. To investigate the magnetic circuit in the free operation mode, the adjusting mechanism may be removed. In this case a spring is assembled which provides a corresponding return force, and thus the plunger is pushed back to its starting position.

![Figure 5. Schematic diagram of the test bench](image)

3.2. Characteristic Map Identification and Evaluation

The characteristic maps required for the evaluation are measured on the described test rig. Here the plunger can be placed in any position with the external linear actuator independent of the applied current in the magnetic circuit. At the same time a current can be adjusted into the winding according to the signal flow diagram in Figure 3. The maps displayed in Figure 6 are identified by a permanently controlled average current value calculated over one dither period. Subsequently, the plunger is gradually displaced by the linear actuator. Thereby the electrical quantities (such as current and voltage) are recorded, as are mechanical values, such as the actual position and the force applied by the magnetic flux in the airgap. Additionally, the surface temperature of the magnet housing is measured with an infrared thermometer. Thus, the influence of temperature can be examined, too. After ascertaining an entire constant current line of the solenoid actuator, the current is increased by one step and the travelling distance is passed again. This procedure is repeated until a complete map is generated over the entire position and current range. The armature position is always displaced in the same direction starting with large airgap position. The appropriate variable of the characteristic map is averaged for each operating point for about two seconds. This leads to smooth and noiseless surfaces.
In Figure 6 the operating course is shown with the equilibrium state regulated by the magnetic force working against the spring force (black lines). The respective lower path describes the valve opening when the magnetic force is increased slowly by a rising ramp input signal until the armature reaches a limit plate after a travel distance of 7mm. Each of the upper paths shows the closing process where the average current is slowly reduced. The resulting hysteresis is the combination of magnetic and mechanical hysteresis. The main results from friction. To visualize these lines only the data of current measurement and the displacement sensor are used to project the operating points onto the map surface. It is assumed that all the working points of the valve are located inside this loop, potentially just slightly outside. Consequently during normal valve operation the working range does not extend over the entire characteristic field.

These plotted maps for $\omega L_{dyn}$ and $R_{dyn}$ depend on the amount of current and position. This is caused by the characteristic field survey. However, for the evaluation the position dependencies of the quantities $\omega L_{dyn}$ and $R_{dyn}$ and the average current are necessary. An inversion of e.g. $\omega L_{dyn} = f(I,x)$ to $x = f(I,\omega L_{dyn})$ is required. A basic prerequisite for this operation is a unique link between $\omega L_{dyn}$ and the armature position for a constant current value. For high average currents, the map complies with these requirements. For low current values the simple inversion is not possible because of the ambiguous relationship between $\omega L_{dyn}$ and the position. Hence an estimated value of $\omega L_{dyn}$ is attributable to several positions at one current value. Depending on the desired application it must be verified whether the correct position can be determined for ambiguous correlations based on a plausibility check. By increasing the preload of the spring it is possible to shift the workspace towards a higher current region if a sufficient power reserve is available. Thus ambiguities and flat characteristic field gradients can be avoided. In that case, higher electrical power must be considered and more thermal waste energy needs to be led away.

Parallel to the measurement of the free floating valve, the algorithm for determining $\omega L_{dyn}$ is running online on a dSpace platform. Figure 7 shows the $\omega L_{dyn}$ map from Figure 6 in the front view and represents these online results (red line) in addition to the values that are calculated from the map-determination just out of the external position and current sensor signals (black line). As expected, for the valve opening the two lines are close to each other. But for the closing path there are remarkable differences. The following error in position detection is analysed at the working point ($I_{wp}$, $x_{wp}$). In this example the armature position is about 4.7mm but the obtained one for $\omega L_{dyn}$ combined with the appropriate constant current line for $I_{wp}$ out of the map carries out a sensorless detected position of about 6.1mm. Consequently the error is 1.4mm.

The comparison between the opening phase and the subsequent return movement of the armature demonstrate that the previously measured characteristic map is obviously only useful for one direction. This behavior is based on the effects of magnetic hysteresis.

A slight slope of the constant current line ($\partial \omega L_{dyn} / \partial x$) equates to poor resolution because there is a large error $e$ in the sensorless determined position for just a small error in $\omega L_{dyn}$ identification. Generally in the displayed surfaces a greater slope can be read out at higher current lines. It turns out that dealing with ambiguous lines and low field gradients is the most difficult part. While the imaginary part explains this
problem at low current values and points out a clear link for high currents, the real part shows opposite behavior with a slightly larger slope at lower current (Figure 6). A parallel analysis of both $\omega L_{dyn}$ and $R_{dyn}$ with current dependent switching is quite conceivable.

Figure 7. Front view of the characteristic map for $\omega L_{dyn}$

3.3. Studies on the Hysteresis

The map used for sensorless position detection is recorded by a stepwise constant current value which is regulated by starting at zero Ampere. Afterwards, the armature is positioned by the external actuator gradually from zero position $x=0\text{mm}$ to maximum position $x=x_{\text{max}}$. This procedure is shown in Figure 8 No.1. According to the figure, there are three other alternatives to measure the characteristic field. On one hand the plungers moving direction can be changed, and on the other hand the current value can be lowered from a high value $I_{\text{max}}$ to the desired constant current line instead of increasing from zero. During the opening phase of a free operating proportional valve, the position value, as well as the current value, increase monotonously. Both conditions are included in the survey map version 1. For the return of the free operating armature both directions are reversed. This is considered in version 4 in Figure 8. Before starting the measurement for a special current value it is switched briefly to a high value $I_{\text{max}}$.

Figure 8. Different possibilities for generating the characteristic map
The differences $\Delta\omega L_{\text{dyn}}$ for the free running valve body between the online calculation (compared to the red line in Figure 7) and the projection into the surface just out of the external sensor signals (compared the black line in Figure 7) are depicted over the plunger position in Figure 9. This is done for the characteristic maps version 1 and 4 as displayed in Figure 8. For the ascending part of the free operating system the error according to the measured field version 1 is the lowest. Upon returning the armature, large disparities arise. As expected, the deviation becomes smaller for the valve closing path by using the different recorded map (version 4). Logically, the error for opening is much larger compared to version 1. Furthermore the difference is smaller during the opening path with version 1 than the difference while closing with map version 4. The reason for that is the chosen value $I_{\text{max}}$. The current ramp maximum value during the free valve body experiment is not the same as the value before starting the characteristic field measurement.

A valve body, particularly a controlled proportional hydraulic valve, does not only operate on the two lines shown in Figure 6, but also in between this hysteresis loop so an arbitrary working point can be reached from different directions and current values. Due to this fact, none of the characteristic maps in Figure 8 lead to satisfactory results for the whole operation area on their own, but instead, a satisfactory result is achieved through a composition of the four maps. To particularize this, some measurements on a constant working point are executed. The selected point is at a position of $x=1\text{mm}$ and a mean current value of $I=0.8\text{A}$. In the first consideration the armature position is fixed by the external actuator (Figure 10). Subsequently the current within the winding is varied by different steps but there is always a back step to the investigated working point ($x=1\text{mm}; \quad I=0.8\text{A}$). $\omega L_{\text{dyn}}$ is calculated online and also plotted versus time. The outcome here is a dependence on the signal sequence of current in the past. Different values for $\omega L_{\text{dyn}}$ can be read out at one plunger position and one current average. Combined with the corresponding flat slope for the constant current line the area of $\Delta\omega L_{\text{dyn}}$ in Figure 10 results in a significant position error.

A second similar experiment is done at the same working point. The impressed current is controlled to the constant value of $I=0.8\text{A}$ over the whole time. The armature position is reconverted with the external actuator in different step sizes. The results in Figure 11 give comparable output. There is also a wide range for $\omega L_{\text{dyn}}$ for the identical stroke position and same average current value. Temperature influence can be excluded. Consequently, a sensorless position detection as described with all the limiting boundary conditions with sufficient accuracy is not feasible without detailed considerations on hysteresis effects.
4. CONCLUSION AND OUTLOOK

The examinations about sensorless position detection in this article exhibit a correlation between armature position, current system response and a defined calculated variable. In every investigated characteristic map there are areas with acceptable gradients over the position. But there also some flat sectors with insufficient resolution wherein small measuring inaccuracies lead to considerable errors in position. This effect occurs particularly at low-average current levels. For a possible application it has to be checked whether every part of the generated surface is needed or some areas can be avoided within the normal operation mode. Furthermore the influence of hysteresis was studied. It was learned that hysteresis has an enormous effect on measured precision. It is necessary to clarify whether hysteresis can be mapped by appropriate models in order to improve accuracy. Further considerations will also deal with the issue of geometric optimization to determine, if small geometrical changes can enhance the inherent sensor effect in addition to further investigations into temperature effects on the magnetic circuit.
REFERENCES


ON-LINE ESTIMATION OF DEAD-ZONES IN THE PILOT STAGE OF PROPORTIONAL CONTROL VALVES WITH MAIN-STAGE ELECTRICAL POSITION FEEDBACK

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ABSTRACT

The two-stage proportional directional valves with main stage electrical position feedback are widely used in large-flow applications for motion control or velocity control. The pilot stage is specially designed with two kinds of dead-zones. However, these dead-zones are usually poorly known, increase with wear and tear and change from valve to valve in mass production because of their small sizes. This paper presents a useful method to estimate the dead-zones. With the help of main stage displacement sensor and the current sensor, we use four current parameters to represent the break points of the dead-zones. With these accurate dead-zones, the asymmetry and unknown dead-zones can be compensated for a single valve in mass production more precisely. What’s more, the measuring process can be done in a short time without shutting down the plant systems. Thus the dead-zones valves can be calibrated online when the tracking performance is degraded. Based on the dead-zone parameters estimated by this method and the dual rate cascade control, a digital valve controller is designed. The comparative experimental results show that this method is effective to estimate the asymmetry and unknown dead-zones. This dead-zone detection also can be applied to other electro-hydraulic proportional valve controlled position control systems with unknown dead-zones.

KEYWORDS: Dead-zone nonlinearity, displacement tracking, online estimation, two-stage proportional control valve

1. INTRODUCTION

The two-stage directional flow control valves are widely used in hydraulic systems with a nominal flow rate of 100~1000L/min, such as high-speed hydraulic catapult systems [1] and off-highway machinery [2]. Typically, they use pilot pressure to amplify the electrical actuator signal to shift the main spool. Substantially, these valves are position control systems with the aim of controlling the displacement of the main spool proportional to the command signal quickly and precisely. In these valve systems, the pilot stage works as the key control component and thus the performance and stability of the entire valve system largely depend on the characteristics of the pilot stage [3-4].

Benefiting from the high pressure-flow gain of the pilot stage and main spool position feedback, the two-stage directional flow control valve with a pilot stage of a four-way directional control valve and the main-stage electrical position feedback control has better dynamic performance as well as static performance than those with open-loop control [5]. In order to get a better balance between manufacturing costs and static performance, dead-zones are introduced in the pilot stage, which is a four-way proportional directional valve
with floating centre position. However, the dead zone non-linearity is among the key factors causing delay and error in the hydraulic actuation response since there is a time delay in position response to overcome the dead-zone [6].

In order to decrease the adverse effect of dead-zones non-linearity in hydraulic valves, the most conventional method is to add a dead-zone inverse (DZI) function to cancel or compensate the dead-zone effect [7-9,11-16], as shown in Figure 1. This method is simple and effective only for the condition that the dead-zone in the pilot stage is known or accurately estimated [10]. However, the actual dead-zone values are usually poorly known, increase with wear and tear, and change from component to component in mass production [11].

Three typical methods are proposed by researchers to solve this problem. The first method is to use the fixed dead-zone value measured by off-line identification as the parameter of DZI function [7-9]. The off-line measurement can be achieved by a valve travel test. This method is easy to implement and has been used in a lot of applications such as in Ref [7]. Since the actual dead-zone value can be varied, this dead-zone compensation is not so perfect because the dead-zone may be over compensated or under compensated. In reality, such a dead-zone compensation usually results in a very conservative controller because the closed-loop bandwidth has to be tuned very low to avoid the risk of limit cycle caused by the overcompensation [13].

The second method is to modify DZI compensation method with an adaptive dead-zone compensation method. The unknown or varying dead-zone value can be tuned with the direct or indirect adaptive update law [15-19]. However, most of the adaptive update laws are proposed based on system model. Though the system model can be worked out by the system modelling or system identification, some system parameters (such as the displacement of pilot spool) are not convenient to obtain and some state variables (such as the orifice discharge coefficients) are not accessible.

The third method is to replace the DZI compensation with an advanced dead-zone compensation method, such as neural networks control in Ref [19-21] and adaptive robust control in Ref [22-23]. Though the effectiveness of these methods has been proved both in simulation and experiment, it is complex to implement and the effectiveness has not been widely proven by industry application.

![Figure 1. Dead-zone inverse function in the hydraulic valves](image)

The goal of this paper is to propose a simple and effective method to detect the varying dead-zone values in the pilot stage of a two-stage directional flow control valve. Also, it is efficient and safe for mass production and online detection. Then these values can be used for the traditional DZI compensation designs. Specifically, the dead-zone is described by solenoid current instead of valve spool displacement or input voltage [24], which is more precise and easier for controller design.

In this paper, a typical two-stage solenoid-operated proportional valve and the specially designed dead-zone in its pilot stage are introduced. Based on this valve system, an improved dead-zone description involved with the housing clearance of the valve spool is discussed to show the dead-zone uncertainty for a single valve system. Since there is no displacement sensor in the pilot stage and there is always a current control loop in the solenoid controller, the solenoid current is used to describe the actual dead-zone and the effectiveness is proved theoretically. Then an online automatic dead-zone detection process is designed in a digital controller. Finally, with regular DZI function, comparative experimental results show that this method is effective to attenuate the large trajectory tracking error caused by the uncertain dead-zones. Detailed information will be given in the following chapters.
2. PROBLEM STATEMENT

2.1. The dead-zone in a two-stage directional flow control valve

The dead-zones are common nuisances that exit in many systems, such as the two-stage directional flow control valve system studied in this paper. This valve system consists of a four-way solenoid-actuated directional valve, a main stage with a linear variable differential transformer (LVDT) and the control electronics, as shown in Figure 2 (a).

![Figure 2. (a) Structure of two-stage directional flow control valve and (b) the dead-zones in the pilot stage](image)

The pilot stage is a four-way proportional directional control valve, which is used to control the pressure and direction of fluid flow in the pilot stage. The main stage is a pilot-operated four-way spool valve, which can be seen as a hydraulic cylinder controlled by the pilot control valve. The main spool is centered by a push-pull preload spring. With the LVDT and the control electronics, a position feedback control loop is constructed to control the start, stop, direction and amount of fluid for smooth acceleration and deceleration of a hydraulic actuator.

In the inactive condition, both work ports of the pilot valve are connected to the tank. Therefore, the main spool is held in the center position by the centering spring. If the left solenoid is energized, the pilot valve spool is moved by the solenoid force to the right. Once the overlap is overcome, the connection to tank of right work port is blocked and the right control pressure chamber is established. The flow from pilot oil supply to the right control chamber of the main stage is provided, and then the pressure applied on the control cross-sections causes the main spool to move. In the closed loop controlled condition the main spool is force balanced and is held in this closed loop controlled position. Therefore, the spool stroke and control opening change in proportion to the command value.

There are two dead-zones in both sides of the pilot control valve. One is the center dead-zone, which is introduced to block the control oil when the valve is inactive. When the main spool is shifted to the left, the pilot spool should first be moved to the right to overcome an overlap of \( L_{cL} \) which is defined as the right center dead-zone, as shown in Figure 2 (b). The typical length of the center dead-zone in our research is about 1mm, which is about 50% of the spool stroke. The center dead-zone is necessary because the work ports of the pilot stage should be connected to return tank so that the main spool returns to its center position when the two-stage valve is inactive. Further, it is an energy-saving design because there is no hydraulic energy loss in this center position.

The other dead-zone is the switching dead-zone. When the main spool is held in a desired position on the left, the control pressure \( P_c \) should be kept constant. This can be achieved by switching the pilot between \( P \) to work ports and \( P \) to return tank. A small overlap which can block the control chamber is added between these two switching conditions. In other words, the pilot spool should overcome an overlap of \( L_{cL} - L_{cR} \) (in right direction) to switch between \( P_c \) to \( P_c \) and \( P_c \) to \( T \), as shown in Figure 2 (b). This overlap is defined as the switching dead-zone because it should be overcome to change the moving direction of main spool. The \( L_{cR} \)}
is defined as the right return dead-zone. The length of switching dead-zone in the pilot stage is about 0.2mm, which is about 10% of the spool stroke.

Though the two directions of the pilot stage are designed symmetry, the actual right and left dead-zones may be different because of low cost manufacturing process. Thus there are two center dead-zones \((L_{rc} - L_{rt})\) and two switching dead-zones \((L_{rc}, L_{rt})\) in the pilot stage. Note that there is a spool housing clearance between the spool and valve body, the actual lengths of both center dead-zones and switching dead-zones are shorter than the overlaps viewed from Figure 2 (b). This problem will be discussed in the following chapters.

Note that the main stage may also have overlaps between the main spool and main valve body. This dead-zone in servo valves is about 1~3% of maximum displacement, whereas it can be as high as 30~50% in proportional valves [3]. In this paper, we only discuss the dead-zones in the pilot stage.

### 2.2. The problem caused by the varying dead-zone values

In order to improve the dynamic performance of this two-stage control valve, a DZI function is used to attenuate the large trajectory tracking error caused by dead-zone. Traditionally, the dead-zone values are obtained by a valve travel test and thought to be constant for controller design in mass production.

However, the dynamic performance improvement is limited since the dead-zone value measured by the travel test is not precise enough. What's worse, these dead-zone values always change from valve to valve in mass production. A typical sinusoidal trajectory tracking test plot with constant dead-zone values is shown in Figure 3. The dead-zone values are measured by a valve travel test in right side (positive signal input) and applied to the left side (negative signal input). But the tracking error in the left side is much larger than that in the right side. This problem can be called dead-zone asymmetry compensation. What's more, the tracking position oscillating around the setting command during 0 to 0.5s since the dead-zone values in the DZI function are larger than the actual dead-zone. This tracking performance can be called dead-zone over-compensation. While during 0.5s to 1.0s, the dead-zone values are smaller than the actual dead-zone and the tracking position lags behind the setting command, which can be called dead-zone under-compensation. Though this performance inconsistent can be accepted in most application of these valves, it is harmful for some high-precision hydraulic control systems.

If we can detect the dead-zone values in a more precise way, the problems of asymmetry-compensation, over-compensation and under-compensation may be solved. Besides, traditional valve travel test is not an efficient method for mass production and may be harmful to plant systems. Therefore, the problem to be solved in our research is developing a precise, automatic and safety dead-zone detecting method for this proportional position control system.

![Figure 3. A typical sinusoidal trajectory tracking test plot with fixed dead-zone values](image-url)
3. DEAD-ZONE MODEL AND ANALYSIS

3.1. The dead-zone model

The dead-zone in a system can be understood as the control region of the input variant, where the output variant does not change with the input variant. Similarly, the dead-zone in the pilot stage can be described as the relationship between the spool displacement and the flow rate of the control valve. The typical model for centre dead-zones is a piecewise linear model with asymmetry break points and linear gains [11,12]. Since there are two dead-zones in the control valve, a cascade dead-zones model is proposed by adding a switching dead-zones function after the centre dead-zones model. Thus, both of the dead-zones and switching dead-zones can be involved and the relationship between pilot spool displacement and orifice opening is obvious to understand. The cascade dead-zones can be described as follows:

\[
x_c = DB_1(x_p) = \begin{cases} 
  x_p - (L_w + L_n) / 2 & \text{if } x_p \geq (L_w + L_n) / 2 \\
  0 & \text{if } -(L_w + L_n) < x_p < (L_w + L_n) / 2 \\
  -x_p + (L_w + L_n) / 2 & \text{if } x_p \leq -(L_w + L_n) / 2
\end{cases}
\]

\[
x_s = DB_2(x_c) = \begin{cases} 
  x_c - (L_w - L_n) / 2 & \text{if } x_c \geq (L_w - L_n) / 2 \\
  0 & \text{if } -(L_w - L_n) < x_c < (L_w - L_n) / 2 \\
  -x_c + (L_w - L_n) / 2 & \text{if } x_c \leq -(L_w - L_n) / 2
\end{cases}
\]

The diagram of the cascade dead-zone model is shown in Figure 4.

Assuming the pilot spool moves in the range of \([-L_{\max}, L_{\max}]\). When the left solenoid is active, the pilot spool first closes the connection between the tank port and Port A at \(x_p = L_w\) and after passing the dead-zone region \((L_w \leq x_p \leq L_n)\), opens Port A to the pressure supply port at \(x_p = L_n\). The flow rate of Port A can be written by

\[
Q_{ap} = \begin{cases} 
  -C_{pa} A_p \sqrt{2p_a / \rho} & \text{if } 0 < x_p < L_w \\
  0 & \text{if } L_w \leq x_p \leq L_n \\
  C_{pa} A_p \sqrt{2(p_a - p_p) / \rho} & \text{if } L_n < x_p < L_{\max}
\end{cases}
\]

Similarly, when the right solenoid is active, the flow rate of Port B can be obtained:

\[
Q_{bp} = \begin{cases} 
  -C_{pb} A_p \sqrt{2p_b / \rho} & \text{if } -L_n < x_p < 0 \\
  0 & \text{if } -L_n \leq x_p \leq -L_w \\
  C_{pb} A_p \sqrt{2(p_a - p_p) / \rho} & \text{if } -L_{\max} < x_p < -L_n
\end{cases}
\]

In practice, the majority of variable area orifices of commercial spool proportional valves take the form of a segment of a circle due to the ease of manufacturing of circular ports. And several same circular ports can be designed around the circumference. In this case, the orifice area can be obtained by
\[ A_k = f(n, R_k, y_k) \]
\[
= \begin{cases} 
  \pi nR_k^2 \left[ \cos^{-1} \left( \frac{1 - \frac{y_k}{R_k}}{1 - \frac{y_k}{R_k}} \right) \right], & \text{if } 0 < y_k \leq R_k \\
  \frac{\pi nR_k^2}{2} + 2R_k(y_k - R_k), & \text{if } R_k < y_k
\end{cases}
\]

Then the relationship between orifice areas and pilot spool displacement can be calculated from

\[
\begin{align*}
A_{b_k} &= A_k = f(n_{b_k}, R_{b_k}, -L_{b_k} - x_p) & \text{if } -L_{b_{\text{max}}} \leq x_p \leq -L_{b_k} \\
A_{p_b} &= A_k = f(n_{p_b}, R_{p_b}, L_{p_b} + x_p) & \text{if } -L_{p_b} \leq x_p \leq L_{p_{\text{max}}} \\
A_{p_a} &= A_k = f(n_{p_a}, R_{p_a}, L_{p_a} - x_p) & \text{if } L_{p_{\text{max}}} \leq x_p \leq L_{p_a} \\
A_{a_d} &= A_k = f(n_{a_d}, R_{a_d}, x_p - L_{a_d}) & \text{if } \ L_{a_d} \leq x_p \leq L_{a_{\text{max}}} 
\end{align*}
\]

A typical plot of the pilot valve orifice area for Port A and Port B versus the pilot spool stroke is shown in Figure 5. In this plot, we assume that \( R_{b_k} = R_{p_b} = R_{p_a} = 1.0\, \text{mm} \), \( n_{b_k} = n_{p_b} = 4 \), \( n_{p_a} = n_{a_d} = 2 \), \( L_{a_b} = L_{a_c} = 1.1\, \text{mm} \), \( L_{b_k} = L_{p_b} = 0.9\, \text{mm} \). We can see that the orifice area gradient in tank-side is larger than that in pressure-side.

Also, the dead-zones are only decided by four parameters \([L_{a_d}, L_{a_b}, L_{a_c}, L_{a_c}]\).

![Figure 5. Pilot valve orifice area versus spool stroke](image)

### 3.2. The dead-zone values involved with housing clearance

For two-stage servo-valves with custom matched spool and sleeve, the spool housing clearance between pilot spool and valve sleeve is small (about 2~5 microns [3]), and the dead-zone values are considered as the overlaps or under-laps between the valve spool and valve body. However, in most proportional control valve, the spool housing clearance of the pilot proportional control valve is in the range of 8~15 microns [3] and will increases with tear and wear.

As shown in Figure 6, when the pilot spool passes through a distance of \( L_{a_d} \) within the dead-zone region \((L_{a_d} \leq x_p \leq L_{a_c})\) with a speed \( v_{a_d} \), a leakage flow \( q_{a_d} \) from pressure supply to the right control chamber and a leakage flow from the right control chamber \( q_{a_d} \) to the tank in the pilot valve should be considered. Also, there is a leakage flow \( Q_{a_d} \) from the right control chamber to the tank in the main stage. If the leakage caused by the housing clearance is large enough, the actual dead-zone break points will be smaller than the overlaps or under-laps.
Also, the asymmetry problem may be taken into consideration by using different parameters for the left side and right side. Since these leakages are decided by some structure parameters, such as spool housing clearance, and environment parameters, such as oil viscosity, the actual dead-zone values are not simply equal to the overlaps or under laps and are sensitive to some structure parameters and environment parameters. If we want to measure the dead-zone more precisely, an online detection method should be developed.

4. IMPROVED ONLINE DEAD-ZONE DETECTION METHOD

Since the dead-zone in the control valves is sensitive to some structure and environment parameters, a better way to detect the dead-zone precisely is detecting the varying dead-zone automatically with the valves in the plant systems.

However, there are two difficulties to develop a method like that. One is the safety requirement that the actuator controlled by the valves may not be allowed to move during system maintenance in some plant systems, which means the displacement of the main valve spool should be controlled in a small range. The other difficulty is the efficiency that the detection should be fast and automatic. It is difficult because the dead-zone detection may be invalid with fast speed since the dynamics of the control valve cannot be neglected.

In order to solve these two problems, an improved dead-zone detection method is proposed with both of the displacement $x_m$ and the time interval $T_o$ controlled in a small range, as shown in Figure 7.

In this paper, a current description is proposed for the cascade dead-zones. The description is based on the assumption that the current of the proportional solenoid has a linear relationship with the magnetic force acted on the pilot spool within the stroke of dead-zones at slow speed. This assumption can be explained as follows:

The mechanical model of the pilot stage can be evaluated by the motion equation

$$F_{pm} = m_p \ddot{x}_p + F_{pf} + F_{pv} + F_{pq} + F_{pk}$$ (7)
Since the current response in the solenoid is much faster than the response of the control valve, the magnetic force exerted by the proportional solenoid is calculated assuming a linear behavior

$$F_{pm} = k_s i$$  \hspace{1cm} (8)

The static and dynamic friction forces are evaluated by the Karnopp’s friction model [23]. Static and dynamic friction coefficients are considered constant. The viscous friction can be described by

$$F_{pv} = C_p \dot{x}_p$$  \hspace{1cm} (9)

When the pilot spool moves within the dead-zone stroke, the flow is cut down, so the flow force $F_{pq}$ exerted by the oil flow equals to zero. The force exerted by the spring can be calculated by

$$F_{pk} = 2k_p x_p$$  \hspace{1cm} (10)

Thus, the motion equation can be simplified as

$$k_s i = m_p \ddot{x}_p + C_p \dot{x}_p + 2k_p x_p + F_{pf}$$  \hspace{1cm} (11)

If the speed of the pilot spool is slow enough, $\dot{x}_p$ and $\ddot{x}_p$ can be considered to be zero. Thus the current of the solenoid is linear with the spool displacement, which means the solenoid current can be used to estimate the pilot spool displacement. In engineering practice, if the pilot spool speed is less than 1/10 of the speed when reaching the response frequency of the pilot control valve (over than 20Hz), this requirement can be satisfied and we can obtain the permitted maximum speed of the pilot spool for detection is 2mm/s and the detection period is no less than 1s. Based on the above analysis, four current values can be used to estimate the dead-zones in the pilot stage:

$$I_{rc} = I_{rc}^*, \hspace{0.5cm} I_{ct} = I_{ct}^*, \hspace{0.5cm} I_{lt} = I_{lt}^*, \hspace{0.5cm} I_{tl} = I_{tl}^*$$  \hspace{1cm} (12)

5. CONTROLLER DESIGN

This paper proposes a dual rate cascade control, with a faster inner loop current controller and a slower outer loop trajectory controller. A cascade dead-zones inverse function with four parameters is introduced between two controllers to compensate the cascade dead-zones in pilot stage. Further, a dead-zone detection function is also introduced to update the parameters for the inverse function to optimize the valve performance. A block diagram of the overall control system is shown in Figure 8. All the control functions are implemented by a 32-bit microcontroller based on ARM processor core.

5.1. The current controller

An incremental PI controller for the current loop is proposed and a high current gain is used to track the value given by the dead-zone inverse function rapidly.
Table 1. The technical data of the proportional solenoid

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated voltage input</td>
<td>24V</td>
</tr>
<tr>
<td>Nominal current</td>
<td>2A</td>
</tr>
<tr>
<td>Nominal force</td>
<td>40N</td>
</tr>
<tr>
<td>Resistance</td>
<td>2.6Ω</td>
</tr>
<tr>
<td>Inductance estimated</td>
<td>40mH</td>
</tr>
<tr>
<td>Plunger travel</td>
<td>2.2mm</td>
</tr>
</tbody>
</table>

The solenoid is an industrial proportional solenoid designed with technical data shown in Table 1. For simplicity, the solenoid is considered to be a first-order inertia linear loop with a time constant of about 15ms. The sampling frequency used for current controller is 10 kHz, which is fast enough for current feedback control. Then, the frequency of PWM driver is set to be 25 kHz, which is 2.5 times of the sampling frequency of current controller. This means that there are more than two PWM cycles before the PWM duty is changed.

The PWM duty cycle $D(n)$ output by the current controller can be written:

$$
\begin{align*}
D(n) - D(n-1) &= k_p e_n + k_i e_{n-1}, n > 1 \\
D(1) &= D_0, e_n = I_n - I_{nw}
\end{align*}
$$

(13)

where $I_n$ is the reference input given by the trajectory controller and $I_{nw}$ is the solenoid current measured by the current sensor. $D_0$ is the initial duty, which represents the offset current for the solenoid to overcome the static friction force. Note that a current protection model should be added to prevent the solenoid current from growing too high.

5.2. The trajectory controllers and dead-zone inverse function

There are two position trajectory controllers: an outer trajectory controller for regular control and an inner trajectory controller for dead-zone detection. The outer trajectory controller is a typical PID controller. This controller is designed on the assumption that the current controller has perfect tracking capability. Since the pilot stage has a slow dynamic behaviour, typically less than 50Hz, the position controller can be designed with a sample frequency of 1 kHz, which is also fast enough for position control. Since the sample frequency of inner loop is 10 times of the outer loop, the two loops can be easily achieved without limit cycles.

The current output $I_p$ by the outer trajectory controller can be given as:

$$
I_p = I(n) = \begin{cases} 
  k_p e_n + k_i \sum_{i=1}^{n} e_{i-1} + k_u (e_n - e_{n-1}) & \text{if } n > 1 \\
  0 & \text{if } n = 1
\end{cases}
$$

(14)

$$
e_m = \begin{cases} 
  0 & \text{if } p_m < |M| \\
  p_m - p_m & \text{if } p_m \geq |M|
\end{cases}
$$

(15)

where $p_m$ is the reference input given by the reference command and $p_m$ is the main spool displacement measured by the LVDT. Note that there is a small dead-zone $M$ set to improve the anti-interference performance.
The inner trajectory controller is a typical PI controller. Since this controller is effective with small input and should be fast and precise enough to maintain the main spool in a small displacement, the integration parameter is larger while the proportion parameter is smaller than outer trajectory controller.

There is also a cascade dead-zone inverse sandwiched in the position controller and current controller. The control function is given in equations (16) and (17). These parameters set by manufacturer before delivering can be reset by the online dead-zone detection function by the user.

\[
I_c = DZI_c(I_p) = \begin{cases} 
(I_p + (I_{rc} + I_{rr})/2 & \text{if } I_p > 0 \\
0 & \text{if } I_p = 0 \\
(I_p - (I_{rc} + I_{rr})/2 & \text{if } I_p < 0 
\end{cases}
\]  

\[
I_s = DZI_s(I_e) = \begin{cases} 
(I_e + (I_{rc} - I_{rr})/2 & \text{if } I_e > 0 \\
0 & \text{if } I_e = 0 \\
(I_e - (I_{rc} - I_{rr})/2 & \text{if } I_e < 0 
\end{cases}
\]

6. EXPERIMENTAL VERIFICATION

6.1. The experimental set-up

Figure 9 shows the schematic diagram of the experimental set-up and the test rig for proportional valve. The function generator produces an analogue signal as the control input to the digital controller. The digital controller is a microcontroller designed and constructed specially for improving the main spool dynamic tracking performance. The PWM signal with controlled duty cycle is fed to the PWM modulator and converted to the drive current for both of the solenoids. The main spool is connected with a LVDT and a signal conditioner is used to produce an analogue signal proportional to the main spool displacement.

Though there is a 12-bit A/D converter available in the digital controller, a 16-bit DAC card is used to samples the important information for better accuracy. Thus, four signals of the control system can be obtained: the input signal, the main spool position and both of the solenoid currents. Besides, the UART and J-LINK tool are used for communication between PC and the digital controller. Both of the current sensors for two solenoids have been calibrated with a high-precision ampere meter. The LVDT signal conditioner has also been adjusted to convert ±5mm to ±10V.

The hydraulic power supply was set in a typical condition for this two-stage directional flow control valve. The maximum pressure was 10MPa, and the maximum flow rate was 100L/min. The oil temperature was controlled in the range of 40±2°C.
6.2. The performance of current controller and trajectory controller

The PI controller for the current loop is tuned with the plunger locked at x=0. A Ziegler Nichols [25] tuning procedure was adopted. The parameters for the current controller are $k_p = 3.00, k_i = 0.45, D_i = 54\%$. A test was carried out for right and left solenoids of the pilot stage with this inner current loop. The two curves in Figure 10(a) show that two solenoids are manufactured with almost same dynamics. When switching from one solenoid to another, there is a small period of lag because of the initialization functions for each of the solenoids. The result indicates that the inner current loop is effective for both of the solenoids.

Figure 10. (a) Current response to step input for solenoids (b) Position and current response to 90% step input

A Ziegler Nichols tuning procedure was also adopted for the outer position controller. The response of position and solenoid current with a step of 90% maximum input is shown in Figure 10(b).

The main spool can track the input signal within 43ms. The overshoot is about 2.6%. The current response reveals the adjusting process that the center dead-zone inverses work between 50ms and 100ms. During that period, the current increases rapidly and the pilot spool moves to overcome the center dead-zone. However, when the main spool is approaching the desired position, the switching dead-zone inverse comes into effect. The current jumps between 1A and 1.6A which makes the main spool adjust and reach the desired position quickly.

6.3. The results of dead-zones detection

Figure 11(a) shows the process of detecting the dead-zones for two pilot stages in a two-stage valve. We can see that the displacement of main spool is less than 3% of the single-side stroke, and the time interval for a test cycle is less than 1s. Note that the dead-zone in the main valve is about 15% of its stroke. That means this process can be completed without affecting the working state of two-stage valve.
With the help of UART port, the four parameters were transported to the PC and shown in Table 2. Three set of tests with different solenoids and amplifiers were carried out and the results are shown in Table 2. We can see the dead-zone differences are caused by the pilot stages, rather than the solenoids or amplifiers. Though the two pilot stages are manufactured with same process, the dead-zones are different with about 10–20% tolerances.

<table>
<thead>
<tr>
<th>Test sets</th>
<th>$I_{ce}$ (A)</th>
<th>$I_{ci}$ (A)</th>
<th>$I_{re}$ (A)</th>
<th>$I_{ri}$ (A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference valve</td>
<td>1.560</td>
<td>1.315</td>
<td>1.325</td>
<td>1.216</td>
</tr>
<tr>
<td>Ref. valve with different solenoids</td>
<td>1.589</td>
<td>1.325</td>
<td>1.347</td>
<td>1.198</td>
</tr>
<tr>
<td>Ref. valve with different digital controllers</td>
<td>1.590</td>
<td>1.351</td>
<td>1.347</td>
<td>1.230</td>
</tr>
</tbody>
</table>

6.4. The performance improvement with dead-zones detection

The comparative trajectory tracking results is shown in Figure 11(b). The trajectory tracking error caused by asymmetry dead-zones is significantly attenuated by the cascade dead-zones inverse. Note that when the main spool switching from one side to other side, the pilot spool should overcome the dead-zones, which causes the maximum tracking error (about 10% shown in Figure 11(b)). Since the output power of the solenoid is limited and the time to overcome the dead-zone cannot be shortened any more, thus this tracking error cannot be attenuated by optimizing the controller design, which is not the key point in this paper.

The tracking error improvement can be found during the positive or negative signal tracking history. Before the dead-zone detection, the dead-zone parameters are symmetry for both sides and the root-mean-square errors for the positive and negative signal input are 2.60% and 4.05% separately, while after dead-zone detection the root-mean-square errors reduced to 2.13% and 2.37% for the positive and negative signal input.
7. CONCLUSIONS

In this paper, a typical two-stage solenoid-operated proportional valve is introduced. Based on this valve system, an improved dead-zone description is proposed to analyze the dead-zone uncertainty for a single valve system. The housing clearance of the valve spool is also involved to show the break points of the dead-zone can be varied from different valves and using conditions. With the main stage displacement sensor and the current sensor, four current parameters can be used to represent the break points of the dead-zones. What's more, these parameters can be detected online by the valve controller and can be useful in many discontinuous applications such as forging machine.

Then a dual rate cascade control is proposed, with a faster inner loop current controller and a slower outer loop trajectory controller. A cascade dead-zones inverse function with four parameters is introduced between two controllers to compensate the cascade dead-zones in pilot stage. Further, a dead-zone detection function is also introduced to update the parameters for the inverse function to optimize the valve performance. With regular DZI function, comparative experimental results show that this method is effective to attenuate the large trajectory tracking error caused by the uncertain dead-zones.

8. ACKNOWLEDGMENT

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ABSTRACT

Hydraulic directional control valves are highly integrated mechatronic systems employing different areas of expertise such as mechanics, electrical engineering, fluid mechanics and control theory. During operation of such a hydraulic valve, strong nonlinear effects occur in all individual components. Hence, for the precise positioning of the valve piston a complex closed loop controller with a large number of coupled parameters is used. A systematic modeling leads to knowledge of the internal processes which can be used for the development of new and efficient control strategies. The aim of this work is the modeling of all valve components which have an influence on the optimization of the valve controller. Modeling from a control engineering point of view is only feasible if the model accuracy is adequate while the required simulation time is acceptably short, to allow extensive testing and potentially enable real-time control applications. In respect of this requirements, model classes with lumped parameters are suitable. Existing models usually deal exclusively with individual subsystems and often use models which have very high computational load. This paper describes the systematic upgrading of familiar static models with lumped parameters for electromagnets with a dynamic component. A new identification process is developed that yields a sufficient model accuracy starting from a small number of measurable signals. The simulation quality is validated on unseen data and is compared with the simulation performance of a data-based approach.

KEYWORDS: Hydraulic directional control valve, control engineering, physical modeling, cascaded identification process

1. INTRODUCTION

The detailed modeling of hydraulic valves is necessary for the application of a variety of modern control and condition monitoring concepts. Models can although be used for system analysis and are therefore important for a variety of design and optimization tasks. A systematic modeling leads to knowledge of the internal physical processes. Based on this knowledge, new control strategies can be developed and tested extensively in simulation if the computing time is adequate. Controller optimization for highly nonlinear systems is realized by extensive prototype testing or an automated Hardware-in-the-Loop (HiL) optimization [1]. Models with a good simulation quality and a small computational load can be used to...
minimize the optimization time and the technical measurement effort at the prototype stage [2]. Moreover, some advanced control concepts incorporate system models as state observers or for predictive control, requiring models with real-time capabilities.

Several valve models can be found in the literature. Indeed, the semi empirical models presented in [3, 4] provide a small computing time. However, these models are fully adapted to an explicit input stimulus and thus are not generally applicable. The identification measurements are done in an open loop setup since the piston reaches the mechanical dead end instantly. This data is not representative, since during the real operation of the direct acting valve, the piston never reaches the mechanical dead ends due to its closed loop control. Moreover, these approaches do not deal with the modeling of dynamic processes. The measured dynamic interactions during the selected input stimulus are mapped into static characteristic curves. Other valve models employ complex numerical simulation methods, for example the finite elements method (FEM) which require a high computing time. Like in [5, 6] the goal of such approaches is to develop new valve geometries on a model-based analysis. The novelty of this contribution is the modeling of a directional valve as a complete system for control engineering and the analysis of the model performance employing the closed loop piston position control.

An overview about the modeled valve is given in chapter two. The third and the fourth chapter of this paper describe the modeling of submodels along with the required parameter identification processes. Chapter 5 provides a comparison of the simulation performance of the physical valve model compared to a data-based valve model. In chapter 6 the entire model including the controller structure is simulated in the closed loop and the performance is compared with measurements of the real valve.

2. DIRECTIONAL CONTROL VALVE

The modeling is done for a 4WRPDH6 type valve of Bosch Rexroth AG. Figure 1 shows the valve and its schematic structure according to the analysis of the internal effects. The partitioning into submodels is done in regard of the measurability of signals.

The aim of the closed loop control is the positioning of the valve piston. The position of the piston adjusts the oil flow as a function of the supply pressure at port P. If a flow rate $Q$ should be routed from the pressure port P to port A, the valve piston must be moved in a positive direction (to the right in Figure 1). On the other side, between port B and the tank connection T, a pressure relief takes place. The control concept is cascaded, consisting of an outer position controller and the inner current control. Overall, the position error is converted in a pulse-width modulated voltage $V_{PWM}$. This signal is the input signal of the electromagnetic actuator. The larger the duty cycle is, the higher is the mean current through the coil windings. In this manner, the force the solenoid applies to the piston can be controlled. The force generated by the electro-magnetic actuator is
directly related to the magnetic flux linkage $\psi$. However, this quantity is not easily measurable. Therefore the solenoid model is divided into a current model and a force model. Generally, the solenoid changes its characteristics depending on the armature position since the movement leads to a change of the air gap dimensions. The valve piston is moving as soon as the applied magnetic force $F_m$ exceeds all opposing forces. In order to realize a movement in both directions, the force of the return spring is opposite to the solenoid force $F_m$.

The sum of all forces acting on the movable part of the valve is evaluated in the mechanics model. The movable part consists of the piston, the armature and the solenoid slide and is colored dark in Figure 1. When the piston is in center position, the position $x$ is selected to be zero. A small deflection from this hydraulic zero position leads to an immediate flow rate $Q$. A full valve opening is specified as $x = \pm 100\%$.

A hydraulic flow through the valve is not considered during the parameterization of the controller since the number of possible flows is arbitrary. The goal is to find robust control parameters that provide a good level of quality for all operating scenarios. For this reason the ports A and B are mechanically locked while optimizing the controller. However, a defined pressure $p$ is applied at the pressure port P to provide a defined operating environment. In this paper a hydraulic model is not considered. In case of interest to model hydraulic effects like flow forces impacting on the valve performance the paper [9, 10] can be used. It is shown how hydraulic effects in valves can modeled without complex simulation tools in order to preserve the condition of a small computational time.

3. CURRENT MODEL

The interaction between the electromagnetic quantities is described by the differential equation

$$V_{PWM} = i R + \frac{d\psi(x, i)}{dt}$$

The magnetic flux linkage $\psi$ is not easily measurable within the magnet. Hence, a simple and commonly used approach, for example in [3, 11], is to divide the total voltage drop $V_{PWM}$ into an ohmic voltage drop $V_R$ and an inductive voltage drop $V_L$. In this case, the inductive voltage drop is unknown and has to be calculated recursively.

$$V_L(t_n) = V_{PWM}(t_n) - R \, i(t_{n-1})$$

Here the term $t_n$ describes the current time step. Subsequently, the flux linkage $\psi$ is estimated from the time integration of the inductive voltage $V_L$. The existing dynamics between the electric current $i$ through the coil windings and the magnetic flux linkage $\psi$ within the magnet are not sufficiently described by equation 3.2. For a pulse-width modulated stimulation these dynamics cannot be neglected. An electric current causes a temporal change of the magnetic field which subsequently leads to induction effects. This electromagnetic induction generates eddy currents that result in opposing magnetic fields. Induction effects are caused by both, the armature movement and the current injection. From equation 3.2 and the additional considerations the block diagram in Figure 2 is developed. In this paper the well-known approximation approach with the static nonlinearity [3, 11], is enhanced by a linear transfer function. The 2D-lookup table, modeling the static characteristic of the electromagnetic quantities, is created by using the FEM. Based on the calculated flux linkage $\psi$ and the measured armature position $x_{meas}$ a static current profile $i_{stat}$ is computed. The linear dynamics are identified utilizing this calculated current profile $i_{stat}$ as input and the measured current profile $i_{meas}$ as output. Thus, the output of the linear dynamics $x_{lin}$ is the new output of the current model. Overall, the extended current model is of a Hammerstein structure which is well-known in control theory for modeling nonlinear systems. The best result is obtained for a transfer function with a denominator and numerator of the fourth order. The target position profile $x_{set}$ is specified within the technically relevant operating range and the real armature position $x$, the pulse-width modulated voltage $V_{PWM}$ and the current $i$ are measured. These measured signals are used for the identification process of the linear function and the additional parameters $P$ and $K$ which serve to correct static systematic model errors like an inaccurate
resistance value $R$. The identification process for the values $P$ and $K$ is performed with an evolutionary algorithm, such as presented in [12], which searches for a global optimum of a defined error-criterion between the measured current profile $i_{\text{meas}}$ and the simulated current profile $i_{\text{sim}}$. The evolutionary algorithm starts with a random initial population of parameters which are subsequently evaluated on the basis of the fitness function, the defined error-criterion. The best individuals are selected, according to their simulation quality. A recombination of the random parent individuals takes place. Afterwards these are changed randomly within the mutation phase. The resulting individuals are the so called offspring individuals and are evaluated again. This process is iterated until the termination criterion is reached. In this case, for each value of $P$ and $K$ provided by the evolutionary algorithm, the parameters $a$ and $b$ of the transfer function are identified using the Gauss-Newton algorithm. Only then the full model in Figure 2 is simulated with the measured signals, and the fitness function is evaluated.

For further analysis of the model quality criterions are defined. The normalized root-mean-squared error (NRMSE) is calculated as

$$NRMSE = \frac{\sqrt{\frac{1}{N} \sum_{i=1}^{N} (y_i - \hat{y}_i)^2}}{y_{\text{max}} - y_{\text{min}}} \cdot 100\%.$$  \hfill (3.3)

The Parameter $N$ describes the number of the measurement points $y$ and the simulated output $\hat{y}$. The error is normalized using the extreme values $y_{\text{max}}$ and $y_{\text{min}}$. For this criterion, the value of zero is the best result. The coefficient of determination ($R^2$) is defined as

$$R^2 = \left(1 - \frac{\sum_{i=1}^{N} (y_i - \hat{y}_i)^2}{\sum_{i=1}^{N} (y_i - \bar{y})^2}\right) \cdot 100\%.$$  \hfill (3.4)

It indicates which percentage of the measured variance is captured by the simulation. For this criterion a value of $R^2 = 100\%$ is the best result, while $R^2 = 0\%$ correspond to a model which is only as good as a constant output mean $\bar{y}$. When the linear dynamics are omitted, a value of maximum $R^2 = 71.94\%$ can be achieved for the simulated current profile $i_{\text{sim}}$. The simulated result for unseen validation data and the model structure in Figure 2 is shown together with the experimental result in Figure 3. The result relative to the identification signals is not shown, here the specified position $x_{\text{set}}$ has a similar rectangular character but has a shorter measurement time. As it can be seen, the extension of the static nonlinearity with linear dynamics leads to a significantly improved simulation quality on unseen data. For the current simulation as shown in Figure 3, criteria result in a value of $NRMSE = 4.21\%$ and $R^2 = 94\%$ for the entire simulation time $t$. In [3] hysteresis effects are modeled explicitly, here the duration of the input signal is at least one order of magnitude higher as the longest duty cycle of the pulse-width modulated voltage input $v_{\text{PWM}}$. The static nonlinearity in the block diagram of Figure 2 is not able to model magnetic hysteresis effects. Nevertheless, the simulation result is very good with an error $NRMSE < 5\%$. Therefore, hysteresis effects can be neglected for a pulse-width modulated input voltage.
4. FORCE AND MECHANICS MODEL

A measurement of the dynamic magnetic force $F_m$ for a given current profile $i_{\text{meas}}$ is not possible since there is no measurement instrument that can deal with parasitic force effects like mass inertia and friction. Maxwell's traction formula is not applicable since an exact calculation of the dynamic magnetic flux linkage $\psi$ is not possible. Moreover, the solenoid has a special constructive design to reach a small dependency of the magnetic force from the air gap length, explained in [11]. An alternative approach for the force model is a combination of characteristic solenoid force curves and a dynamic force component. Figure 4 shows this force model as a block diagram. The model is again of a Hammerstein structure. The characteristic force curves can be measured for different constant current levels and a variable armature position or they are obtained from a FEM-simulation. Therefore, they only represent a static relationship between the current $i$, the position $x$ and the force $F_m$. An additional component converts the computed static force $F_{m, \text{stat}}$ into a dynamic force $F_m$. In this way, as much a priori knowledge about the solenoid force as possible can be used for modeling. However, the challenge is that the force dynamics block as well as the dynamic force profile $F_m$ are unknown. Again, linear dynamics are assumed. Therefore the parameters of a transfer function as presented in Figure 2 (top right) has to be identified.

![Figure 4. Block diagram of the force model. The parameters of the dark colored component and the dynamic solenoid force profile $F_m$ are unknown.](image-url)
The mechanical interactions during the movement of the piston are modeled as a lumped spring-mass system with an additional friction model. In [3, 4, 11] the friction force $F$ is modeled using a simple viscous damping $d$ and a coulomb friction term $F_{coul}$. To model constant levels of position that are not caused by mechanical stops, the static friction must also be considered. Thus, a Stribeck friction model [13] of the form

$$F_I = \text{sign}(\dot{x}) \left( F_h - F_{coul} \right) \exp\left(-\frac{\vert\dot{x}\vert}{V_s}\right) + \dot{x}d$$

(4.1)

is used. The term $F_h$ is the static friction force, $V_s$ is a curvature parameter and $\delta$ is a form factor. The second order differential equation is presented in Figure 5 as a block diagram.

![Figure 5. Left: Cross sectional view of the valve body (based on [7, 8]). The dark colored components, the armature, the slide and the piston are movable. Right: Block diagram of the mechanics model.](image)

If the simulated piston velocity $\dot{x}_{sim}$ crosses or is equal to zero, and the absolute value of the difference force $F_{diff} = F_m - F_c - F_p$ is less than the static friction $F_h$, the velocity $\dot{x}_{sim}$ is set to zero and the friction force $F_f$ compensates the difference force $F_{diff}$. The unknowns in the block diagram are the friction parameters in equation 4.1 and the input signal $F_m$. The mass $M$ of all movable components, the spring constant $c$ and the spring preload force $F_p$ are known from data provided by the manufacturer.

The parameters of the force model and the mechanics model cannot be individually identified since the real solenoid force $F_m$ is not measurable. In the following section, a method is presented to determine the parameters of the two models simultaneously. Assuming that the friction parameters are known, the dynamic solenoid force profile $F_m$ can be calculated from the measured piston position profile $x_{meas}$ and the measured current profile $i_{meas}$. Starting with the measured piston position $x_{meas}$ in Figure 6 the resulting force $F_{res}$ can be determined by differentiating twice. Before each differentiation the relevant signal is filtered offline with a third-order Savitzky-Golay filter. With the help of the Stribeck friction model, the spring constant $c$ and the mass $M$ of the movable valve components the dynamic magnet force $F_m$ can finally be calculated. The static force profile $F_{m,\text{stat}}$ can be obtained as it is presented in Figure 4 by utilizing the characteristic force curves. The input data of the linear force dynamics is the static magnet force $F_{m,\text{stat}}$ and the output data is the dynamic magnet force $F_m$. The parameters of the transfer function are identified with the Gauss-Newton algorithm. However, since the friction parameters are also unknown, they must be identified too. Therefore, the inner identification procedure in Figure 6 is integrated in a wider global identification process shown in Figure 7. This figure represents the sequence of an evolutionary algorithm. The components within the gray box belong to the fitness test. In this case, the identification process in Figure 6 is a part of the fitness test in Figure 7. Thus, first of all the parameters of the linear force dynamics are identified with the friction parameters provided by the evolutionary algorithm, afterwards the force and mechanics model in series are
simulated and evaluated with these parameters and the measured signals, the current and position profile, as input data. Afterwards a cost function is evaluated which is generally an error-criterion between the measured piston position profile $x_{\text{meas}}$ and simulated piston position profile $x_{\text{sim}}$. In this case, the NRMSE-criterion is used. Therefore it makes sense to use linear dynamics. A nonlinear dynamics approach tends to adapt to an output force signal $F_m$ which is calculated with unrealistic friction parameters.

Figure 8 shows the result when simulating the force model in Figure 4 in series with the mechanics model in Figure 5. The friction parameters and the force dynamics parameters are identified with the process in Figure 7. Again the best result is obtained for a force transfer function with a denominator and numerator of the fourth order. Only measured signals are used as input data of the force model to evaluate the simulation quality regardless of the error of the current model and the own error in the position $x_{\text{sim}}$. The physical character with the constant position levels can be reproduced by the model with the help of the static friction simulation. This is an excellent result, since it is the model response to the dynamic input current stimulus in Figure 3. For the simulation result in Figure 8 for unseen data the NRMSE-criterion results in a value of just $NRMSE = 3.38\%$ and the $R^2$-criterion results in a value of $R^2 = 98.16\%$. 

Figure 6. Inner identification process for the parameters of the linear force dynamics. The parameters of the dark colored components are unknown.
Figure 7. Evolutionary identification process for the friction parameters. The identification process in Figure 6 is a part of the fitness test.

Figure 8. Simulation result of the force and mechanics model in series for unseen input data.
5. VALVE MODEL

By concatenating the identified submodels, the physical valve model is created. The valve model structure is presented in the upper half of Figure 9. The lower part shows an alternative data-based model.

The simulation quality is expected to be inferior in comparison to the simulation quality of the submodels since the input signals of the submodels, except the pulse-width modulated voltage input $V_{PWM}$, are simulated and therefore error prone. The sensor model consists of a static conversion of position in meter into a relative position given in percent and an additional noise model. An autoregressive-moving-average (ARMA) model is used to model the sensor noise characteristic. This model can be estimated with the help of a regression method for nonlinear minimization problems, for example with the Levenberg-Marquardt algorithm. The maximum measured noise amplitude is very small. Nevertheless, for a realistic modeling of the closed loop control, especially when differentiating elements are used in the controller, a noise model is reasonable. A data-based model of the valve is intended to provide a basis for the comparison of the simulation quality. This is a so called black box model in which only the external behavior, the input-output relation, is of interest. From a high variety of linear and nonlinear data-based models the nonlinear autoregressive-with-exogenous-input (NLARX) model shows the best generalization for the measured valve input and output data. The current control must be a part of the black box model. In order to measure the high frequency pulse-width modulated input voltage $V_{PWM}$ accurately, the sampling frequency must be sufficiently higher. The data-based model cannot handle this large amount of data. By downsampling the accurately measured pulse-width modulated signal, important information gets lost, so that the model is not applicable. Therefore, a data-based model is identified for the relation between the output of the position controller $\hat{\ell}_{\text{set}}$ and the position $x$.

For the simulation result in Figure 10 for unseen input data the NRMSE-criterion results in a value of $\text{NRMSE} = 4.46\%$ for the physical valve model and in a value of $\text{NRMSE} = 8.34\%$ for the data-based valve model. The $R^2$-criterion reaches a value of $R^2 = 96.79\%$ for the physical valve model and a value of $R^2 = 88.76\%$ for the data-based model. If only the criterions are considered, then the advantage of the physical valve model is the slightly higher simulation quality without the disadvantages of a black box model at the same time. However, more important is the fact, that only the physical valve model is able to model constant levels of position and thus to reflect the physical characteristics of the valve. Sometimes, the absolute error of the physical valve model is very large, for example at time $t_2$. This inaccuracy of the piston position would cause a completely different oil flow rate in real application of the valve. A highly nonlinear system cannot be modeled flawless for the entire working area with lumped parameters. In addition, the error propagates through the submodels. Nevertheless, since the physical characteristics can be simulated synchronous to the measured signal, the assumption at this point is that the valve controller with a robust set of parameters is just able to compensate the error of the physical valve model.
6. VALVE MODEL IN THE CLOSED LOOP

Since the modeled valve has a digital controller, the original controller source code can be integrated into the simulation environment. The electrical circuits within the power electronics of the valve are not explicitly modeled because their electrical time constants are significantly smaller and can be neglected. The physical valve model as well as the data-based valve model are simulated in the closed loop, according to the structure in Figure 1, and are compared with the real measured controlled piston position $x_{\text{meas}}$. In this simulation, only the target position profile $x_{\text{set}}$ is specified, and it represents the only non-simulated signal. The real controller and the controller used in the simulation environment use the same parameters. The experiment is performed at a defined pressure $p$ and mechanically locked ports A and B.

Figure 11 shows that only the physical valve model can be stabilized by the controller. The measured and simulated curves are almost identical. From this result it can be inferred that the valve model represents all relevant physical processes within the valve for control engineering. A model based parametrization of the controller is conceivable. The data-based model cannot be used to parametrize the controller since the main physical processes within the valve are not modeled by this approach. The lack of static friction leads to an oscillatory behavior of the piston position $x_{\text{sim}}$.

7. CONCLUSION

In this paper a directional control valve is modeled on the basis of lumped parameters. For modeling the dynamic nonlinear effects within the valve, submodels with a classic Hammerstein structure are used. Thus, pulse-width modulated input signals can be applied. A newly developed cascaded identification process offers the possibility to determine the magnet force $F_m$ and the friction parameters simultaneously. The identified submodels and the physical valve model show a very high simulation quality on unseen input data and are mainly physically interpretable. The simulation result of the closed loop operation with the physical valve model confirms that the valve is represented properly for investigations in the domain of control theory.
The potential of control theory can be better exploited in the field of hydraulic valves with the help of this physical model.

In future the presented model is extended by a hydraulic submodel. It can be used to test the controller and its parameters under simulated operating conditions. Flow rate scenarios can be simulated which are difficult to realize in the experiment but are important for a customer.

REFERENCES


ABSTRACT

Four-port spool type two or three position directional control valves are widely used in hydraulic drives and controls of machinery and equipment. Their design is relatively simple. They basically consist of the housing usually in the form of a cast, a spool and control mechanism. They can be produced as direct or pilot operated. Their advantage is the method of connection to the hydraulic system because most manufacturers follow ISO 4401 (CETOP RP 121H, DIN 24340) standard which allows for interchangeable use of various versions of valves and also manufactured by different manufacturers. One of the main disadvantages of the spool valves is difficulty in obtaining hermetically sealed version, which can be achieved in other types of valves, for example poppet type ones. In the case of logic poppet valves leak-tight occurs at the interface of the poppet cone with the seat edge ensuring complete flow shut off. Presently many manufacturers of hydraulic components offer single logic valves or specific sets performing certain logic functions but they cannot be connected to the hydraulic system as standard directional control valves subplate mounted to ISO 4401 standard. Cracow University of Technology and Ponar Wadowice are investigating the possibility of building four-port valves, in which replacement of the spool by hermetically sealed logic elements would be possible. New innovative pilot operated four-port directional control valve has been developed. It provides total seal for each flow path and in the system can be used interchangeably with a conventional valve. The paper presents the CFD analysis and bench experimental results for the innovative valve. The results show that it is possible to build a hydraulic directional control valve logic type keeping ISO 4401 (CETOP, DIN) connection standard. As it is clear from the research, the innovative solution has good dynamic performance, achieves flow curves similar to the standard hydraulic directional valve, and in some cases even better. In addition, advantage of the new valve made up of logic valves is easy change of configuration of the internal connections, which is not possible in the case of standard spool valves. The new type of directional valve allows to perform additional functions in hydraulic system.

KEYWORDS: directional control valve, logic valve, CFD modelling

1. INTRODUCTION

Directional control valves are among the core components of hydraulic systems. Their task is to change the direction of liquid flow. The basic design solution in this area are spool valves with different control ways: mechanical, electromagnetic, hydraulic, etc. Solenoid controlled valves for small flow ranges are operated directly, while for larger ranges of flow indirectly, using a pilot valve. The directional control valves are produced by many manufacturers of hydraulic components [10]. For the valves that are built in the hydraulic...
system by means of subplates, standard ISO 4401 [5] (or earlier CETOP RP 121H, DIN 24340 is commonly used which allows interchangeable use of the valves of different manufacturers. The most common directional control valves produced according to these standards are the valves for flow rates up to 1000 dm³/min, what corresponds to the sizes NS4 to NS32. Spool type directional control valves for higher flow rate are produced by only a few manufacturers [7]. In case of the need for such a valve for new application or failure of the valve in the existing system, there may be a problem for the speedy delivery of it, as they are usually manufactured to order. The paper presents feasibility study of NS52 size of directional control valve (for flow rate up to 2000 dm³/min) by use of logic valves to ISO 7368 standard. This would create interchangeability of these valves for spool ones.

2. APPLICATION OF LOGIC VALVES TO CONTROL THE DIRECTION OF WORKING LIQUID FLOW

In the catalogues of manufacturers of hydraulic components, various designs of logic valves can be found, as well as different types of their mounting [8,9,11,12]. In this paper, the logic valve NS50 [9] produced by Ponar Wadowice is discussed. Its design is shown in Figure 1. The logic valve consists of a sleeve body 3, spring 4 and poppet 5. The valve is built in the housing and closed by a cover 1, through which a control signal is fed by means of an orifice 2. The poppet 5 is graded and has working surfaces A1, A2, A3 in the proportions shown in Figure 1. In addition, a sealing ring, which provides a leakproof closure between chambers B and X, is placed in the valve sleeve. Figure 2 shows a schematic diagram of one flow path by use of the poppet valve as a pilot pressurized internally. In the closed position, the shown valve unit cuts off the flow in both directions without any leakage. To build a four-way directional control valve, four [2] or more valves logic [3] should be used.

Figure 1. Single logic valve assembly [9]  
Figure 2. Schematic diagram of controlling logic valve by means of poppet valve pressurized internally [3]

Figure 3 shows a schematic diagram of four-port directional control valve built up of four logic valves controlled by three position poppet valve. It can be used to control hydraulic actuator and ensures leakproof...
cut-off of the actuator without the need for additional pilot operated check valves. In this case, the shuttle valves are applied to supply the pilot line with pressure, which occurs at the moment the highest one.

\[\text{Figure 3. Scheme of four-port directional control valve built of logic valves for controlling a double-acting actuator}\]

Considering the above scheme and using the idea presented in the paper [2] a new directional control valve as shown in Figure 4 has been designed. It is compatible with DIN 24340 standard. Four logic valves NS50 to ISO 7368 [6] have been used to build the innovative solution. Logic valves 2 are inserted into the body 1 (Figure 5) from the top and closed with a cover 2, on which a pilot valve 3 is mounted (Figure 4). Inside the cover there are communication channels made. Assembly of the logic valves is a cuboidal housing in a line, enables the design of simple communication channels which can be performed with simple machining operations such as drilling or milling. The communication channels in the body, the working fluid flows through, have mainly circular cross sections and their connections form knees. Geometries of the communication channels are shown in Figure 6. These channels are symmetrical relative to a transverse plane to the longitudinal axis of the body. Thus, the geometry of PB path will mirror PA path and correspondingly, geometry AT path will mirror BT path. For the newly designed valve, name L4-WEH52 is adopted by analogy with spool type codes used by some companies [10].
Figure 4. Directional control valve L4-WEH52: 1 – logic valves assembly, 2 - cover, 3 – pilot directional control valve

Figure 5. Logic valves assembly for mounting according to DIN 24340 standard: 1 - housing, 2 – logic valves, P – pump port, T – return line, A,B – actuator ports
3. CFD ANALYSIS OF FOUR-PORT DIRECTIONAL CONTROL VALVE BUILT OF LOGIC VALVES

3.1. Model of flow

Knowing basic data on the investigating phenomenon such as fluid viscosity, density and kind of flow an adequate model for the analysis can be preliminarily defined. Literature [1,2,4] shows that in similar analysis with hydraulic oil as working medium model $k-\varepsilon$ generally was used. The turbulence kinetic energy $k$ and its rate of dissipation $\varepsilon$ are obtained from the following transport equations:

$$
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_j} (\rho k u_j) = \frac{\partial}{\partial x_j} \left[ \left( \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - Y_M + S_k \tag{1}
$$

$$
\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_j} (\rho \varepsilon u_j) = \frac{\partial}{\partial x_j} \left[ \left( \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 \varepsilon \frac{\rho k}{k} \left( G_k + C_3 \varepsilon \right) - C_2 \rho \varepsilon^2 + S_\varepsilon \tag{2}
$$

In these equations, $G_k$ represents the generation of turbulence kinetic energy due to the mean velocity gradients, calculated as described in modeling turbulent production in the $k-\varepsilon$ models. $G_b$ is the generation of turbulence kinetic energy due to buoyancy, calculated as described in effects of buoyancy on turbulence in the $k-\varepsilon$ models. $Y_M$ represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate, calculated as described in effects of compressibility on turbulence in the $k-\varepsilon$ models. $C_{1\varepsilon}$, $C_{2\varepsilon}$ and $C_3\varepsilon$ are constants. $\sigma_k$ and $\sigma_\varepsilon$ are the turbulent Prandtl numbers for $k$ and $\varepsilon$, respectively. $S_k$ and $S_\varepsilon$ are user-defined source terms. The turbulent (or eddy) viscosity $\mu_t$ is computed by combining $k$ and $\varepsilon$ as follows:

$$
\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}, \tag{3}
$$

where $C_\mu$ is the constant of model.

Remaining assumptions:
- no slip on the walls,
- the liquid is incompressible (hydraulic oil),
- the liquid properties are constant,
- the model is in conditions of thermal equilibrium,
- viscosity of hydraulic liquid is 41 mm$^2$/s.

As a result of conducted simulations for each of the analyzed flow paths distribution of pressure on the walls of channels and velocity in the designated intersections have been received.

3.2. Flow path $PA$

For CFD analysis with the use of ANSYS/Fluent software [13] a discrete model (Figure 7) of flow path $PA$ was built. This model consists of five boundary layers, contains 101097 nodes and 541508 elements. The calculations were made for the following boundary conditions:

- flow rate: $Q = 500, 1000, 1500, 2000$ dm$^3$/min,
- outlet pressure: $p = 0.1$ MPa,
- fluid density: 870 kg/m$^3$,
- the kinematic viscosity of liquid: 41 cSt.

Figure 8 shows the distribution of pressure on the walls of the flow path $PA$ at $Q = 2000$ dm$^3$/min. As it is apparent from Figure 8, pressure is evenly distributed at the channel inlet while during the flow through the valve, there is a stream division into four smaller ones, which are finally joined at the output of the logic valve. This is reflected in the distribution of pressure on the walls of communication channels, as well as in the pressure distribution at stream cross-section (Figure 9). Figure 10 shows the distribution of velocity vectors along the stream lines. As a result of local changes in the cross section of communication channels, the velocity of liquid flowing through flow path $PA$ changes reaching the highest values in the gap of the valve seat and at the output hole of the logic valve.
Figure 8. Pressure distribution on the walls at 2000 dm$^3$/min – flow from port P to A

Figure 9. Pressure distribution on a center plane at 2000 dm$^3$/min – flow from port P to A
3.3. Flow path BT

For CFD analysis, a discrete model (Figure 11) of flow path BT was built. This model consists of five boundary layers, contains 114465 nodes and 614125 elements. The calculations were performed for the corresponding boundary conditions as for the flow path PA. Figure 12 shows the distribution of pressure on the walls of the communication channels at $Q = 2000$ dm$^3$/min. As it is apparent from Figure 12, at the inlet to the channel there is a slight disorder of the pressure distribution caused by changes in the fluid flow direction, while during the flow through the logic valve such phenomena as for path PA can be observed. We are dealing with the division into four smaller streams, then the summation at the output of the logic valve, what is reflected in the distribution of pressure on the walls of the channels, as well as at the stream cross section (Figure 13). Figure 14 shows the distributions of the velocity vectors along the streamlines. Despite local changes of direction, fluid velocity through the channel BT changes its value in the low range, with the highest values in the gap of the valve seat and at the output hole of the logic valve.
Figure 11. Mesh of flow path BT

Figure 12. Pressure distribution on the walls at 2000 dm$^3$/min – flow from port B to T
3.4. Flow characteristics

The results of the calculations carried out by the ANSYS/Fluent software allow to determine pressure loss curves. Figure 15 shows the curves for directional control valve L4-WEH52 made of logic valves and for spool valve WEH52E, which is taken from the catalogue [7]. Pressure losses during the flow through the valve L4-WEH52 are the same for paths PA and PB, what the curve 1a shows. The curve 2a shows the
pressure losses for the flow paths $AT$ and $BT$. The directional control valve WEH52E is characterized by the pressure loss curves 1b – flow paths $PA$, $PB$ and $AT$, and 2b - flow path $BT$. Comparison of these curves suggests that by using the logic valves, four-port hydraulic directional control valve in DIN 24340 (or ISO 4401) standard can be built which could operate at flow rate $2000 \, dm^3/min$ and have performance comparable to corresponding spool directional control valve.

![Figure 15. Comparison of pressure loss curves for directional control valves, L4-WEH52: 1a - $PA$ and $PB$ paths, 2a - $AT$ and $BT$ paths, WEH52E: 1b - $PA$, $PB$ and $AT$ paths, 2b - $BT$ path](image)

4. SUMMARY

The paper deals with the task of building four-port directional control valves for large flows of $2000 \, dm^3/min$ and more. This type of spool valves in ISO 4401 (or CETOP, DIN) standard are not usually available on sale. They can be custom-made, so their availability in the market is limited. The paper proposes the design of this type of valves with the use of logic valves. ISO 4401 (CETOP RP 121H, DIN 24340) standard has been developed mainly for the spool valves. However, as demonstrated in the paper, it can successfully be used also for the directional valve made up of logic valves. This paper presents the design project where standard four logic valves NS50 are built in a row in the housing. Production of such assembly is relatively simple. By means of simple machining operations, the cavities and communication channels can be made in the housing. On mounting of logic valves, the housing is closed with the cover, in which there are control communication channels and the pilot valve installed. The innovative valve was tested by CFD analysis using an ANSYS/Fluent software. Comparison of the pressure loss curves for the new solution and existing WEH52E [7] shows that similar values of flow resistance are obtained. The results prove that the directional control valves for large flows built by using logic valves can be an alternative solution. In addition, it can be
seen that the poppet type valve is characterized by a simple adaptation of the design to other needs, for example:

- the ability to work at higher pressures,
- easy to build customized solutions of unique functions,
- the ability to control the flow throttling on selected flow paths,
- higher speed of switching the direction of flow is also expected.

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DEVELOPMENT OF HIGH PERFORMANCE HIGH FLOW FAST SWITCHING HYDRAULIC DIRECTIONAL CONTROL VALVES

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ABSTRACT

Fast switching hydraulic directional control valves (DCVs) are the key components for digital hydraulics and for applying modern techniques such as PWM in on/off hydraulic drive and control systems. The speed of response and the openings of the currently available pilot operated DCVs limit the use of these modern control techniques to hydraulic systems with relatively low flow rates. Valve speed of response, even for valves designed for this specific purpose, restricts the maximum switching frequency and duty time modulation. To achieve valve fast switching response and high flow capacity, some new configurations for two stage spool valves are proposed. In this paper one of these configurations is presented and its performance is investigated. In this type two 3-way DCVs, each driven by a piezoelectric actuator, are used for piloting the valve spring-centred main spool. The main spool side areas subjected to pilot pressure are chosen to consume small pilot flow and provide enough pilot pressure force for stroking the main spool at high speed. Usage of a two land main spool, which has a decentring static flow force during a part of its stroke, as well as a four land spool which always has static centring flow force are studied. The governing equations are derived and the response is numerically simulated for valves equivalent to servo valves size NG16 available in industry. The results of simulations show the high speed of response and high flow rate of the proposed valve.

KEYWORDS: fast switching, hydraulic, valve, directional, piezo actuator, servo

1. INTRODUCTION

Fast switching valves are one of the fundamental components in the applications of digital hydraulics. Some ongoing developments in valve design and control target towards producing faster, cheaper and more reliable valves with high flow rate capacities that could be integrated in various applications as an economical alternative to proportional and servovalves.

Different switching valve design concepts have been proposed and tested with the target to decrease the switching time; to be typically less than 1 ms, increase the flow rate; to be more than 50 l/min, and improve fatigue life. Winkler [1] presented a piloted seat type switching valve design with a 1.5 ms switching time utilising the Hörbiger plate concept which exploits multiple metering edges to achieve a high flow rate amounting to 90 l/min at 5 bar pressure drop with small displacement. Winkler [2] also presented a piloted valve design using as a pilot stage solenoid operated 3/2 spool valve with nominal flow rate of 10 l/min at 5 bar pressure drop and a total switching time of about 2 ms. The pilot valve drives the main stage consisting of multiple poppet valves to achieve a nominal flow rate of about 100 l/min at a switching time of about 1 ms.
for a pressure drop of 5 bar. Branson [3] presented a design utilising the Hörbiger plate concept and exploiting the high force capacity and dynamics of a piezostack actuator having 12.5 KN blocking force and 68 μm stroke to actuate the DCV valve that could achieve 66.5 l/min at 20 bar pressure drop, and a switching time 1.1 ms at a 15 bar pressure drop. Kudzma [4] presented a servo valve driven 3/2 spool valve with multiple metering edges to achieve high flow rate amounting to 65 l/min at 15 bar pressure drop and to achieve a switching time as small as 0.69 ms.

In order to realize switching times less than 1 ms at flow rates per land exceeding 75 l/min at 5 bar pressure drop, this paper proposes a simple two stage fast switching valve with two 3/2 poppet valves piloting a spring-centred main spool of a 4/3 directional control valve [5]. Each 3/2 pilot valve is driven using a piezostack actuator to exploit the high bandwidth of the actuators combined with the low mass poppets to decrease the pilot valve switching time. DCVs are simple to design and manufacture and are easy to integrate in industrial applications. Two spool types are proposed for the main stage of the DCV: a two land and a four land spool. The layout of the valve is presented, and its governing equations are derived for both types of spools, and the dynamic performance of the valve is estimated at no load, at three modes of operation: high frequency PWM mode, low frequency PWM mode and on/off control with main spool position feedback.

2. VALVE LAYOUT

The proposed valve, which is shown schematically in figure 1, is a 4/3 pilot operated DCV. The valve has a pilot stage consisting of two 3/2 poppet valves.

![Figure 1. Valve layout](image)

The layout of one of the pilot poppet valves is shown in figure 2 in the switched off position. It consists mainly of a poppet (1), a housing (2), a poppet seating piece (3) and a piezostack actuator (4). The pilot supply oil of this valve, of pressure p_{ps}, is through the port X, and the valve is connected to the tank through port Y. The
valve outlet pressure used for stroking the main spool is $p_c$. The pilot supply pressure is connected to both sides of the poppet through holes drilled in it.

![Figure 2](image)

Figure 2. (a) Poppet valve layout, (b) Poppet geometry, (c) Valve symbol

While in the switched off position, the valve output port of pressure $p_y$ is connected to the tank through the port Y as shown in figure 2-a. The valve's poppet is directly driven by a piezostack actuator that switches the valve on when energized. To compensate for the small piezoactuator stroke, the poppet diameter $D_n$ is increased to achieve the required pilot flow rate, keeping in mind maintaining the poppet mass small. When the piezoactuator is energized, it expands, displacing the poppet to close the connecting port between $p_c$ and Y, and in the same time forming an orifice between the pilot pressure line and the valve's output port $p_c$. Once the input signal is removed, the piezostack rebounds to its original length and the poppet is hydraulically restored to its original position. This is achieved without using a spring, but through selecting proper poppet geometry, namely the diameters $D_1$, $D_2$ and $D_n$ shown in figure 2-b, to insure nearly the same effective force on the poppet in the switching on and off directions. This design concept allows minimizing the poppet thickness for the lowest inertia, without large deformation, as well as achieving fast switching off time in comparison with the cases in which return springs are used. When a piezoactuator is energized it pushes its poppet (1) to connect the pilot supply pressure $p_{sp}$ to the corresponding piloting chamber (7) through the resulting open orifice area and the conduits in the valve housing. The pressure difference between the main spool ends displaces it against the centring disc spring (8) on the opposite side. When the pilot valve is switched off, the poppet is displaced backwards, chamber (7) is then connected to the tank line Y, the pressure difference across the spool is removed, and the centring springs return the main spool to its central position. Preloaded multiple disc springs are preferably used because they are compact and have the high stiffness necessary for quickly switching off the main spool. Multiple springs could be placed in series and parallel groups to achieve the required stiffness.

3. VALVE MATHEMATICAL MODEL

The equations which govern the dynamics of the pilot and main stages of the valve are derived as follows.
3.1. Pilot Stage Mathematical Model

The equation of motion of the pilot valve poppet of mass $m_p$ can be written as:

$$f_{\text{piezo}} - f_p = m_p \ddot{x}_p + c_p x_p, \quad 0 < x_p < x_{pm} \quad (1)$$

where the main external forces acting on poppet are assumed to be the piezoactuator driving force $f_{\text{piezo}}$, the poppet restoring pressure force $f_p$, and the viscous friction force acting on the poppet with viscous damping coefficient $c_p$. The maximum value of the poppet displacement $x_p$ is limited mechanically to $x_{pm}$. The static flow forces acting on the poppet are assumed to be negligibly small in comparison to the driving and pressure forces.

The piezoactuator force $f_{\text{piezo}}$ is given according to [6] by:

$$f_{\text{piezo}} = K_{pz} d_{33} V - K_{pz} x_p \quad (2)$$

where $K_{pz}$ is the piezoactuator stiffness, $d_{33}$ is the piezoactuator strain constant and $V$ is the piezostack control voltage. The piezostack actuator is utilized to only push the poppet. The restoring pressure force $f_p$ which opposes the piezoactuator force is used to return the poppet to its initial position, upon de-energizing the piezoactuator, and is given by:

$$f_p = \frac{\pi}{4} p_{sp} (D_1^2 - (D_n^2 - D_2^2)) \quad (3)$$

Figure 3 shows a pilot valve partially open. Applying the continuity equation, the pilot flow rate $Q_{pl}$ from this pilot valve is given by:

$$Q_{pl} = Q_{1p} - Q_{2p} \quad (5)$$

where:

$$Q_{1p} = \text{sign}(p_{sp} - p_c) C_{dp} \pi D_n x_p \sqrt{\frac{2 |p_{sp} - p_c|}{\rho}} \quad (6)$$

$$Q_{2p} = \text{sign}(p_c) C_{dp} \pi D_n (x_{pm} - x_p) \sqrt{\frac{2 |p_c|}{\rho}} \quad (7)$$

where $C_{dp}$ is the coefficient of discharge, $Q_{1p}$ is the pilot flow rate due to the pressure difference $(p_{sp} - p_c)$, $Q_{2p}$ is the flow rate due to the pressure $(p_c)$.
3.2. Main Stage

The displacement of the main spool $x_v$ which has mass $m_v$ is governed by the equation:

$$ (p_{c1} - p_{c2})A_s + f_{ss} + f_{dyn} - kx_v - c\dot{x}_v = m_v\ddot{x}_v, \quad 0 < x_v < x_{vm} \quad (8) $$

where $p_{c1}$ and $p_{c2}$ are the piloting chambers (7) pressures, $f_{ss}$ is the static flow force acting on the main spool, $f_{dyn}$ is the dynamic flow force acting on the main spool, $k$ is the stiffness of the spool centering springs and $c$ is the spool viscous damping coefficient. The spring force is to be high enough to quickly return the spool to its central position when the piezoactuator is de-energized. This can be enhanced through the springs initial compression.

The static flow forces acting on the main spool depend on the spool type. In what follows four land and two land spools are investigated. The static flow force acting on the four land spool can be calculated, according to [7]:

$$ f_{ss} = 2C_dC_vw x_v (p_s - p_l) \cos(\theta_j) \quad (9) $$

where $C_d$ and $C_v$ are the discharge and velocity coefficients respectively, $w$ is the spool valve area gradient, $p_s$ is the main stage supply pressure, $p_l$ is the load pressure across the output ports of the main valve and $\theta_j$ is the jet angle of the inflow and outflow of the main stage control orifices.

The static flow forces acting on the two land spool had been estimated in [8] at different valve openings and supply pressures at no load, using computational fluid dynamics simulation. The results obtained have been used and integrated in the simulation process using look up tables.

The dynamic flow forces acting on the spool are given by [7]:

$$ f_{dyn} = (L_2 - L_1) C_d w \sqrt{\rho} (p_s - p_l) \dot{x}_v \quad (10) $$

where $L_1$ is the damping length at tank chamber and $L_2$ is the damping length at supply chamber and $p_l$ is the pressure difference between main valve output ports.

Applying the continuity equation to the two piloting chambers (7), it can be shown that:

$$ Q_{pl1} - c_i(p_{c1}) = \frac{V_1}{\beta} \frac{dp_{c1}}{dt} + A_p \dot{x}_v \quad (11) $$

$$ -Q_{pl2} + c_i(p_{c2}) = \frac{V_2}{\beta} \frac{dp_{c2}}{dt} - A_p \dot{x}_v \quad (12) $$

where $Q_{pl1}$ and $Q_{pl2}$ are the pilot valve flow rates from and to the two pilot valves, $p_{c1}$ and $p_{c2}$ are the control pressures at the pilot chambers (7), $V_1$ and $V_2$ are the volumes of the piloting chambers, $A_p$ is the main spool cross sectional area at the piloting chambers, $c_i$ is the internal leakage coefficient for the leakage flow from the pilot chamber to the tank chamber of the spool valve and $\beta$ is the oil’s bulk’s modulus.

For a general main spool configuration, the load flow rate $Q_l$ of the valve’s main stage, assuming symmetrical valve orifices, can be calculated as [7]:

$$ Q_l = Q_1 - Q_4 \quad (13) $$

$$ Q_l = Q_3 - Q_2 \quad (14) $$

where:

$$ Q_1 = \text{sign}(p_s - p_1)C_d A_1 \sqrt{\frac{2|p_s - p_1|}{\rho}} \quad (15) $$

$$ Q_2 = \text{sign}(p_s - p_2)C_d A_2 \sqrt{\frac{2|p_s - p_2|}{\rho}} \quad (16) $$
\[ Q_3 = \text{sign}(p_2)C_dA_3 \sqrt{\frac{2|p_2|}{\rho}} \]  
(17)

\[ Q_4 = \text{sign}(p_1)C_dA_4 \sqrt{\frac{2|p_1|}{\rho}} \]  
(18)

A_1, A_2, A_3 and A_4 are the valve orifices' areas, which depend upon the valve geometry, spool displacement and the valve’s lapping conditions.

4. VALVE PERFORMANCE SIMULATION

Using equations from (1) to (18) the valve dynamic performance can be evaluated using a MATLAB Simulink model. The parameters used during simulations are shown in table 1, which are typical for a servovalve of size NG16.

<table>
<thead>
<tr>
<th>Parameter Name</th>
<th>Value</th>
<th>Unit</th>
<th>Parameter Name</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Poppet mass ( (m_p) )</td>
<td>11e-3</td>
<td>kg</td>
<td>Spool valve diametral clearance</td>
<td>10e-6</td>
<td>m</td>
</tr>
<tr>
<td>Four land spool mass ( (m_{s4}) )</td>
<td>130e-3</td>
<td>kg</td>
<td>Oil density ( (\rho) )</td>
<td>860</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Two land spool mass ( (m_{s2}) )</td>
<td>100e-3</td>
<td>kg</td>
<td>Oil Bulk’s modulus ( (\rho) )</td>
<td>1.5e9</td>
<td>Pa</td>
</tr>
<tr>
<td>Four land spool centering</td>
<td></td>
<td></td>
<td>Spring Stiffness ( (k) )</td>
<td>2000e3</td>
<td>N/m</td>
</tr>
<tr>
<td>Spring Stiffness ( (k) )</td>
<td></td>
<td></td>
<td></td>
<td>2500e3</td>
<td>N/m</td>
</tr>
<tr>
<td>Two land spool centering</td>
<td></td>
<td></td>
<td>Spring Stiffness ( (k) )</td>
<td>1000</td>
<td>N.s/m</td>
</tr>
<tr>
<td>Spool diameter ( (d_s) )</td>
<td>20e-3</td>
<td>m</td>
<td>Poppet diameter 1 ( (D_1) )</td>
<td>24e-3</td>
<td>m</td>
</tr>
<tr>
<td>Spool overlap</td>
<td>5</td>
<td>%</td>
<td>Poppet diameter 2 ( (D_2) )</td>
<td>8e-3</td>
<td>m</td>
</tr>
<tr>
<td>Main spool stroke ( (x_{vm}) )</td>
<td>1.6e-3</td>
<td>m</td>
<td>Maximum Poppet Displacement ( (x_{pm}) )</td>
<td>100e-6</td>
<td>m</td>
</tr>
<tr>
<td>Spool valve area gradient ( (w) )</td>
<td>0.035</td>
<td>m</td>
<td>Coefficients of Discharge ( (C_{dp} &amp; C_{dp}) )</td>
<td>0.6</td>
<td>-</td>
</tr>
<tr>
<td>Piloting chamber diameter ( (d_p) )</td>
<td>12.5e-3</td>
<td>m</td>
<td>Piezostack maximum voltage ( (V_o) )</td>
<td>160</td>
<td>V</td>
</tr>
<tr>
<td>Pilot chamber initial volume ( (V_o) )</td>
<td>2.2e-7</td>
<td>m³</td>
<td>Piezostack blocking force at ( V_o ) ( (F_{pma}) )</td>
<td>3000</td>
<td>N</td>
</tr>
<tr>
<td>Spool viscous damping coefficient ( (c) )</td>
<td>900</td>
<td>N.s/m</td>
<td>Piezostack no load displacement at ( V_o ) ( (x_o) )</td>
<td>160e-6</td>
<td>m</td>
</tr>
<tr>
<td>Damping length ( (L_2-L_1) )</td>
<td>5e-3</td>
<td>m</td>
<td>Piezostack natural frequency ( (w_o) )</td>
<td>8000</td>
<td>Hz</td>
</tr>
<tr>
<td>Inflow and outflow jet angle ( (\theta) )</td>
<td>69</td>
<td>deg.</td>
<td>Piezostack Length ( (L) )</td>
<td>116e-3</td>
<td>m</td>
</tr>
</tbody>
</table>
The shown pilot stage poppet diameters; namely $D_n$, $D_1$ and $D_2$ had been determined after several trials so as to yield the required performance.

4.1. Pilot Stage Simulation Results

During the simulation runs, the piezoactuator was provided with an input signal at the time zero, to drive the poppet to its maximum position, and the signal was removed after 0.5 ms. The resulting no-load performance characteristics of the pilot valve are shown in figure 4 at different supply pressures ranging from 5 to 315 bar. The switching on time of the valve is seen to range between 0.08 ms for a flow rate of 12 l/min at pressure drop of 5 bar and around 0.13 ms for a flow rate of 90 l/min at the pressure drop 315 bar. The switching off time of the valve ranges between 0.1 ms at 315 bar pressure drop to about 0.2 ms at 50 bar pressure drop. At 5 bar pressure drop the switching off time is noted to be noticeably longer than 1 ms, this can explained by the fact that the valve depends on the pilot supply pressure $p_{sp}$ for the poppet to return to the switched off position. At 5 bar pilot supply pressure, the restoring force applied to the poppet is not large enough to restore the poppet to its original position before 1 ms. The pilot stage is seen to have a very low switching time despite the comparatively high flow rate. This can be attributed to the small mass of the poppet, the high dynamic force capacity of the piezoactuator and the large poppet diameter.

![Figure 4. No-load switching response of pilot valve under different supply pressures: (a) Poppet displacement (b) Pilot valve flow rate](image)

4.2. Main Stage

The main stage switching characteristics for the valves with four and two land main spools at different pressure drops across the main valve $p_{sv}$, where $p_{sv}=p_2-p_1$, are shown in figures 5 and 6, respectively. The valve pilot stage supply pressure is assumed to be 315 bar. For the valve with four land main spool the switching on time is seen to be ranging from 1 ms for a flow rate 50 l/min at 5 bar pressure drop to 1.3 ms for a flow rate of 435 l/min at 315 bar pressure drop. The switching off time, estimated when the poppet reaches 5% of its stroke, is seen to be at about 1.8 ms at 10% of poppet stroke. The simulation runs in the case of a two land main spool were not carried out for $p_{sv}<100$ bar since the static flow forces were not available for this range of pressures. The two land main spool is noted to be slightly faster than the four land spool especially during switching off, with a switching on time ranging between 1 ms at 315 bar pressure drop to 1.4 ms at 100 bar pressure drop and switching off time of about 1.4 ms.
5. VALVE CONTROL

A number of control methods are proposed to test valve operation at no load. Digital control techniques like Pulse Width Modulation (PWM), Pulse Frequency Modulation (PFM) and Optimised Pulse Modulation (OPM) are some of the techniques used to drive digital switching valves [9]. The PWM technique at high and low frequencies, and an on/off controller with spool position feedback are considered in the following.

Simulations were conducted for valves with four and two land spools when the supply pressure is 315 bar. The switching performance of the valves, driven by a high frequency PWM input signal of 2000 Hz is shown in figure 7 for the four land spool, and in figure 8 for the two land spool.

Figure 7-a represents the case with 25% duty cycle and shows that the pilot valve poppet switches between its extreme positions while the main spool position increases till finally oscillating around a position corresponding to 30% of the spool stroke after nearly 2 ms. Figure 7-b shows the response at 75% duty cycle. The pilot valve poppet is seen to oscillate also between its extreme positions, and after about 2.5 ms, the main spool oscillates around a position 90% of the spool stroke. Figure 8 shows the switching behaviour under the same input signals for a valve with a two land main spool. Figure 8-a shows the response for a 25% duty PWM signal. The main spool is seen to reach a position corresponding to 25% of the full stroke in about 1.5 ms. Figure 8-b shows that for a 75% duty cycle the spool tends to oscillate around 90% of the full stroke after about 2.5 ms. It is worth noting that, with this method of control, the final position of the main spool as a percentage of its full stroke is not proportional to the percentage of the duty cycle.
The response of the valves when the PWM input frequency is as low as 150 Hz is shown in figure 9 for the valve with four land spool and figure 10 for the valve with two land spool. At this frequency and 25% and 75% duty cycles, and for the two types of spools, both the pilot and main valves fully switch on and off, noting that the DCVs are assumed to be of 5% overlap. In this type of control, the average main valve spool position as percentage of its maximum displacement corresponds linearly to the percentage duty cycle. However the operating duty cycle range is limited by the switching time of the valves. Figures 9 and 10 show that the two land spool is slightly faster than the four land spool during switching off.

Figure 7. No load response of valve with four land main spool to input signal with 2000 Hz PWM frequency for (a) 25% duty cycle (b) 75% duty cycle.

Figure 8. No load response of valve with two land main spool to input signal with 2000 Hz PWM frequency for (a) 25% duty cycle (b) 75% duty cycle.

Figure 9. No load response of valve with four land main spool to input signal with 150 Hz PWM frequency for (a) 25% duty cycle (b) 75% duty cycle.
Figures 10 and 11 show the valve response for a different control technique. In this method the input signal to the piezostack actuator is switched on and off according to the difference between the input signal and a feedback signal proportional to the main spool displacement. When the spool position signal is less than the reference signal, the actuator is turned on, and when the spool position signal is equal to or more than the reference signal the actuator is turned off. This is effectively an on/off controller with main spool position feedback. The figures show the step up and step down response of the pilot and main valves to 50% and 75% of full stroke set-points. In figure 10-a and figure 10-b it is seen that the main spool having four lands is tracked to 50% and 75% of the full stroke in less than 1 ms and oscillates around the set point at a frequency related to the switching frequency of the input signal. After 2.5 ms the spool position is tracked back to its center position in about 2.5 ms. Figure 11-a and figure 11-b show the response of the main valve which has a two land spool to the same set points. The results are very similar except for a relatively low step down time, compared to the value for a valve with four land spool.

Figure 10. No load response of valve with two land main spool to input signal with 150 Hz PWM frequency for (a) 25% duty cycle (b) 75% duty cycle

Figure 11. No load response of valve with four land main spool using feedback on/off control on spool position for (a) 50% of stroke set-point (b) 75% of stroke set-point
6. CONCLUSIONS

A new pilot operated fast switching high flow directional control valve of characteristics comparable to proportional and servovalves is proposed for implementation with digital control techniques. The pilot stage of the valve is a novel design consisting of one or two 3/2 poppet valves, depending on the main valve number of working positions. Each pilot poppet valve is to be actuated by a push piezostack actuator, and the poppet return stroke is realized hydraulically at high speed. Two types of main spools are proposed for the main stage. A mathematical model is derived for a 4/3 directional control valve and used to simulate its dynamic performance using typical parameters of a servo valve size NG 16. The switching response of the pilot stage at no load has been found to range between 0.1 and 0.2 ms with a flow rate of 10 l/min for a 5 bar pressure drop. The main stage valve switching time has shown to range between 1 and 2 ms with a flow rate of 50 l/min at a 5 bar pressure drop across the valve.

Three control techniques to drive the valve have been investigated. Simulation of the valve performance have been carried out when PWM control techniques with 2000 Hz and alternatively 150 Hz frequencies are applied, at 25% and 75% duty cycles. Simulations have been also carried out when an on/off controller with feedback for the main spool position is applied, with set points 50% and 75% of the main spool full stroke. The three drive techniques showed the valve high speed of response for the four land and two land spools, amounting to the response speed of servo valves of the same size.

Further investigations into flow characteristics of the proposed pilot poppet valve, and selection of control parameters are required. Experimental validation of the simulation results is planned.

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8. REFERENCES


THE CONCEPT OF SECONDARY CONTROLLED HYDRAULIC MOTORS APPLIED TO THE PROPULSION SYSTEM OF A RAILWAY MACHINE

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ABSTRACT

A hydraulic propulsion system is a convenient choice for the railway construction and maintenance machines because of their large mass and repetitive working cycles with frequent stops. The common approach in industry uses valve-controlled layouts that have high fuel consumption and are often difficult to control effectively. These architectures introduce unnecessary energy dissipations and energy recovery during braking is problematic to implement thus friction brakes are utilized. Moreover, in some commercialized machines an overshoot on the desired stopping position is commonplace. Consequently, a reverse motion is necessary to match the desired working point resulting in unneeded fuel consumption and reduced productivity. In this regard, the presented research study proposes and implements an architecture with secondary controlled hydraulic motors capable of improving the global efficiency of the hydraulic system via regenerative braking while ensuring precise stopping. Since the limited adhesion is a crucial topic in railway applications, the system under investigation is simulated using a high-fidelity dynamic model of the hydraulics combined with a detailed modeling of the wheel/rail interface in order to carefully evaluate the slip coefficients of the wheels. In conclusion, this work shows that secondary control is a feasible solution for railway machinery.

KEYWORDS: displacement control, secondary controlled hydraulic motors, regenerative braking, railway application, wheel/rail interface

1. INTRODUCTION

In the field of mobile hydraulics, diesel combustion engines power the vast majority of machines. In accordance with the U.S. Energy Information Administration [1], diesel fuel has reached a high price with negative indications for the medium-term. Consequently, fuel consumption of mobile hydraulics is an economic issue with significant impact. Furthermore, there is an increasing tendency of introducing stringent emission standards regulations for diesel engines all over the World (e.g. Tier 4 in USA), thus engine downsizing will be common in the near future. Both aspects dictate a continuous strive toward system efficiency improvements. In response, many groups in academia and industry are now working on concepts to reduce the fuel consumptions of mobile machines. The authors’ research group at Purdue’s Maha Fluid Power Research Center has conducted much research on displacement-controlled actuation since this technology has been proposed successfully in [2].

After removing the energy losses generated through fluid throttling in valve-controlled systems, more savings can be achieved by implementing energy recovery. In this regard, much research has been conducted on
hybrid architectures for on-road vehicles (e.g. [3]). Concerning off-road applications, efficient powertrain concepts based on hydrostatic transmissions were evaluated (e.g. [4]). Nevertheless, the energy efficient actuation with secondary controlled hydraulic motors (SCHMs) is not so popular when applied to propulsion systems of mobile vehicles\(^1\). Kohmäscher and Murrenhoff [8] considered this concept for a forklift obtaining good energy savings with respect the original solution with a hydrostatic transmission. Liscouet et al [9] compared the energy consumption of a refuse truck when powered by a power split transmission or by a system with SCHMs showing the convenience of the latter option. For this reasons, the aim of this paper is to gain more insight into a layout with SCHMs applied to railway construction and maintenance machines that is an option not considered in literature.

With reference to the considered application, many of these machines have large mass and perform repetitive working cycles with frequent stops, which creates a good opportunity to implement energy recovery during regenerative braking. The common approach in industry uses valve-controlled hydraulic systems that lead to high fuel consumption. These architectures introduce unnecessary energy dissipations and energy recovery is problematic to implement thus friction brakes are utilized to slow down the vehicles. In addition, some of these machines allow the immediate reuse of the recovered energy to power other active actuators that represent the most efficient option. Moreover, some commercialized railway machines are often difficult to control effectively because a significant overshoot on the desired stopping position is commonplace. Consequently, a reverse motion is necessary to match the desired working point resulting in unneeded fuel consumption and reduced productivity. This is a challenging requirement considering that the maximum wheels’ slip needs to be limited. Therefore, the target of the presented research study is to demonstrate via virtual simulations that a propulsion system based on SCHMs is a feasible solution: recovering energy, controlling the maximum wheels’ slip properly, and eliminating the overshoot on the stopping position are aspects included in this investigation.

2. THE CONSIDERED APPLICATION

2.1. The reference machine

A railway track maintenance machine serves as a reference for this study. The standard machine’s operation consists of automated working cycles with frequent stops. These cycles create an opportunity to implement energy recovery during regenerative braking (an option not implemented in the original machine). Considering a representative automated working cycle, the unique machine’s movement is limited in extension (the exact amount varies depending on different factors). In order to match the desired stopping point, the vehicle is equipped with many sensors. As a result, the distance the machine needs to travel is already known before performing the movement. In addition, since the machine’s working cycles include traveling while operating other actuators, the energy recovered during the slowdown is instantaneously used to power these active actuators. Finally, the machine has both the front and rear axles driven and friction brakes are installed to maintain the stopped position.

2.2. The proposed circuit layout

The hydraulic propulsion system (Fig. 1) consists of a variable displacement pump (VP), driven by the combustion engine (CE), that supplies the variable displacement traction motors (VMs) connected to the machine’s axles through gearboxes (GB). All these units are over-center type. An accumulator (ACC\(_{HP}\)) is connected to the high-pressure line while a relief valve (RV\(_{HP}\)) and a check valve (CV) are installed for safety reasons. The purpose of the high-pressure accumulator in secondary controlled circuits can be:

1. For energy storage;

\(^1\) Examples available in literature concern applications like aircraft lift systems [5] or excavator’s swing motors [6], [7].
2. To increase the capacitance of the system helping the system’s control. In the application discussed in this paper, the second case is utilized because all the recovered energy can be used by other active actuators simultaneously. Then, due to the closed circuit configuration, a low-pressure system is required: a fixed displacement charge pump (CP), a relief valve (RV_LP), and a low-pressure accumulator (ACC_LP) are introduced. This system also serves the purpose of providing flow to the displacement adjustment systems of the variable displacement units. Furthermore, the CE powers also the working hydraulics that is not included in the schematic. Finally, if such an architecture is implemented in a real machine, it is recommended to add on/off valves in order to isolate the high-pressure line when the motors are not used, avoiding leakages. At this stage of the investigation, they were not considered for simplicity.

Even though [4] showed the layout with one independent hydrostatic transmission per axle has an advantage over the architecture with SCHMs for this particular machine, the solution with SCHMs is more flexible for similar railway applications because it makes the energy storage easily feasible. In particular, in some machinery during the regenerative braking the active actuators are not able to consume all the recovered energy. Thus, an architecture with standard hydrostatic transmissions is not convenient because storing this excess of energy is problematic. In addition, the combination of SCHMs and high-fidelity modeling for the rail/wheel interaction were not considered before. Therefore, these aspects represent valid motivations to investigate this circuit layout.

3. THE DYNAMIC MODEL OF THE SYSTEM

A dynamic model of the system under investigation was generated using the MATLAB-Simulink environment. The predominant characteristic is the high-fidelity approach that was chosen to simulate both the hydraulics and the machine’s dynamics.

3.1. Modeling of the hydraulics

The approach used to model the hydraulics is not intended as a new contribution (detailed coverage can be found in [10]). However, the model is included in the paper for completeness and to detail the assumptions made for this study.

3.1.1. Modeling of the hydraulic units

In a displacement-controlled circuit, the positive displacement machines play a major role. Losses introduced by the positive displacement machines have a significant impact on overall system efficiency and fuel consumption. Therefore they were accurately taken into account using losses ($Q_S \geq 0$ and $T_S \geq 0$) calculated via least-square polynomial interpolation from steady-state experimental data. This experimental data was obtained through steady state measurements for different shaft speeds, differential pressures and
displacement settings of a reference variable axial piston unit. The same losses were assumed for all four quadrants of unit’s operation. In order to simulate different sizes of positive displacement machines, volumetric and torque losses were calculated assuming the same efficiency for all unit sizes by applying linear scaling laws (Eq. (1)).

\[
\lambda = \frac{\sqrt[3]{V_{\text{max,scaled}}}}{V_{\text{max,ref}}} \quad \begin{align*}
\dot{V}_{\text{ref}} &= \lambda \cdot \dot{V}_{\text{scaled}} \\
Q_{S,\text{scaled}} &= \lambda^2 \cdot Q_{S,\text{ref}} \\
T_{S,\text{scaled}} &= \lambda^3 \cdot T_{S,\text{ref}}
\end{align*}
\] (1)

Since these units operate both as pumps and as motors, different equations are required to define their effective flow rates (Eq. (2)).

\[
\begin{align*}
Q_e &= \beta \cdot V_{\text{Max}} \cdot \dot{V} - Q_S \quad (\text{pumping}) \\
Q_e &= \beta \cdot V_{\text{Max}} \cdot \dot{V} + Q_S \quad (\text{motoring})
\end{align*}
\] (2)

Equation (3) describes the effective shaft’s torque \(T_e\) for both operating modes.

\[
T_e = \beta \cdot V_{\text{Max}} \cdot \Delta p + T_S
\] (3)

Another key aspect of the variable displacement units is the modeling of the displacement adjustment system. For the swash plate type axial piston machines used in this study, the swash plate adjustment system contains a double acting hydraulic cylinder rigidly connected to the swash plate. The position of the control piston of the adjustment cylinder is controlled by a proportional control valve (PDCV) which is connected to the low-pressure system. As explained in detail by [11], the dynamics of the swash plate adjustment system are determined by the dynamics of the PDCV. Therefore, the dynamics of the swash plate control system can be modeled using a second order transfer function neglecting the dynamics of the mechanical-hydraulic sub-system. The adjustment cylinder velocity \(\dot{x}_C\) is computed via Eq. (4) where \(Q_{\text{PDCV}}\) is the valve’s flow rate, dependent on the valve’s command, and calculated using a constant flow gain. \(A_C\) is the cylinder’s area.

\[
\dot{x}_C = \frac{Q_{\text{PDCV}}}{A_C}
\] (4)

3.1.2. Modeling of the hydraulic accumulators

Bladder accumulators were selected for the proposed layout. Therefore, the accumulator’s gas is assumed to obey the polytrophic law \((p \cdot V^\gamma = \text{constant})\) and the oil inside the accumulator is assumed at the same pressure of the gas (Eq. (5)).

\[
\dot{p}_{\text{Gas}} = \gamma \cdot \left(\frac{p_{\text{ACC}} + p_{\text{atm}}}{V_{\text{Gas}}}\right) \cdot Q_{\text{ACC}}
\] (5)

The flow rate \(Q_{\text{ACC}}\) exchanged between the line and the accumulator is computed applying the orifice equation to the intake port of the accumulator (Eq. (6)), where \(c_o\) represents the flow coefficient, \(A_{\text{ACC}}\) the area of the orifice, and \(\Delta p_{\text{ACC}}\) the pressure difference across the orifice. Additionally, the volume of the gas is calculated making use of the nominal accumulator volume \(V_0\) and of the pre-charge pressure \(p_0\) (again Eq. (6)).

\[
Q_{\text{ACC}} = c_o \cdot A_{\text{ACC}} \cdot \sqrt{\frac{\gamma \cdot \Delta p_{\text{ACC}}}{\rho}} \quad V_{\text{Gas}} = V_0 \left(\frac{p_0 + p_{\text{atm}}}{p_{\text{Gas}} + p_{\text{atm}}}\right)^\gamma
\] (6)

3.1.3. Modeling of the low-pressure system

For the charge pump, efficiency maps are taken into account to evaluate both the volumetric and the mechanical-hydraulic efficiencies thus the effective flow rate and the effective torque are expressed in Eq. (7).
In addition, the flow rate of the relief valve \( Q_{RV} \) is evaluated using a constant flow gain (Eq. (7)): the same approach is used also for the high-pressure RV. Finally, the low-pressure accumulator is modeled using Eqs. (5) and (6).

\[
Q_{e,CP} = V_{Max,CP} \cdot \dot{\theta}_{CE} \cdot \eta_{Vol,CP} \quad T_{e,CP} = V_{Max,CP} \cdot \Delta \rho \cdot \frac{1}{\eta_{Mh,CP}} \quad Q_{RV} = k_{RV} \cdot \Delta \rho_{RV}
\]  

(7)

3.1.4. Pressure build-up equations

A crucial aspect is the determination of pressures in the lines. This step comes from the conservation of mass principle expressed in Eq. (8).

\[
\Delta \rho = \frac{1}{C_H} \int Q \cdot dt
\]  

(8)

Assuming an equal hydraulic capacitance for both lines and including the different sources of flow, Eq. (9) represents the pressure build-up equations for both the high-pressure and low-pressure line. The losses in the transmission lines are neglected.

\[
\rho_{HP} = \frac{Q_{e,VP} - Q_{e,VM,F} - Q_{e,VM,R} - Q_{ACC,HP} - Q_{RV,HP} + Q_{CV}}{C_H}
\]

\[
\rho_{LP} = \frac{-Q_{e,VP} + Q_{e,VM,F} + Q_{e,VM,R} + Q_{e,CP} - Q_{ACC,LP} - Q_{RV,LP} + Q_{RV,HP} - Q_{CV}}{C_H}
\]  

(9)

3.2. Modeling of the combustion engine

The rotational dynamics of the engine is described by Equation (10). \( J_{CE} \) includes also the inertia of the VP.

\[
\dot{\theta}_{CE} = \frac{1}{J_{CE}} \left( T_{CE,ind} - T_{CE,Fr} - T_{CE,Load} \right)
\]  

(10)

The indicated torque \( T_{CE,ind} \) is derived via Eq. (11) from the scaled wide open throttle curve of a known engine: the throttle \( u_{CE} \) is adjusted by a PI controller to maintain a constant engine speed.

\[
T_{CE,ind} = u_{CE} \cdot \left( T_{CE,WOT} + T_{CE,Fr} \right)
\]  

(11)

The engine’s friction \( T_{CE,Fr} \) is defined in Eq. (12) as suggested by [12]. It is function of the engine’s characteristics \( k_{CE} = 75 \text{ kPa for a direct injection engine} \), the engine’s displacement \( D_{CE} \), the engine’s speed \( \dot{\theta}_{CE} \), and the piston’s mean speed \( S_{CE} \).

\[
T_{CE,Fr} = \frac{D_{CE}}{4 \cdot \pi} \left( k_{CE} + \frac{48 \cdot \dot{\theta}_{CE}}{10^3} + 4 \cdot S_{CE}^2 \right)
\]  

(12)

The torque due to the load \( T_{CE,Load} \) in a multi-actuator displacement-controlled machine is equal to the sum of all the variable displacement units’ torques (propulsion system and working hydraulic) and the charge pump’s torque.

3.3. Modeling of the machine’s dynamics

Modeling the vehicle’s dynamics focuses on capturing the longitudinal dynamic of the machine (lateral motion is neglected due to the target of the paper). Three degrees of freedom are considered: the longitudinal position \( x \) of the vehicle’s center of mass, and the angular positions of the two axles \( \theta_f \) and \( \theta_r \). This means the interaction of the rail/wheel is carefully taken into account resulting in a high-fidelity model of this interface.

In Fig. 2, the free-body diagrams of the rolling stocks (left) and of the axles (right) are shown.
The equilibrium of the bodies is written as follows: Eq. (13) refers to the rolling stocks on a flat surface where, due to the low velocity during operation, the aerodynamic resistance force ($F_{\text{Res}}$) is neglected.

$$\begin{align*}
\rightarrow & \quad m_{\text{RS}} \cdot \ddot{x} + F_{\text{Res}} = H_F + H_R \\
\uparrow & \quad m_{\text{RS}} \cdot g = Z_F + Z_R \\
O_R : & \quad Z_F \cdot (l_1 + l_2) + m_{\text{RS}} \cdot \ddot{h} = m_{\text{RS}} \cdot g \cdot l_1
\end{align*}$$

Equation (14) considers a generic axle (the same equations are valid for both the front and the rear ones). A precautionary rolling coefficient was used ($f_{\text{Roll}} = 0.0024$) and the inertia of the axle in the horizontal direction was neglected.

$$\begin{align*}
\rightarrow & \quad F_{\text{Ad}} = H \\
\uparrow & \quad N = Z \\
O : & \quad T_{\text{Axle}} = F_{\text{Ad}} \cdot R + N \cdot f_{\text{Roll}} \cdot R + \ddot{\vartheta}_{Axle} \cdot J + T_{\text{Brake}}
\end{align*}$$

After some algebraic manipulations, Eq. (15) describes the longitudinal dynamics of the machine.

$$\begin{align*}
\ddot{x} &= \frac{F_{\text{Ad},F} + F_{\text{Ad},R}}{m_{\text{RS}}} \\
\ddot{\vartheta}_{Axle,F} &= \frac{T_{\text{Axle},F} - F_{\text{Ad},F} \cdot R - N_F \cdot f_{\text{Roll}} \cdot R - T_{\text{Brake}}}{J_F} \\
\ddot{\vartheta}_{Axle,R} &= \frac{T_{\text{Axle},R} - F_{\text{Ad},R} \cdot R - N_R \cdot f_{\text{Roll}} \cdot R - T_{\text{Brake}}}{J_R}
\end{align*}$$

In order to solve this set of equations, it is necessary to define the adhesion forces and the drive shaft torques. Equation (16) considers the influence of the transmission’s gear ratio ($i_{\text{Tr}}$) and the transmission’s efficiency ($\eta_{\text{Tr}}$) on the resulting axle drive shaft torque.

$$\eta_{\text{Tr}} = \frac{T_{\text{Axle}}}{T_{\text{VM}}} = \frac{T_{\text{Axle}}}{T_{\text{VM}}} \cdot \frac{1}{i_{\text{Tr}}}$$

3.4. Modeling of the rail/wheel interaction

The rail/wheel interaction is a complex nonlinear phenomenon. In view of the investigation’s targets, it is necessary to obtain an accurate estimation of the adhesion force for different working conditions. By means of the approach introduced by [13], different situations can be modeled with very good accuracy.

The adhesion force, expressed in Eq. (17), depends on the adhesion coefficient and on the vertical load ($N$).

$$F_{\text{Ad}} = \frac{2N\mu}{\pi} \left[ \frac{K_A \cdot \epsilon}{1 + (K_A \cdot \epsilon)^2} + \arctan(K_S \cdot \epsilon) \right]$$

The other terms included in the equation are the friction coefficient ($\mu$), the gradient of tangential stress ($\epsilon$), the slip coefficient ($s$), the reduction factors in the area of adhesion ($K_A$), and in the area of slip ($K_S$). They are described hereinafter.
3.4.1. The friction coefficient

The friction coefficient is computed via Eq. (18).

$$
\mu = \mu_0 \left[ 1 - \frac{\mu_\infty}{\mu_0} \right] e^{-B \left( \dot{x} - \dot{\theta}_{A,\infty} R \right) + \frac{\mu_\infty}{\mu_0}}
$$

The required parameters are the maximum friction coefficient at zero slip velocity ($\mu_0$), the friction coefficient at infinite slip velocity ($\mu_\infty$), and the exponential friction decrease coefficient ($B$). The variation of these terms depends on the external conditions (e.g. dry or wet) as explained by [13].

3.4.2. The contact at the rail/wheel interface

The contacts between wheels and rails are assumed as flat elliptical contact areas ($a$ and $b$ are the semi-axes) with normal stress distribution according to Hertz’s theory. The parameters required to describe the interface are the material’s properties (for both the wheels and the rails) and two geometric dimensions, namely the dynamic radius of the wheel ($R$) and the curvature radius of the rail ($R_{R1}$) highlighted in Fig. 3.

Figure 3. Longitudinal (on the left) and transversal cross-sections of the rail/wheel interface

The contact interface influences the adhesion force through the gradient of tangential stress (Eq. (19)).

$$
\varepsilon = \frac{1}{4} \frac{G \cdot \pi \cdot a \cdot b \cdot c_{11}}{N \cdot \mu} \cdot s
$$

The shear modulus of the materials ($G$) and other terms defined in the sequel are included: they are calculated by using the theory of smooth, non-conforming surfaces in contact [14].

First, it is convenient to define some operators (Eq. (20)) that compact the following equations. $K_1$ and $K_2$ take into account the materials of the components by introducing the Young’s modulus ($E$) and the Poisson’s ratio ($\nu$) with the same steel being assumed for both the wheels and rails. Instead, the geometry of the bodies in contact is described by the parameters $K_3$ and $K_4$: the radiuses $R_W$ and $R_{R2}$ do not appear in the formula since the implicit assumption is $R_W = R_{R2} = \infty$ (they refer to flat surfaces). Finally, taking advantage of $K_3$ and $K_4$, another operator of interest is introduced, the angle $\varphi$.

$$
K_1 = K_2 = \frac{1 - \nu^2}{E \cdot \pi} \quad K_3 = \frac{1}{2} \left( \frac{1}{R} + \frac{1}{R_{R1}} \right) \quad K_4 = \frac{1}{4} \left( \frac{1}{R} \right)^2 + \left( \frac{1}{R_{R1}} \right)^2 \quad \varphi = \arccos \left( \frac{K_4}{K_3} \right)
$$

The semi-axes of the contact ellipse are calculated via Eq. (21), where the two coefficients $m$ and $n$ are available in literature [15] using the angle $\varphi$ as reference. Lastly, the Kalker’s coefficient ($c_{11}$) can be obtained utilizing the polynomial fit shown below.

$$
a = m \left( 3 \cdot \pi \cdot N \cdot \frac{K_1 + K_2}{4 \cdot K_3} \right)^{\frac{1}{3}} \quad b = n \left( 3 \cdot \pi \cdot N \cdot \frac{K_1 + K_2}{4 \cdot K_3} \right)^{\frac{1}{3}} \quad c_{11} = 3.2893 + 0.975 \cdot \frac{a}{b} - 0.012 \cdot \left( \frac{a}{b} \right)^2
$$
3.4.3. The slip coefficient

The slip coefficient is calculated via Eq. (22) using the formulation introduced by [16]. In order to avoid numerical issues (e.g. division by zero during machine’s standstill or when the vehicle’s velocity is very low), this definition was preferred to other definitions available in literature.

\[
\dot{s} = -\frac{\left| \dot{x} \right|}{L} \cdot \frac{x}{s} + \frac{\left| \dot{x} \right| - \dot{s} \cdot R \cdot \text{sign}(x)}{L}
\]  

(22)

$L$ represents the relaxation length: it has effects on the dynamic response and is chosen empirically. Then, as detailed in the reference, a damping term needs to be added when the vehicle’s velocity becomes close to zero.

4. DEFINITION OF THE CONTROLLER

4.1. The control problem

All the information necessary to the complete definition of the vehicle’s motion is available ahead of time (e.g. maximum acceleration, stopping point, etc.). Therefore, it is possible to define the desired velocity profile. The final target is the implementation of a system that controls the vehicle by varying the displacements of the hydraulic units. In this regards, it is necessary to differentiate the commands for the two hydraulic motors to maintain the maximum slip/spin of the wheel within an acceptable limit due to the uneven mass distribution on the axles. In order to clarify the last statement, Fig. 4 illustrates the variation of the adhesion coefficient versus slip (the parameters for dry and wet conditions listed in [13] were used).

![Figure 4. Variation of the adhesion coefficient versus slip](image)

Even though it is not accurate to provide a definite value for the maximum slip as different parameters influence the acceptable region of operation, the steady-state value should generally remain below 0.2%. Recalling the adhesion force of a wheel is given by the adhesion coefficient times the vertical load (Eq. (17)), if the same behavior were imposed to two axles where different vertical loads are acting, then the working point of the less loaded axle would migrate toward higher slip values in order to match the requested adhesion force. This behavior is not acceptable because undesirable wear is related to slip [17]. Therefore, it is necessary to implement a controller capable of differentiating the commands directed to the front and the rear motors.

4.2. The control strategy and the proposed controller’s structure

The considered control strategy is the standard one taken into account with SCHMs architectures (e.g. [6]). The primary unit (VP) adjusts its displacement to maintain the reference constant pressure in the high-pressure line. Conversely, the displacements of the motors (VMs) are adjusted to match the desired machine’s motion. Concerning the definition of the controller, a solution not found in literature is proposed for the control of the secondary units (Fig. 5).
The reference high-pressure and the desired vehicle’s dynamics represent the inputs. Regarding the primary unit, both a feed forward term pointed out in Eq. (23) and a feedback term are used to define the desired displacement ($\beta_{SET,VP}$) after saturating the resulting value between $\pm 100\%$. Specifically, $\beta_{FF,1}$ estimates the displacement necessary to manage the flow rates of the motors while $\beta_{FB,1}$, resulting from a PI controller acting on the pressure error $e_P$, corrects this approximation. Then, the PI controller of the variable displacement unit acts on the error $e_{VP}$ between the commanded displacement and the effective displacement ($\beta_{VP}$). The resulting valve command $u_{VP}$ is propagated to the VP block, which represents the variable displacement unit.

$$\beta_{FF,1} = \frac{V_{VM} \cdot \dot{x}_{SET} \cdot \left( \beta_{VM,F,SET} + \beta_{VM,R,SET} \right)}{V_{VP} \cdot \eta_{CE} \cdot \eta_{VM} \cdot \eta_{TR} \cdot \eta_{VM}}$$

$$\beta_{FB,1} = \frac{\dot{x}_{SET} \cdot m_{RS} \cdot R}{V_{VM} \cdot \Delta p_{VM} \cdot i \cdot \eta_{TR} \cdot \eta_{VM}}$$

Moving to the hydraulic motors, again a feed forward term ($\beta_{FF,2}$) and a feedback term ($\beta_{FB,2}$) are proposed. The feed forward ($\beta_{FF,2}$ in Eq. (23)) provides an estimation of the motors’ displacements that are necessary to generate the total torque required to match the desired vehicle’s dynamics. The feedback corrects this approximation by using a PID with anti-windup on the error $e_{\dot{x}}$, resulting from the commanded and simulated vehicle’s velocity comparison. By adding these two contributions, the displacements $\beta_{SET,VM}$ is computed. At this point, a variable gain ($\alpha$) is used to suitably manipulate the final commanded displacement for both motors ($\beta_{SET,F}$ and $\beta_{SET,R}$). Then, the internal controllers of the secondary units act as described beforehand for the primary unit. Concerning the gain $\alpha$, it assumes two values (one for the acceleration, another one for the deceleration) in order to always have more tractive or braking torque on the rear axle.

5. SIMULATION OF THE PROPOSED SOLUTION

The reference case considered to test the system refers to a working cycle of 3.5 seconds: after the acceleration and the deceleration that both last 1.25 seconds, the machine remains at standstill. In order to balance the recovered power, a constant resistive torque was added to the CE’s shaft. This is a simplified approach to count the load dependent on the working hydraulics. Concerning the system’s simulation, the starting point is the scenario resulting from the static sizing suited for the working cycle described above (reference case). Only sizes of components available on the market were considered. In detail, a compromise in the selection of the high-pressure reference value was adopted in order to avoid excessive pressure (it means higher losses) or motors that are too big (unsuitable solution) in the case of low desired pressure. For this reason, other simulations involving different system’s parameters will be presented with the intention of understanding the influence of these variables in the system’s response.
5.1. The reference case

The analysis starts from the comparison between the commanded and the simulated vehicle’s velocity (Fig. 6 left side). The result in terms of tracking is satisfactory. As visible considering the velocity error (lower plot), the only significant discrepancy takes place during the transition from machine’s acceleration to deceleration. Nevertheless, due to the short duration, it does not affect the system’s behavior: in fact, the final position remains within the requested limits. As a note, the friction brakes are set after the final position is reached for the simulated cycle (at 2.5 seconds) in order to maintain it during standstill. Thus, the slowdown is realized only with regenerative braking.

![Figure 6. Machine’s velocity and relative error (left) and pressures with relative error (right)](image)

The right side of Fig. 6 presents the pressures in the two lines and the reference value for the high-pressure. As confirmed by the pressure error (lower plot), there is a good match between the commanded and the achieved pressures. The small spikes at 1.5 seconds are caused by the transition from machine’s acceleration to deceleration because the three units go over-center to perform regenerative braking. Displacements of all three units are plotted in Fig. 7, left side (the commanded values are not reported because they are extremely close to the simulated ones). At about 1.5 seconds, the primary unit reaches full displacement: it causes the unpleasant discrepancies in the tracking of both the high-pressure and vehicle’s velocity that were mentioned previously. However, this behavior was accepted to improve the energy efficiency of the system: if a bigger primary unit were installed, the tracking would be better but the unit’s displacement during the machine’s acceleration and part of the slow-down phase would remain very low, contributing to a lower system’s efficiency.

![Figure 7. Servo-pumps’ displacements (left) and slip coefficients of both axles (right)](image)

The slip coefficients of the two axles are shown on the right side of Fig. 7. First, the working conditions of the two axles are identical due to the appropriate intervention of the controller. This is a positive aspect because it means the system works properly exploiting all the available adhesion while maintaining a limited slip. Furthermore, there is always a smooth transition from one working condition to another without spikes and this aspect represents an interesting advantage of the proposed layout. Ultimately, the system’s controllability was proven because the machine’s stopping position and the slip coefficients of both axles are completely acceptable for these applications. Moreover, the functioning results extremely stable as confirmed by the smooth variations of both the slip coefficients and the servo-pumps’ displacements: this is a remarkable point in favor of this solution.
5.2. System’s sizing study

In order to gain a better understanding of the system’s behavior, some system parameters were varied. In detail, the reference high-pressure, the primary unit’s size, and the accumulator’s nominal volume are modified one at a time. The four resulting scenarios are summarized in Table 1: the parameters highlighted are the ones that are adjusted while Case 1 represents the reference case simulated beforehand. The target is to evaluate the impact of different high-pressures, diverse units’ sizes, and other accumulators’ volumes.

### Table 1. Synthesis of the considered scenarios

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Case 1</th>
<th>Case 2</th>
<th>Case 3</th>
<th>Case 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference high-pressure</td>
<td>bar</td>
<td>210</td>
<td>280</td>
<td>210</td>
<td>210</td>
</tr>
<tr>
<td>Maximum displacement $V_{P_1}$</td>
<td>cm$^3$/rev</td>
<td>75</td>
<td>75</td>
<td>105</td>
<td>75</td>
</tr>
<tr>
<td>Maximum displacement $V_{M_L}$ and $V_{M_R}$</td>
<td>cm$^3$/rev</td>
<td>55</td>
<td>55</td>
<td>55</td>
<td>55</td>
</tr>
<tr>
<td>Accumulator nominal volume</td>
<td>L</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>10</td>
</tr>
</tbody>
</table>

The plots presented in Fig. 8 summarize the system’s behavior after these adjustments. Starting from the top, the only significant change in the pressure tracking is given by Case 4 (the bigger capacitance of the line deals better with the over-center transition of the units). Nevertheless, the only improvement in the velocity tracking (plot below) takes place in Case 2: due to the higher value of the high-pressure, the motors need to perform a smaller variation in terms of displacement. Therefore, the response is more precise during the transients, especially from vehicle’s acceleration to deceleration.

Comparing simulation results for the four different sizing cases, the following conclusions can be made:

- Machine’s stopping positioning: it is basically equivalent in all the simulated scenarios (gap with the target always below one centimeter). For the reasons stated above, Case 2 is better but the difference is not remarkable for the considered application.
- Slip coefficients: there are no important differences. The working conditions of the two axles are always identical and the transients are always performed smoothly.
- Energy consumed by the propulsion system (Fig. 8 on the right). Case 1 consumes less energy (34.04 kJ) while the other cases are most demanding. In details, Case 4 (35.29 kJ) absorbs more energy because the bigger accumulator affects during the vehicle’s acceleration. Then, Case 3 (36.82 kJ) behaves considerably worse due to bigger size of the primary unit. Since it is working at a lower displacement for most of the time, the losses are bigger. This trend is very similar to Case 2 (37.26 kJ): in this situation, the losses in the hydraulic units depends on the higher pressure.

![Figure 8. High-pressure error (left), velocity error (left), and consumed energy (right)](image-url)
Since the only relevant change resulting from this analysis refers to the consumed energy, one can conclude that the sizing of the three units and the accumulator should be chosen to favor the energy saving since the results in terms of machine’s dynamics do not change considerably. Therefore, units as small as possible and high-pressure as low as possible should be preferred. Lastly, it seems appropriate to install a small accumulator if energy storing is not required.

6. CONCLUSIONS

This paper analyzed an energy efficient displacement-controlled architecture for the propulsion system of a railway maintenance machine. After motivating its introduction and describing its structure, the dynamic model used to run virtual simulations was explained. A detailed description of the high-fidelity rail/wheel interaction model was included within the models discussion. The control problem for the application was considered and a suitable controller was proposed. The, the system utilizing regenerative braking was simulated for a realistic scenario of the machine’s motion. The system does not store energy because other hydraulic actuators installed in the machine instantaneously use the recovered brake energy. The results of this preliminary evaluation of the new system’s behavior are encouraging since the machine behaves as expected and the functioning is smooth: in other words, controlling the wheels’ slip properly, and eliminating the overshoot on the stopping position were confirmed. Therefore, this work shows that secondary control is a feasible solution for railway machinery.

NOMENCLATURE

**List of abbreviations**

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACC</td>
<td>Hydraulic accumulator</td>
</tr>
<tr>
<td>CE</td>
<td>Combustion engine</td>
</tr>
<tr>
<td>CG</td>
<td>Center of gravity</td>
</tr>
<tr>
<td>CP</td>
<td>Charge pump</td>
</tr>
<tr>
<td>CV</td>
<td>Check valve</td>
</tr>
<tr>
<td>GB</td>
<td>Gearbox</td>
</tr>
<tr>
<td>HP</td>
<td>High-pressure</td>
</tr>
<tr>
<td>LP</td>
<td>Low-pressure</td>
</tr>
<tr>
<td>O</td>
<td>Center of the machine’s wheels</td>
</tr>
<tr>
<td>PDCV</td>
<td>Proportional direction control valve</td>
</tr>
<tr>
<td>RV</td>
<td>Relief valve</td>
</tr>
<tr>
<td>VM</td>
<td>Variable displacement hydraulic motor</td>
</tr>
<tr>
<td>VP</td>
<td>Variable displacement hydraulic pump</td>
</tr>
<tr>
<td>$F$</td>
<td>Front axle</td>
</tr>
<tr>
<td>$R$</td>
<td>Rear axle</td>
</tr>
<tr>
<td>$\text{ref}$</td>
<td>Reference value of a magnitude</td>
</tr>
<tr>
<td>$\text{scaled}$</td>
<td>Scaled value of a magnitude</td>
</tr>
<tr>
<td>$\text{SET}$</td>
<td>Commanded value</td>
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**List of symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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</thead>
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<tr>
<td>$a$</td>
<td>Semi-axis of the contact ellipse</td>
<td>[mm]</td>
</tr>
<tr>
<td>$A$</td>
<td>Area</td>
<td>[mm$^2$]</td>
</tr>
<tr>
<td>$B$</td>
<td>Friction decrease coefficient</td>
<td>[s/m]</td>
</tr>
<tr>
<td>$b$</td>
<td>Semi-axis of the contact ellipse</td>
<td>[mm]</td>
</tr>
<tr>
<td>$c_{11}$</td>
<td>Kalker’s coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>$c_e$</td>
<td>Flow coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>$C_H$</td>
<td>Hydraulic capacitance</td>
<td>[m$^3$/bar]</td>
</tr>
<tr>
<td>$D_{CE}$</td>
<td>CE’s volumetric displacement</td>
<td>[cm$^3$]</td>
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<td>Units</td>
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<tr>
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<td>-------------</td>
<td>-------</td>
</tr>
<tr>
<td>$E$</td>
<td>Young’s modulus of the material</td>
<td>[N/m²]</td>
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<tr>
<td>$F_{Ad}$</td>
<td>Adhesion force</td>
<td>[N]</td>
</tr>
<tr>
<td>$F_{Res}$</td>
<td>Aerodynamic resistance force</td>
<td>[N]</td>
</tr>
<tr>
<td>$f_{Roll}$</td>
<td>Rolling resistance coefficient</td>
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</tr>
<tr>
<td>$g$</td>
<td>Gravity acceleration</td>
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<tr>
<td>$G$</td>
<td>Shear modulus of the material</td>
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<td>$H$</td>
<td>Horizontal force</td>
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<td>$h$</td>
<td>Height used to localize the CG</td>
<td>[m]</td>
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<tr>
<td>$i_{Tr}$</td>
<td>Gear ratio</td>
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<tr>
<td>$J$</td>
<td>Moment of inertia</td>
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<tr>
<td>$k$</td>
<td>Flow gain</td>
<td>[L/min/bar]</td>
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<td>Constant (rolling parts’ material)</td>
<td>[m²/N]</td>
</tr>
<tr>
<td>$K_3, K_4$</td>
<td>Constant (rolling parts’ geometry)</td>
<td>[1/m]</td>
</tr>
<tr>
<td>$K_A$</td>
<td>Reduction factor (area of adhesion)</td>
<td>[-]</td>
</tr>
<tr>
<td>$K_S$</td>
<td>Reduction factor (area of slip)</td>
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</tr>
<tr>
<td>$L$</td>
<td>Relaxation length (slip coefficient)</td>
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<tr>
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<tr>
<td>$m$</td>
<td>Parameter used in the Hertz’s theory</td>
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</tr>
<tr>
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<tr>
<td>$N$</td>
<td>Vertical force</td>
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<tr>
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<td>Pressure</td>
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<tr>
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<td>$\dddot{x}$</td>
<td>Rolling stocks’ linear acceleration</td>
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Z  Vertical force  [N]
β  Normalized displacement  [-]
γ  Polytrophic coefficient  [-]
Δp  Pressure difference  [bar]
ε  Gradient of tangential stress  [-]
η  Efficiency  [-]
λ  Scaling factor  [-]
µ  Friction coefficient  [-]
µ0  Friction coefficient at zero slip velocity  [-]
µ∞  Friction coefficient at infinite slip velocity  [-]
ν  Poisson's ratio of the material  [-]
φ  Parameter used in the Hertz's theory  [rad]
θ  Shaft angular position  [rad]
ϕ  Shaft angular velocity  [rad/s]
ϕ  Shaft angular acceleration  [rad/s²]

REFERENCES

INVESTIGATION AND IMPROVEMENT OF THE ENERGY EFFICIENCY OF HYDRAULIC DEEP DRAWING PRESSES

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ABSTRACT

The importance of the energetic properties of hydraulic deep drawing presses is constantly growing because of the rather high installed power, rising energy costs and recent activities of legislation and standardization. However, only little research has been done in this direction in the past. Consequently, the systematic technical improvement has been very difficult. This unsatisfying situation has been the motivation for doing research in this field. The article presents exemplarily the activities for a modern single-action test machine with a nominal slide force of 2500 kN. The first step was the experimental analysis of the energy efficiency, followed by simulation studies using a comprehensive model with lumped parameters. Additionally, the simulation was utilized for virtual testing of modified hydraulic drive structures. The final step was the implementation of selected measures for the slide drive and for the die cushion drive in a test machine. After the setup phase, the new drive system worked well and brought a maximum reduction of the machine's consumption of electric energy of about 30 % for the regarded forming tasks and user settings. Apart from this, the experimental results showed that the utilized simulation methodology can predict the real energetic properties with high accuracy.

KEYWORDS: hydraulic deep drawing press, energy efficiency, energy measurement, energy simulation

1. INTRODUCTION

Today, the importance of energy consumption and energy efficiency in industrial production processes is growing. The reasons for this situation are increasing energy costs and new requirements from legislation and standardization [1, 2]. Examples are the European Energy-related Products Directive (2009/125/EC), which defines the framework for the implementation of machine specific energetic requirements, the standard series ISO 14955 “Machine tools – Environmental evaluation of machine tools” and the ISO 50001 “Energy management systems – Requirements with guidance for use”. In comparison with cutting technologies like milling and grinding, forming processes are characterized by a low specific energy demand. Nevertheless, as mentioned by former investigations, especially the forming machines show further potential for energy savings [3, 4].

This situation has been the motivation for starting the joined research project ENEHYT, in which the press manufacturer Schuler Pressen GmbH, the hydraulics specialist Moog GmbH, the automotive supplier MA Automotive Deutschland GmbH and the Institute of Fluid Power (IFD) at TU Dresden worked together. The
consortium extensively analyzed the energy efficiency of modern hydraulic deep drawing presses and experimentally evaluated new, modified drive concepts. The focus lay on machines with a pump controlled slide drive, because these plants are widely used in industrial mass production [5].

The first aim was the systematic experimental analysis of the energetic properties of hydraulic press machines. The second objective was the preparation of methods and recommendations for planning advantageous and energetically well-sized drive systems, which are fully comparable to current press drive systems with regard to dynamics and precision. The third task was the investigation of the dependencies between the user settings and the resulting energy efficiency.

This article concentrates especially on the activities for a hydraulic deep drawing press with a nominal slide force of 2500 kN, which serves as research machine at the TU Dresden. Initially, the press was experimentally analyzed in combination with two different forming tools. Afterwards, a comprehensive simulation model with lumped parameters was developed and utilized to look more into detail and to compare alternative drive structures. At the end, selected new concepts were physically implemented and experimentally tested in the research machine.

2. STATE OF THE TECHNOLOGY

Hydraulic deep drawing presses are equipped with powerful and precisely controllable drive systems. Besides part production, they are often used for the try-out of new forming tools. Figure 5 shows a typical machine structure. The slide drive, which is situated in the head piece, generates the forming motion. Today, variable displacement pumps mostly control the slide speed in modern production presses. Furthermore, single-action machines have a die cushion, which is located below the press table. In deep drawing processes, the cushion controls the material flow by generating an adjustable clamping force, which acts on the flange of the drawn part. The die cushion force and motion is normally controlled by high response proportional valves or pressure control valves. Mostly, there are further hydraulic systems for auxiliary functions, e.g. cooling.

Since the end of the 1980s and the beginning of the 1990s, experimental investigations of the energetic properties of hydraulic presses have been known [3, 6]. Based on self-defined energetic system boundaries, the partial and overall efficiencies were determined for different forming tasks and for variable machine settings. Additionally, simple simulation models were developed. However, because of innovations in the field of hydraulic components and system structures, the applicability of those results to current press drives is limited. For modern machines, there is a lack of systematic and holistic investigations, which focus especially on the energy efficiency.

Former activities in machine design concentrated on the development of highly efficient new drive structures [7, 8]. The main approach was the extended utilization of hydraulic displacement control with variable pumps, because these systems offer minimized throttle losses and a lower cooling effort. Apart from small-suggested energetic losses, in principle energy recuperation is possible through changing the pumps to motor mode. Through recovering the kinetic energy of the slide, the compression energy of the hydraulic oil and the displacement work of the die cushion, a press manufacturer reached energy savings of about 30 % [9]. The energy recuperation in the die cushion drive with variable displacement units was also suggested in [10, 11]. However, one should keep in mind that any process of energy transportation or transformation is accompanied by losses. Consequently, in [11] a recovery rate of only 65 % is exemplarily given.

A newer approach is the utilization of speed variable constant pumps. In comparison with variable pumps running at constant rotational speed, they normally show better energy efficiency, especially in partial load. In [12, 13] a new and modularized hydraulic slide drive with two speed variable internal gear pumps, each connected to one cylinder chamber, is presented. This system allows efficient energy recuperation, for example when the slide moves downwards. In [14] a press manufacturer describes an efficient and integrated concept for slide and die cushion drive, which contains several speed variable pumps. In [15] a research institution presents the results of a scientific analysis of a press with servo-mechanical slide drive
and hydraulic die cushion. The latter was also equipped with speed variable pumps, which reached a maximum efficiency of 68 % in recuperation mode, when the slide moved the die cushion. The utilization of speed variable pumps, which is driven by energy efficiency, is a big trend not only in deep drawing presses, but also in other kinds of forming machines, for instance in press-brakes [16].

All those publications about new drive concepts for presses deliver no information about the dependencies between forming process or machine settings and energy efficiency. Consequently, the realistic energy saving potential under the conditions of industrial utilization remains unknown.

3. MACHINE AND FORMING TOOLS

The presented research activities are based on a modern hydraulic deep drawing press, which has been installed at the IFD recently, see Figure 1. This machine is equipped with a hydraulic slide drive with a nominal force of 2500 kN, in which a unit of constant speed induction motor and variable displacement swashplate pump controls the slide movement. Additional switching valves allow changing the moving direction and help to fulfill safety requirements. The hydraulic die cushion drive with four single cylinders is controlled by high response proportional valves. This drive switches to closed-loop force control, when the slide displaces the die cushion during deep drawing. Otherwise, the system works in closed-loop position control. The installed overall power of the plant is about 73 kW.

![Research machine](image)

The overall energy efficiency of the machine is the quotient of mechanical work, which is conducted into the forming process, and consumption of electric energy for the whole press cycle. Accordingly, this number strongly depends on the force-stroke-characteristics of the process. For being able to analyze that dependency, two different testing tools where selected, which cover a wide range of typical forming tasks for hydraulic deep drawing presses, see Figure 2. The deep drawing tool produces a rectangular tub and the cutting tool performs a circular cut. The blank material is for both the high-tensile steel HCT500X with a thickness of 1 mm.

The process specific forming work \( W_U \) can be obtained by integrating the forming force \( F_U \), which results from plastic deformation and friction between sheet metal part and tool surfaces, over the position \( z_U \). The integration limits are zero and the draw depth \( h \):

\[
W_U = \int_0^h F_U \, dz_U .
\]  

(1)

The drawing tool generates a comparatively high forming work of about \( W_U = 20.1 \) kJ, because the process load acts nearly constantly over the whole draw depth of up to 90 mm. In contrast to this, the cutting tool generates a single force peak over a short stroke of about 2 mm. Thus, the forming work reaches here only \( W_U = 0.12 \) kJ.
4. EXPERIMENTAL ANALYSIS OF THE DRIVE SYSTEM

The initial step of the energetic analysis of the research machine was the definition of appropriate system boundaries. Figure 3 shows the developed scheme. Electric energy from the public net goes into the plant. Motor-pump-units, which represent the generatoric section of the drive systems, perform the conversion to hydraulic energy. The hydraulic circuits are the conductive section, in which the energy is transported and controlled. They consist of valves, blocks, pipes, hoses and further components. Hydraulic cylinders represent the actoric section of the drive systems. They transform the hydraulic energy into mechanical work. Mechanical parts, such as slide, die cushion and other, are the linkage between the drive systems and the forming tool. In experiments it was not possible to distinguish between friction effects at cylinders, which are caused mainly by the sealing systems, and the friction of mechanical guidance elements. Thus, the overall friction losses are assigned to the press mechanics. Losses in form of heat accompany all processes of energy transportation and transformation. Consequently, the outgoing energy, which leaves the machine, consists of useful forming work and a considerable share of waste heat.

![Figure 3. Energetic system boundaries for the research machine](image-url)

The object of the experimental machine analysis was the quantification of the amounts of energy, which are transferred between the subsystems of the press. They are represented by arrows in Figure 3. The power is...
the mathematical product of a potential variable and a flow variable for each physical domain. Hence, one must record these two components separately for each measuring point.

The standard DIN 40110-2 defines a methodology, which allows the determination of the electric power in polyphase systems. The active power $P_\Sigma(t)$ in $n$-wire-systems can be calculated for each point in time $t$ with the phase currents $i_\mu$ and virtual star voltages $u_{\mu0}$ [17]:

$$P_\Sigma(t) = \sum_{\mu=1}^{n} u_{\mu0} \cdot i_\mu. \quad (2)$$

The latter can also be replaced by the voltages measured with respect to a self-defined neutral point. According to the standard, the collective active power $P_\Sigma$, which will be called electric power $P_{el}$ for simplification later, is the arithmetic average:

$$P_\Sigma = P_\Sigma(t). \quad (3)$$

The hydraulic power $P_{hyd}$ is the product if pressure $p$ and flow rate $Q$:

$$P_{hyd} = p \cdot Q. \quad (4)$$

The mechanical power $P_{mech}$ depends on velocity $v$, which is the derivative of the position $z$, and force $F$:

$$P_{mech} = v \cdot F. \quad (5)$$

For each physical domain, the energy $E$ or work $W$ is the integral of the power $P$ over time between the points $t_1$ and $t_2$, which represent for example the beginning and the end of a press cycle:

$$E = \int_{t_1}^{t_2} P \, dt. \quad (6)$$

The efficiency $\eta$ is a characteristic number, which allows the evaluation of systems running in stationary operation. It is the quotient of outgoing useful power $P_{out}$ and incoming power $P_{in}$:

$$\eta = \frac{P_{out}}{P_{in}}. \quad (7)$$

In machine operation, the efficiency may fluctuate very strongly. Hence, it is better to compare the outgoing energy $E_{out}$ and incoming energy $E_{in}$ for complete working cycles instead. This is done by the energy efficiency $\varepsilon$:

$$\varepsilon = \frac{E_{out}}{E_{in}}. \quad (8)$$

The energy efficiency depends on the specific machine load $x$. The latter was defined here as follows:

$$x = \frac{W_U}{W_{nom}}. \quad (9)$$

$W_{nom}$ is the nominal work that the slide drive can apply, it depends on the nominal slide stroke $h$ and nominal slide force $F_{nom}$:

$$W_{nom} = \frac{h \cdot F_{nom}}{2}. \quad (10)$$

The divisor 2 results from the fact that deep drawing processes can only use maximum half of the available slide stroke. Otherwise, the removal of the drawn part from the tool would be impossible.

Based on the above-mentioned methodology experimental investigations were carried out, which covered all operational states of importance for deep drawing presses: standby, idle, tool changing and production. Furthermore, the machine settings were varied to determine their influence on the energy efficiency. For
deep drawing and cutting every single experiment was repeated at least ten times to allow a statistical analysis. A measuring system with sensors for voltage, current, pressure, volume flow, position, force and temperature was installed at the press. A PC with a data acquisition card, which has a maximum sampling rate of 250 kHz, a resolution of 16 bits and 64 single channels for analogue voltages -10...+10 V, recorded all sensor signals synchronously. The chosen sampling rate setting was between 1 and 3 kHz. The power and energy calculation took place after the completion of the experiments.

Figure 4 shows measuring results from the research machine for three different operational states. For deep drawing and cutting the press ran in single stroke mode, the slide velocity was set to the maximum value and the overall slide stroke was 160 mm respectively 70 mm. The drive system had been warmed up before to reach constant thermal boundary conditions.

![Figure 4. Experimental results for the research machine](image)

The picture shows the temperature of the oil tank, the overall cycle time, the cumulated electric energy consumption of the machine, of its subsystems and of the different operation phases, the mechanical work of slide, die cushion and forming process and the distribution of forming work and heat losses. The machine’s energy efficiency for deep drawing is 9.3 %. For cutting, the value reaches only 0.27 %. This is mainly due to the difference in specific machine load. In these numbers, the influence of the forming task becomes obvious. Like in this typical example, the specific load of hydraulic deep drawing presses is usually low. They are mostly oversized in order to fulfill various and increasing process demands during a very long lifetime of several decades. This affects the energy efficiency negatively. The research machine’s energy consumption is for deep drawing 2.6 times higher than for idle state, when the cycle time of the latter was scaled to comparable length. A similar relation between these operational states was also determined for other presses [5]. In general, the losses of the generatoric section of the slide drive are comparatively high.

5. SIMULATION-BASED ANALYSIS OF THE DRIVE SYSTEM

Because of the high complexity of hydraulic press drive systems, the experimental machine analysis is limited to the level of subsystems. For a detailed investigation of component losses a comprehensive simulation model with lumped parameters was developed. The model contains the simplified machine and
drive structure, which is shown in Figure 5. Subsystems of importance for the energy efficiency are the press mechanics, the slide drive, the auxiliary circuit, the die cushion drive, the cutting damper, the cooling circuit and the forming process. The latter is represented by simplified process load models based on characteristic lines. A PLC controls all the drives. Hence, the model contains its basic functionality.

The modelling procedure started with the analysis of the real machine and its documentation. At first, submodels for all functional units of relevance were developed, which include the static, dynamic and energetic properties. The model parameters were determined from measurements, data sheets and theoretical calculations. Then, the submodels were combined in the machine model. The latter was implemented in system simulation software, which provides predefined component model structures for several physical domains [18].

A model check based on experimental data took place for different operational states and machine settings, e.g. velocity, draw depth, thermal machine conditions. Figure 6 exemplarily shows the comparison between model and real machine behavior for a deep drawing process. The picture proves that the model’s prediction quality for the energetic behavior is sufficient. Higher deviations only occur for the power losses of electric motor and hydraulic pump in the cooling circuit. This is due to uncertain or missing information from the component manufacturers. For a whole press cycle, which consists of fast motion down, pressing, force relief and fast motion up, the model calculates the machine’s overall electric energy consumption 6 % smaller than measurable in reality. The model deviation for the forming work is -2 %.

The model was utilized for an extensive study of component losses, regarding the whole machine in both selected production cycles. Based on this, the energetically most important or most inefficient components were identified. Afterwards, several structural modifications of the slide drive and die cushion drive were developed, virtually implemented and compared with each other. The result of the simulation study was that high energy savings are achievable with alternative and more efficient hydraulic systems. The model predicted a maximum energy saving potential of about 30 % for the whole machine in deep drawing and cutting processes as well as in idle state.
6. EXPERIMENTAL EVALUATION OF A MODIFIED DRIVE STRUCTURE

To proof the calculated energy saving potential in this scientific study with experimental data, selected measures were implemented into the research press. Figure 7 shows the chosen structure. The original pump-controlled slide drive was equipped with a frequency converter, which allows lowering the rotational speed of the motor-pump-unit. In idle state the losses of motor and pump can be reduced significantly this way. Additionally, the set values for the valves in the slide drive were optimized. The valve-controlled die cushion drive was replaced by a pump-controlled system to reduce the throttle losses. It has four brushless servomotors, which are designed to reach maximum dynamics, and radial piston pumps with a constant displacement volume. Frequency converters are responsible for the closed-loop control of the motor speed. The die cushion and the slide drive have a common intermediate circuit. Hence, the electric energy, which is recuperated from the die cushion during deep drawing, is directly fed in the slide drive and does not leave the machine. Due to the utilization of differential cylinders in the die cushion drive, the system’s oil volume depends on the position of the piston rods. Thus, a low-pressure hydraulic accumulator is necessary. After the modification, the research machine was undergone first tests to figure out the static and dynamic
properties. The main point of interest was the die cushion drive, because user requirements for this system are very strict. The new pump-controlled structure completely fulfilled the demands, regarding the quality of closed-loop force and position control. The drive reached a good dynamic behavior, which is comparable to valve-controlled systems.

Figure 7. Modified structure of the research machine (simplified)

In the next step, the energy efficiency of the modified machine was experimentally analyzed for the same operational states, in which the original drive system had already been tested before. Figure 8 exemplarily compares the energetic properties of both drive structures for the deep drawing cycle.

Figure 8. Experimental results for the research machine with drawing tool before and after modification (single stroke mode, maximum velocity, slide stroke 160 mm, warm)

The overall saving of electric energy with the modified drive system is about 29 %, which is really close to the simulation results. The highest reduction occurs during the fast motion up, because the die cushion can now

$$
M \approx 4x \text{Phyd} \div Pel
$$

$$
F_{Z} \approx F_{S} \text{ and } F_{U} \text{ during fast motion}
$$

$$
W_{\text{forming process}} = \sum Q_{\text{main drives}} + Q_{\text{cooling circuit}} + Q_{\text{mechanics}} + Q_{\text{other}}
$$

$$
W_{\text{original system}} = W_{\text{modified system}} + \Delta W_{\text{energy}}
$$
be move without throttle losses. The energy recuperation in the die cushion drive during press mode saves also energy. The forming work shows a difference of about 3.6 kJ when comparing the measured values of the original and modified system. This is due to slightly varying material properties of the utilized blanks and lubrication conditions between blank and forming tool. However, the forming work is comparatively small compared to the other types of energy in the drive system. Thus, this effect is tolerated here. For future investigations regarding the energy efficiency of press drives, an improvement could be reached for instance by substituting the real forming tool with a hydraulic load system, which can apply dynamic load forces with high accuracy. The distribution chart at the right hand side shows, that the losses in the conductive section of the die cushion drive were reduced significantly. In contrast to this, the share of other consumers grew from a negligible value for the original structure to 6 % for the modified drive system. The reason for this is the increased number of electric and electronic components.

Figure 9 displays the energetic properties of both drive systems for the cutting process. The overall energy saving is about 22 %. This is primarily the result of optimizing the set values for the valves in the slide drive, which leads to reduced losses in the conductive section of this drive during force relief. For cutting the die cushion is not in use, but in the original system the motor-pump-unit was permanently running. The modified system allows completely switching off the drive, which enables further savings. But the new system has also some drawbacks. As already mentioned, the energy consumption of the electric and electronic devices, which form the category “other”, is significantly higher. The new frequency converter for the slide motor causes additional losses in the generatoric section of the slide drive. Furthermore, the switching between idle motor speed (800 rpm) and operational speed (1500 rpm) at the beginning and at the end of the regarded press cycle is accompanied by losses. Hence, during fast motion down, press mode and fast motion up the modified drive system shows all in all no remarkable energy saving potential here.

Figure 9. Experimental results for the research machine with cutting tool before and after modification (single stroke mode, maximum velocity, slide stroke 70 mm, warm)

The modified press drive system was also experimentally analyzed in idle state. It reaches an overall energy saving of about 32 % in comparison with the original system. Here, the most effective measure is the reduction of the rotational speed of the slide motor.
7. SUMMARY

Over the last years, the importance of the energy efficiency in industrial production processes has been constantly growing. For hydraulic deep drawing presses the knowledge from former scientific investigations is not sufficient. Consequently, a systematic technical improvement has been hardly possible. This situation was the motivation for analyzing modern press drive systems especially with regard to their energetic properties and occurring losses.

An experimental machine analysis under the conditions of industrial utilization delivered important information about the status quo. Subsequently, a simulation model with lumped parameters was developed. It enabled a simulation study, in which the distribution of energy losses was determined in detail. Besides, the model served for a virtual test of modified hydraulic drive structures. Finally, selected modifications were practically implemented into the research machine and underwent experimental investigations.

All in all, the results exemplarily show that powerful hydraulic press drives offer a quite large potential for energy savings. With the modified hydraulic drive system, the energy consumption of the analyzed research machine was significantly reduced. Nevertheless, there is a lot of space for further improvements, which becomes obvious in the relation between the remaining heat losses and the useful energy output.

The presented research work delivered important knowledge and impulses for being able to systematically improve the energy efficiency of hydraulic deep drawing presses concerning machine design and operational conditions. This is an important contribute to save natural resources and the environment. Apart from this, the results help to maintain and to improve the competitiveness of hydraulic drive technology.

8. ACKNOWLEDGEMENTS

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NOMENCLATURE

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ANALYSIS OF A HYDROSTATIC TRANSMISSION SYSTEM FOR HORIZONTAL AXIS WIND TURBINES

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ABSTRACT
The paper presents a conception and analysis of a hydrostatic transmission for a horizontal axis wind turbine. This alternative design, when compared to either a gearbox or direct transmission, presents a promising solution for the transfer of the extracted power from the wind to the electric generator and for the regulation of the rotor speed. It also results on the decoupling of the rotor speed from the generator speed. The variation of the rotor speed allows for the maximization of the wind energy capture. On the other hand, it keeps the generator operating with synchronous speed in relation to the electrical grid. In this context, a dynamic model in AMESim of the hydrostatic transmission was developed, whose purpose is to evaluate both the hydraulic circuit and controller. The control task is carried out by one of two classic PID controllers, switching according to wind and rotor speed, whose output determines the motor volumetric displacement. The model also represents the interaction between the wind and rotor as well as the connection of the generator with the electrical grid. Considering a 150 kW wind turbine, the system response is studied for wind speed step inputs and for turbulent wind profiles. The input and output power and the output electric frequency are analyzed as well as the rotor and generator torques. The adopted solution for the transmission and control system proved to be effective regarding the proposed objectives.

KEYWORDS: hydrostatic transmission, control strategy, wind turbine

1. INTRODUCTION
Wind power stands out as one of the most promising sources for the generation of electricity. Wind turbines may generate electricity close to the final consumer, reducing the need for high voltage transmission lines. Being highly modular, wind farm capacity can be gradually enlarged to keep up with consumer demand [1]. To these qualities, it is also worth noting to the compatibility between the energy production by wind farms and the use of land for agricultural and livestock related industries [2].

Due to the interest related to the development of wind power technology, a great attention has been given to the aspects related to the overall performance of wind turbines. In this sense, the control of horizontal axis wind turbines can be intended to maximize the wind energy capture, to reduce dynamic loads at the turbine structure and to maintain generated power quality according to the standard grid requirements [3].

An aspect that increases the complexity of the wind turbine and, consequently, the cost of generated power is the transmission system, responsible for transmitting the mechanical power extracted by the turbine rotor to the electric generator. This power transmission is usually accomplished by direct coupling between the
rotor and the generator or by the use of a mechanical gearbox connected to the rotor and to the generator, which steps up the rotor speed to higher speeds more appropriate for the generator.

Whichever transmission system is used, the rotational frequency of the generator will be equal or proportional to that of the rotor. Asynchronous, or induction, generators require reactive power from the grid [4]. According to [5], synchronous generators offer more efficient and robust operation than induction generators, and the use of a permanent magnet synchronous generator (PMSG) eliminates the need for field excitation. The power grid requires the generators to run at synchronous rotational speed in order to attend to the standard grid requirements.

Modern wind turbines make use of power electronics (frequency converters) to decouple the generator rotational speed from the grid frequency. Thus, the generator may vary its rotational speed to best adapt itself to the rotor speed. However, the use of power electronics affects the overall efficiency of the system. According to [6], the maximum efficiency of an electric generator operating in conjunction with a frequency converter is in the range of 94% to 95%. It is also acknowledged that the use of a mechanical gearbox to step up the rotor speed greatly increases the weight, cost, and complexity, and decreases the reliability of the turbine [7].

Considering the issues described above, the adoption of a hydrostatic transmission integrated into a wind turbine is being considered a promising alternative regarding the development of these machines. It can merge the functions of a power transmission system, gearbox and frequency converter. This can decouple the rotor rotational speed from the generator rotational speed. The hydrostatic transmission can be responsible for regulating the rotor speed, allowing it to operate with maximum aerodynamic efficiency while maintaining constant generator speed [5]. It would also permit installing the majority of the turbine components at ground level. This reduces the weight of the tower and the nacelle of the wind turbine, which in turn reduces the loads on the tower [8], along with manufacturing and maintenance costs.

This topic is being studied by several authors. For example, in [9] the authors generate different concepts for the transmission system. Some of them are different arrangements between pumps and motors, while other solutions combine mechanical components with hydraulics. [5] studies a hydrostatic transmission control method that acts upon rotor speed to achieve a high power coefficient. As discussed in [10], hydrostatic transmissions with digital components are being studied and might be an interesting alternative. Different control solutions that have a focus on the transition between operational regimes have been studied in [11] and [12].

In this paper a hydrostatic transmission for 150 kW wind turbine is studied. The aim was to use off-the-shelf components that result in a simple, yet reliable, configuration. These mid-size wind turbines are ideal for distributed power generation and local energy consumption. The hydrostatic transmission comprises two PID controllers that are switched between operational regimes for best wind turbine performance. Details about the system design are presented and the system is modelled using LMS AMESim. The theoretical results show the model adequacy and allow identifying the benefits and drawbacks of the use of hydrostatic transmission in mid-size wind turbines.

2. WIND TURBINE DESIGN

2.1. WIND TURBINE REQUIREMENTS

The rotor speed must change in accordance with the variations of wind speed in order to achieve maximum aerodynamic efficiency and, thereby, to extract maximum power from the wind. The power coefficient \(C_p\) determines the power extractable from the wind, [3], such that

\[
P_R = C_p \cdot P_v = \frac{1}{2} \cdot \rho_{air} \cdot A_R \cdot C_p \cdot v^3
\]  

(1)
where $P_R$ is the mechanical power extracted by the rotor [W], $P_v$ is the power present in the wind stream [W], $\rho_{air}$ is the air density [kg/m$^3$], $A_R$ is the rotor swept area [m$^2$], and $v$ the horizontal wind speed [m/s].

The power coefficient is dependent on the tip speed ratio ($\lambda$), which is the ratio of the tangential velocity of the tip of the rotor blades to the horizontal wind velocity, i.e.:

$$\lambda = \frac{\omega_R \cdot R_R}{v}$$

where $\omega_R$ is the rotor angular velocity [rad/s] and $R_R$ the rotor radius [m].

The power coefficient value depends on the tip speed ratio as shown in Figure 1. One can observe that the power coefficient reaches its maximum value $C_{p,max}$ for an ideal value of the tip speed ratio $\lambda_{ideal}$ [13]. For any pitch angle ($\beta$), there is an optimum operating point. This means that for a higher power extraction from the wind stream, it is necessary for the rotor to operate with $\lambda_{ideal}$, in order to reach $C_{p,max}$.

![Figure 1. Power coefficient vs. tip speed ratio performance curve for a given $\beta$ (adapted from [3]).](image)

The basic design requirements taken into account in this paper refer to a horizontal axis variable speed wind turbine of 150 kW rated power. It is initially established that this turbine will be connected to an electrical grid that operates with a frequency of 60 Hz. The electric generator selected is a synchronous generator that incorporates two pairs of rotor poles, which results in a synchronous speed of 1800 rev/min (30 1/s). The design of the rotor blades selected for the wind turbine allows for a maximum power coefficient ($C_{p,max}$) of 0.496, that is achieved for a tip speed ratio of 8.4.

The diameter of the rotor is established according to

$$P_{nom} = \frac{1}{2} \cdot \rho_{air} \cdot A_R \cdot \eta_A \cdot C_{p,max} \cdot v^3.$$  (3)

Taking into account a rated power generation ($P_{nom}$) of 150 kW, air density at sea level of 1.225 kg/m$^3$, maximum power coefficient of 0.496, nominal wind speed of 12 m/s, and an overall wind turbine efficiency ($\eta_A$) estimated as 0.76 (considering inefficiencies in the hydraulic pump and motor, electric generator and power transformer), results in a rotor swept area of approximately $A_R = 380 m^2$. The corresponding rotor diameter is $d_R = 22 m$.

Due to aspects associated with wind speed reduction and turbulence close to ground level [3], it was established a tower height of 30 m for this wind turbine. Utilizing data presented in [14], it was estimated an axial moment of inertia for the rotor of $J = 50000 kgm^2$. The wind turbine parameters are summarized in Table 1.
Table 1. Wind turbine parameters.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated generated power</td>
<td>150 kW</td>
</tr>
<tr>
<td>Number of rotor blades</td>
<td>3</td>
</tr>
<tr>
<td>Rotor diameter</td>
<td>22 m</td>
</tr>
<tr>
<td>Rotor axial moment of inertia</td>
<td>50000 kgm²</td>
</tr>
<tr>
<td>Tower height</td>
<td>30 m</td>
</tr>
<tr>
<td>Maximum power coefficient</td>
<td>0.496</td>
</tr>
<tr>
<td>Control mechanism</td>
<td>Hydrostatic transmission</td>
</tr>
<tr>
<td>Rotor speed</td>
<td>Variable</td>
</tr>
<tr>
<td>Generator model</td>
<td>Synchronous generator</td>
</tr>
</tbody>
</table>

2.2. HYDROSTATIC TRANSMISSION DESIGN

An adequate approach for the design of the hydrostatic transmission consists in using a radial piston pump coupled to the rotor and an axial piston motor to the generator [15]. Radial piston pumps have a greater performance at lower actuation speeds, which is common for wind turbines, while axial piston motors have greater efficiency at higher actuation speeds. Advantages of using a hydrostatic transmission comprising only one pump and one motor are the low number of components and the readiness to control the system [9].

The system design discussed in this paper comprises only one radial piston constant displacement hydraulic pump and one axial piston variable displacement hydraulic motor. The selected operational pressure, 25 MPa, allows for a high power density.

The control system uses two PID controllers that are switched according to the rotor and wind speed values. The control method for operation with maximum wind power extraction \( C_{p,\text{max}} \) is based in [5]. However, when the rotor speed reaches the rated value, the control system switches to a second control strategy that consists in maintaining constant rotor speed [16]. The system control with a hydrostatic transmission at both regions is studied in [17] using a gain scheduled linear quadratic regulator.

2.3. HYDROSTATIC TRANSMISSION CONFIGURATION

The main circuit of the hydrostatic transmission comprises the hydraulic pump and motor, two pressure lines, two relief valves, and a hydraulic filter. Because the turbine rotor can revolve in only one direction, due to the aerodynamic characteristics of the rotor blades, only one of the pressure lines of the main hydraulic circuit will be the high pressure line. The pressure relief valve located in the high pressure line serves to protect that line from pressure spikes due to wind or load dynamics. The pressure relief valve on the low pressure side acts as a safety valve.

A charge circuit is incorporated, whose main purpose is to replace fluid leakages from the main circuit and to provide the main circuit with pressurized hydraulic fluid in order to prevent cavitation. Figure 2 presents the diagram of the hydrostatic transmission configuration under analysis [16].

The control signals are also presented in Fig. 2, where \( U_{S1} \) is the signal from the pressure transducer, \( U_{S2} \) is the signal from the tachogenerator and \( U_{S3} \) is the signal from an anemometer that is installed on the outside of the nacelle. \( U_{Z1} \) is an offset signal, set to 80 rev/min (1.33 1/s) (the rated rotational speed of the rotor) and \( U_{V1} \) is the control signal. Table 2 presents the components shown in Fig. 2.
Figure 2. Hydrostatic transmission diagram.

Table 2. Main components of the hydrostatic transmission.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Component</th>
</tr>
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<tbody>
<tr>
<td>0P1, 1P1</td>
<td>Hydraulic pumps</td>
</tr>
<tr>
<td>0S1</td>
<td>Tachogenerator</td>
</tr>
<tr>
<td>0F1, 1F1</td>
<td>Hydraulic filters</td>
</tr>
<tr>
<td>0V1, 0V2, 1V3</td>
<td>Relief valves</td>
</tr>
<tr>
<td>0S2</td>
<td>Pressure transducer</td>
</tr>
<tr>
<td>0A1</td>
<td>Hydraulic motor</td>
</tr>
<tr>
<td>1M1</td>
<td>Electric motor</td>
</tr>
<tr>
<td>1V1, 1V2</td>
<td>Check valves</td>
</tr>
<tr>
<td>R</td>
<td>Reservoir</td>
</tr>
<tr>
<td>Z1</td>
<td>Controller</td>
</tr>
</tbody>
</table>

2.4. CONTROL STRATEGY

Figure 3 illustrates the classical strategy for the operation of variable speed wind turbines, as described in [13], where the three operational regions are presented. For region I, located between the cut-in wind speed ($v_{min}$) and the wind speed for rated rotor speed ($v_{o,nom}$), the objective is to extract the maximum power from the wind stream. This strategy is called Maximum Power Tracking. This is done by maintaining the power coefficient at its maximum value ($C_{p,max}$). To accomplish this, the rotor rotational speed ($\omega_R$) must be continuously altered in accordance with the variations of wind speed in order to operate with an ideal tip speed ratio ($\lambda_{ideal}$).
The wind turbine operates in region II for incoming wind speeds between the $v_{\omega, nom}$ and the nominal wind speed ($v_{nom}$), where the rotor speed is limited to its nominal value ($\omega_{R, nom}$) of 80 rev/min (1.33 1/s), in order to preserve the structural integrity of the wind turbine and to maintain noise levels within admissible limits [6]. If the wind speed reaches $v_{nom}$, the wind turbine will operate in region III, where the extracted power from the wind will be maintained constant by the wind turbine since it will have reached its rated power generation capacity. This is accomplished by pitch or stall control mechanisms present in the wind turbine. Beyond the cut-off wind speed ($v_{max}$), the turbine is shut off to prevent damage. The objective of this study is to verify the capacity of the hydrostatic transmission to control the wind turbine during operation in regions I and II. The generated power versus rotor rotational speed in the three operational regions is shown in Figure 4.

It is known that the mechanical power extracted from the wind ($P_R$) is expressed by

$$P_R = T_R \cdot \omega_R,$$  \hspace{1cm} (4)$$

where $T_R$ is the mechanical torque upon the rotor [Nm].

On the other hand, the rotor torque required to maintain maximum aerodynamic efficiency is a function of generator torque, transmission ratio, and efficiency of the transmission system. As presented in [5], for the operation in region I, the combination of Eq. (4) with equations (1) and (2) gives the ideal torque ($T_{R, ideal}$) upon the rotor as a function of the rotor speed according to

$$T_{R, ideal} = \frac{1}{2} \cdot \rho_{air} \cdot A_R \cdot R^3 \cdot C_{p, max} \cdot (\omega_R)^2 \frac{\lambda_{ideal}^3}{2}$$  \hspace{1cm} (5)$$

Considering that the focus of this study is to use a hydrostatic transmission for the control of the wind turbine and, consequently, eliminating the use of frequency converter between generator and grid, the ideal rotor torque determined by Eq. (5) can be correlated to the required ideal pressure ($p_{ideal}$) upon the hydraulic pump by
\[ T_{R,\text{ideal}} = \frac{p_{\text{ideal}} \cdot D_p}{\eta_{p,m}} \] (6)

where \(D_p\) the pump displacement \([m^3/\text{rad}]\) and \(\eta_{p,m}\) the pump mechanical efficiency.

Furthermore, the generator torque \((T_g)\) is related to the system pressure by the volumetric displacement of the hydraulic motor, as presented by

\[ p_{\text{ideal}} = \frac{T_g}{D_M \cdot \eta_{M,m}} \] (7)

where \(D_M\) the motor displacement \([m^3/\text{rad}]\) and \(\eta_{M,m}\) the motor mechanical efficiency.

When synchronized with an infinite bus bar the synchronous generator rotor frequency is kept constant and equal to the grid frequency [19]. Since the generator speed is constant the electric power delivered to the grid is just dependent of the generator torque. This torque, as show in Eq. 7, is a function of the system pressure. In this way it is possible to adjust the power output by controlling the system pressure. Details regarding the synchronization of the generator with the infinite bus bar are out of the scope of this paper.

In this way, the pressure in the hydrostatic transmission can be controlled by the variation in the motor volumetric displacement, which results in control of the torque upon the rotor. By combining equations (5) and (6), it is possible to determine the ideal system pressure as

\[ p_{\text{ideal}} = \frac{1}{2} \cdot \rho_{\text{air}} \cdot A_R \cdot R_R^3 \cdot C_{p,max} \cdot \eta_{p,m} \cdot (\omega_R)^2 \cdot D_p \cdot \lambda_{\text{ideal}}^3 \] (8)

The operation in region I is governed by Controller 1 shown in Figure 5. While the system pressure is measured by a pressure transducer and sent to the Controller 1 as signal \(U_{S1}\), the ideal system pressure is calculated using the measured rotor speed sent to the controller as signal \(U_{S2}\). The pressure error leads the controller to act upon it by sending a command signal \(U_{V1}\) to the valve responsible for controlling the displacement of the motor that leads to operation with ideal rotor torque.

In case the incoming wind speed reaches \(v_{\omega,\text{nom}} = 10.7\) m/s, the rotor will achieve its rated rotational speed of 80 rev/min \((1.33\ 1/s)\). When this condition is reached, reference signals \(U_{Z1}\) and \(U_{S3}\) switch for the Controller 2 to command operation in region II. The controller acts upon the rotor speed error in order to maintain the rated speed. This constant rotor speed can be maintained by the hydrostatic transmission until the wind speed reaches 12 m/s. Above this incoming wind speed, the wind turbine must rely on pitch or stall control mechanisms. The control of the rotor speed by blade pitching or stall is beyond the scope of the present analysis. Both controllers are classic PID controllers.
3. SIMULATION AND RESULTS

For this analysis, some simplifications were made regarding the interaction of aerodynamic phenomena and the turbine rotor. However, a detailed modelling of the hydraulic system was implemented in AMESim considering the circuit presented in Figure 2 and the block diagram shown in Figure 5.

The specific conditions that are analyzed here are the ability of the system to:

- Maintain the frequency value for generated power in 60 Hz for different operational conditions;
- Maintain the wind turbine operating with maximum power coefficient for wind speeds below 10.7 m/s by controlling the rotor speed, in order to maximize wind energy extraction;
- Prevent rotor speed from exceeding 80 rev/min (1.33 1/s) when operating in region II;

3.1. MODEL AND CONTROL SYSTEM VALIDATION

Initially it is verified if the power extracted by the wind turbine from the wind stream is according to what is expected for rated wind speed value ($v_{\text{nom}}$). A step input is given for the wind speed and presented in Fig. 6 (left). Figure 6 (right) displays the power provided to the electric grid, showing a value of 157.7 kW. This is higher than the expected 150 kW because generator and electric power transformer inefficiencies were not considered in the model.

![Figure 6. Wind speed step input (left). Power extracted by the rotor and power delivered by the wind turbine (right).](image_url)

Figure 7 shows the ability of the system to switch between operational regimes. When the wind speed is above 10.7 m/s and the rotor speed reaches 80 rev/min (1.33 1/s), the second controller takes over and maintains the wind turbine operating at the rotor rated speed. Figure 7 also shows three frequency spikes for the generated electric power. The first and third spikes are due to the step inputs. However, the second spike is due to the second controller taking over the control of the system and limiting the rotor speed. After the wind speed slows down, the first controller resumes command and the wind turbine once again operates with maximum wind energy extraction.

3.2. SIMULATION WITH TURBULENT WIND PROFILES

Simulations with realistic wind conditions were carried out, which are characterized as a high frequency phenomenon with a stochastic behavior. The wind speed varied from 5 m/s to 10.2 m/s, as presented in Fig. 8 (left), so the wind turbine operates according to the Maximum Power Tracking strategy. At 35 s and again at 80 s the wind speed pattern resembles a ramp input. The rotor speed is shown in Fig. 8 (left). It can be seen that due to the great rotor inertia, wind speed disturbances are “smoothed” when interacting with the system. A very important result due to the presence of a hydrostatic transmission is presented in Fig. 8 (right), where it can be observed that due to the superior damping characteristics of the hydrostatic...
transmission, even for a turbulent wind profile, the generated power frequency output is very stable around its rated value, reaching a maximum value of only 60.04 Hz.

![Figure 7. Wind speed input and rotor speed (left) and generated frequency (right).](image1)

The variation of the incoming wind speed results in a variation of the rotor torque, as seen in Fig. 9 (left). In conventional wind turbines, these torque variations are transmitted by the mechanical coupling to the electric generator, which will transfer these variations to the grid as electrical power variations unless they are absorbed by the power electronics. The hydrostatic transmission proved able to decouple the rotor torque from the torque acting upon the generator, resulting in a smooth torque curve acting upon the generator.

![Figure 9. Torque upon the rotor (left). Torque upon the generator (right).](image2)

Fig. 10 (left) shows the ideal pressure, as calculated by Eq. 8, and the system pressure measured in the high-pressure line. It is possible to observe that the pressure presents a smoothed behavior, similar to the rotor speed presented in Fig. 8 (left). Fig. 10 (right) presents the error between the measured and the ideal pressure. It can be observed that the pressure error is close to zero, confirming that the system can be controlled in a very precise way.
Fig. 11 (left) shows the power extracted by the rotor and the power delivered by the wind turbine. As with the torque, the generator power curve is also smoothed due to the hydrostatic transmission damping characteristics. Fig. 11 (right) presents the power coefficient, demonstrating that the system is able to maintain it close to its maximum value of 0.496, even for turbulent wind profiles.

Figure 11. Relation between the extracted power from the rotor and the power delivered by the wind turbine (left). Power Coefficient (right).

4. CONCLUSION

Initially it was verified that the power extracted by the wind turbine from the wind stream was according to what was expected for the case of a rated wind speed \(v_{nom}\) of 12 m/s, and that the system was able to switch controllers for different operational regimes. After verifying that the model adequately represented the wind turbine, simulations with turbulent wind speed profiles were carried out. These simulations showed that the hydrostatic transmission was capable of adequately performing the tasks of a power transmission system, while simultaneously controlling the rotor speed. The control of the rotor speed maintained the wind turbine operating close to maximum efficiency, by maintaining the power coefficient always close to its maximum value of 0.496.

Although the wind turbine efficiency was reasonably high, it might not be enough to compensate for the replacement of traditional mechanical transmission systems. However, the generated power frequency output to the grid was very stable around its nominal value, showing that it might be possible to replace the frequency converters and other power electronic systems in wind turbines that comprise a hydrostatic transmission. Finally, it was also shown that a PID controller is capable of performing well in the task of system control for the proposed conditions.

As a continuation of this project, a test stand is being developed in order to allow for testing of the system, evaluation of alternative hydrostatic transmission configurations, and different control methods.
REFERENCES


NEW ISO25119 COMPLIANT 6-INDIPENDENT WHEELS ELECTRO-HYDRAULIC STEERING SYSTEM FOR AGRICULTURAL MACHINE

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ABSTRACT

Performance requests and machine automation, in conjunction with new regulations for agricultural, earthmoving and construction machines, represent the today’s most difficult challenge for machine designers and researchers. Machine control systems complexity and safety regulation compliance are probably the most complex problems to be solved in new machines design.

The paper describes the design of a steering system, installed in on a big 6 wheels agricultural self-propelled machine, that must comply with new regulations in terms of functional safety.

The original vehicle steering system design should be driven by an electro-hydraulic system controlling the rear wheels, totally controlled using by wire electronics; this architecture requires a too high functional safety performance level. In fact the safety analysis of the system lead to a performance level required whose compliance is a too challenging task, due to the required quality of the software and to the cost a fully redundant hardware, on both electronic and hydraulic side.

A thorough analysis on steering function and system characteristics was performed, in order to reduce the overall risk of the steering function, considering that a pure hydraulic solution is not possible due to the leakages in the steering systems, that can alter the parallelism of corresponding wheels in the two sides of the machine.

In the initial design, the most critical feature was found in the entirely electronic control system of the rear wheels. A complete revision of the functionality, a deeper analysis of working conditions and a new approach to the steering function allowed a new design to be conceived, that fits the function requirement but requests a Performance Level "b" only, due to the basic function change from “rear steering system” to “rear steering error correction system”.

The paper explains the solution, the hazard analysis results and the approach to “risk reduction by electro-hydraulic design” that allowed a better solution, offering the same functionality thanks to a minimal risk approach.

KEYWORDS: Functional Safety, electro-hydraulic systems, multiprocessor control unit, fail operational system
1. INTRODUCTION

In the world of heavy-duty and agricultural machines, complex electronic systems have been introduced, in order to supersede complex and automated operations. Moreover in the field of precision agriculture, the human intervention is considered the last one in terms of precision and is totally overtaken by machine automation under the ISOBUS (ISO11783) protocol.

![Figure 1. The complete Talpa Machine view with the six wheels traction system](image)

The growing request for performance, the continuous requests for pesticide reduction and the need to reduce CO₂ and, in general, greenhouse gases emissions in the atmosphere, many new and very specialized machines are continuously designed and produced. In the agricultural field, due to the very different soil and plant characteristics, is very difficult to design a general purpose machine, except for the tractor. Many self-propelled agricultural machines are designed and produced, but in very few cases the production numbers are high, more often a very successful machine is produced in many versions, each one fitting with some special task requirements for soil or plants. But all the machine models and versions must comply with all regulations, especially the ones applied in the European Union, despite the small production numbers. The functional safety regulations appear a challenging task, because of the regulation requirements are not only related to products but also to methods for design and testing.

This work presents, through an example, a possible method to reduce the global safety requirements through design applied to a steering system.

The machine under study in this paper is a special agricultural machine (Figure 1) that should comply high Performance Levels Required for functional safety (AgPLr) of the steering system, because of its total electronic control; a requirement analysis pointed out that the resulting cost of the solution technically available was not compatible with the market reference price and management requests. The paper demonstrates that a proper electro-hydraulic design helped to reduce the global Performance Level required and made the solution cost effective. In fact the design of the rear steer system was totally revised, moving from a totally electronic solution to a hydraulic solution with only an electronic correction; the solution allowed a substantial revision of the Unit of Observation, changed from “rear electro-hydraulic steering system” to “rear electro-hydraulic steering correction system”; with a very limited dynamic and authority and with a very limited working hours, considering the machine task and mission. As it will be explained in the safety analysis, the limited dynamic and authority of the new steering correction system, lead to an easier controllability of the steering faults and a consequent lower AgPLr, because of the substantial change of function.
2. FUNCTIONAL SAFETY STANDARDS

In recent years new regulations (from [1] to [6]) have been enforced to agricultural, earthmoving and construction machines. Type C standards apply to the field of functional safety, involving both electronic, mechanics and hydraulic design of machine functions and systems.

The mandatory standards are in a complex relationship and, in general, starting from a general Directive (e.g. Machine Directive 2006/42/EC or Agricultural Tractors Directive 2003/37/EC), the compliance shall be granted following general standards (Type A standards), and more focused standards (Type B, as for example the ISO 13849 Standard) for class of machine types, and very specialized standards (Type C) that are dedicated to a single type of machines.

The last standards generated in the field of machinery are the ones for Agricultural Machines, both Tractors and self-propelled or towed/mounted, generally called implements. Being the last generated these standards are the most complete and complex, and were synthetized also referring the automotive functional safety standard ISO 26262, that was conceived with the focus on a market, where the mass production is considerably higher than in the agricultural machines field.

The new regulations are critical, especially for applications where a safe state cannot be identified. The new standard ISO 4254 for agricultural self-propelled machines is now mandatory for new machine design and it requires ISO25119 compliance. It’s common opinion that the most complex systems to be designed, in compliance with the safety regulations, are steer by wire and brake by wire systems. Steer and brake systems share the not negligible requirement that the main function can’t be lost in case of fault, because a safe state cannot be identified in case of fault occurrence for both systems; as a consequence, electronically controlled steer or brake must provide a fail operational characteristic even in faulty circumstances.

Following more critical applications, like avionic control systems, or applications where electronic systems were applied before in respect agricultural, earthmoving and construction machines, the ISO standardization committee TC023 and TC127 synthetized the respective functional safety standards, that present some difference in functional safety approach and analysis, but that are very similar in the risk evaluation flow. Both are based on a risk analysis that, for the Agricultural machines, is shown in the Figure 2.

![Figure 2. ISO 25119-2 Hazard Analysis and resulting System Required Performance Level Table](image)

The schema presented in Figure 2, is divided in two areas: the left one is the basis of the hazard analysis, while the right one is the possible mix of system characteristics that can be applied in order to comply the required Performance Level. The hazard analysis lead to a hazard classification, in terms of Performance Level Required for the control system controlling the hazard. The standard is related to Agricultural Machines (both tractors and implements as it is recalled by ISO 4254) and the Performance Level in terms of functional safety is called AgPLr, Agricultural performance Level required).
Every Hazard is classified following the hazard tree analysis shown in the left of Figure 2, and the value resulting from the classification, can be obtained with different combinations of system performance, in terms of hardware structure, software quality level (SRL), quality of components and diagnostic coverage of system faults.

In fact, in the right part of the Figure 2 the AgPL table is shown, where in the vertical left axis the AgPL is related to values of Software Requirement Levels (SRL) and quality of components (MTTF = Mean Time To Failure), as a function of both Hardware structure and Diagnostic Coverage (DC), listed in the horizontal axis under the table.

It can be noted that the same AgPL can be obtained with different combination of system characteristics, and that a higher hardware cost can be chosen, with less requirements for the software, and it represent an useful solution for small production series; the opposite solution should be evaluated in mass production, where the extra cost for the hardware is multiplied for the entire production, while the software cost is due only once for each machine or function type.

This degree of freedom was used in the project here described, in order to reach the best mix of technologies for the steering function of a big Agricultural self-propelled machine, whose the production will probably be around 100 units per year.

3. SYSTEM ANALYSIS

The machine mission is to distribute in the field at high pressure the material resulting from the “digestion” of the biogas plants, that are common in Europe and USA. These plants generate energy from the bio-transformation of gases of materials and sewage coming from fields and livestock. The machine usage is typically in the field and in the yards, for charge operations from trucks. Part of the life of the machine is also spent for travels in public roads to move from a field to another, typically in the same region. The average usage of the machine is 1400 hours/year and the typical distribution of the hours is shown in Table 1.

<table>
<thead>
<tr>
<th>operation</th>
<th>Hour/year</th>
<th>% of total time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Work in field</td>
<td>800</td>
<td>57,2</td>
</tr>
<tr>
<td>Charging in the yard</td>
<td>150</td>
<td>9,3</td>
</tr>
<tr>
<td>Discharging in the yard</td>
<td>50</td>
<td>3,6</td>
</tr>
<tr>
<td>Travel in public roads</td>
<td>300</td>
<td>21,4</td>
</tr>
<tr>
<td>Operations in the farm</td>
<td>100</td>
<td>7,1</td>
</tr>
</tbody>
</table>

The maximum machine speed in 40 km/h by regulation approval, while the engine horsepower will be from 250 to 280 HP. Functional safety aspects are available both in the powertrain management, because the transmission is fully hydrostatic with 6 “Zero displacement” Bosch-Rexroth hydraulic piston motors, one for each wheel, and in the steering functionality, because all machine management is fully electronic. Other safety aspects are available also for braking systems (both operational and stationary) and for spray bar position regulation.

The accelerator pedal and machine speed management, as well as cruise control and machine layout (wheel track and machine height) are fully electronically controlled.

The machine functionality can be settled to different “modes”, one of them is “travel” and others are “work in the Field – Eco” and “Work in the Field – Full Power”, as well as “Stationary” for charge and discharge operations.

The main difference between them is the maximum speed and the powertrain management. Differences are in number of tracking wheels and steering angle, as well as driving mode (by joystick or steering wheel).
From the steering functionality analysis point of view, the Maschio Gaspardo - Unigreen Division - Talpa Machine is a self-propelled articulated machine, with three couples of wheels (the machine’s chassis is shown in Figure 3); the two rear wheels are independently steering, while the front steer is assured by the articulated front of the machine. The 6 wheels are all independent and individually controlled for traction by Bosch electro-hydraulic components. The front axle and the rear wheels were designed for providing steering capability, due to the length of 11.1 metres of the machine, and to achieve adequate turning radius for a better driveability in public roads.

![Machine Chassis and Steering Mechanical Structure](image)

*Figure 3. Machine Chassis and Steering Mechanical Structure*

The steering angles of the rear wheels are kept independent from each other and are changed, as a function of machine articulated front angle, with respect to the desired trajectory. The minimum steering radius of the machine is 4.5 m.

The central wheels in the final design are not steering, while the steering angle of the rear wheels is controlled following the steering angle of the machine articulated joint. The primary steering function (front steer), is designed according to applicable regulations and is actuated through an hydraulic power steering system.

![First Fully Electronic Circuit proposed by suppliers](image)

*Figure 4. First Fully Electronic Circuit proposed by suppliers*

From the system point of view, following the safety standards, the steering control system can be designed in a very critical way. The hydraulic power steering, connected with steering cylinders for the front side steering, can be a fully hydraulic solution, made using well tried components and hydraulic architecture. On
the contrary, the desired function for the rear wheels is that the steering angle is a percent of the front axle steering angle in front steer angle follower configuration.

For that reason the rear wheels steer solution should be electronically controlled, in order to avoid misalignments and to provide the adequate configurability as a function of machine driving mode.

In fact for the rear axle the solution can’t be fully hydraulic, because of a leakage in the hydraulic circuit can lead to a difference in wheel direction both in respect to front axle and in respect to each rear wheel. Another problem of the fully hydraulic solution could be the 4 to 2 wheels steering switching, due to rear wheels alignment.

A fully hydraulic solution could benefit not to the lack of safety requirements, because for non-electronic systems the agricultural machines shall be compliant with the ISO 13849, but it could benefit of the usage of well-tried components and well-tried system solutions. But for the previously exposed reasons a fully hydraulic solution is not applicable.

The proposed solutions from suppliers were all related to a fully electronically controlled systems, where each rear wheel's hydraulic actuator (the cylinder) is controlled by a proportional directional valve (4:3), where the P is fed by the primary hydraulic oil supplying steering system under a priority valve, checked by a pressure switch sensor for safety. The proposed typical circuit, with the position of the three steer angle sensors, is shown in Figure 4.

The safety analysis for this circuit is performed with reference to the three main parameters of the safety criticality analysis shown in the left side of the Figure 2: Severity \( S_x \), Exposure \( E_x \) and Controllability \( C_x \).

In order to understand the winning and losing aspects of the solution, a brief description of the meaning of these parameters is mandatory. The Severity is related with the negative effects of a uncontrolled fault to the function under analysis. Severity is then related to consequences of the hazard under analysis. Exposure is the percent of time, in respect to total machine usage time in a year, on which the operator, bystanders or other road users are exposed to the risk of the hazard. Basically, it is the time where the function, related to the hazard, is active or potentially being active if a malfunction occurs, and this characteristic is one of the ones used to reduce the overall risk related to wrong steer of the rear wheels. Finally the Controllability is the possibility for the operator, to actively isolate the fault or nullify the bad consequences of it. This parameter is related to the diagnosability of the fault and to the Frequency Reaction Time of the countermeasures that, or automatically either manually by the operator, can be activated against the hazard, in order to avoid or reduce the negative effects of it. The dynamic of the faulty system is a very important parameter for risk reduction.

The Severity can’t be changed, because is related to the nature of the hazard in respect to machine function. While both Exposure and Controllability can be modified by design, and the paper is related to the reduction of both them.

The Exposure is very interesting parameter, because, depending on the electro-hydraulic system structure, can be very different and can affect in a very negative way the AgPL calculation.

In a similar way the easiness of controlling an hazard, can be very different in function of the Diagnostic Coverage (DC) and thus the observability of the system, and in function of the authority of the actuator, both in terms of amplitude and in terms of power of the actuation.

If, at the fault occurrence, the electronic control systems are able to recognize the fault, even if they can’t isolate it, they can modify the whole machine mode in order to reduce the hazard effect, being a real help to the operator on controlling the faulty machine.

These consideration led the design of the modified system.

The AgPLr for this solution is dependent from the Severity, that is high, because a wrong steering can result in an accident in public road, where some road user can be killer or injured, or the operator can lose the machine control having some accident. While the Exposure shall be evaluated high because the steering function is ever active except in the field in some work operation, then the exposure is really high. Finally the
Controllability is very low, because the proportional three way valve is the only actuator for the rear wheel’s steering and a malfunction in it can’t be corrected in any way. So the global evaluation for the AgPLr is equal to level d.

Many aspects of the standard should be analysed and discussed, but some basic consideration shall be given, in order to understand the gravity of this Performance Level required. The AgPLr = d can be obtained only with a redounded hardware structure, but a MTTFd = HIGH in electronic control units is very difficult to be reached, so the only possible solution for the electronic control unit is a fully redounded Category 3 Hardware (Figure 5).

If the cost of the hardware can be a problem, the real problem of this category is the Software Requirement Level SRL = 2 imposed by the solution. The ISO 25119 standard requires for this SRL formal methods to design both requirements, code and testing protocols. The software project development toolchain and environment required is not economically sustainable for a small production, and only automotive companies today are structured to develop software and requirements under formal methods, procedures and using adequate tools.

Practically speaking an SRL = 2 is not available today in all software suppliers for heavy duty and agricultural machines. Only big companies are working to be compliant with this methodology for mass production (e.g. Agricultural tractors).

As a consequence, an AgPLr = d led to the conclusion that will be very difficult to be compliant with the safety standards for Agricultural Machines.

4. THE STEERING ELECTRO-HYDRAULIC SYSTEM

The original design, proposed to the Company by electro-hydraulic components suppliers was the one shown in Figure 4, where a proportional directional valve is fully responsible for steering in function of the front axle steering angle, as a follower of the front steer. A central series valve enable the steering of rear wheels.

This kind of actuation system is very simple and effective in control especially in a big and slow machine like the Talpa, where the maximum speed is 40 km/h.

But, as briefly described in the previous paragraph, the solution can lead to dramatic hazards. Analysing in a deeper way the proposed system in Figure 4, once the rear wheels steering is enabled, all the steering operations are managed through a single valve for each wheel, and a single fault of one of these two valve can lead to a wrong steering angle and then to a critical hazard. A single fault can lead to the safety function loss.
The two sensors on wheels are functional to the main steering functionality, because they will observe the actual steering angle to be related to the steer angle of the articulated front of the machine.

For the entire period of time in which the rear steering is enabled, the operator, bystanders, or other road users, are exposed to a possible hazard, because the function can result in a fault. Then, the system should be designed in the described way. From the DC point of view, the two sensors, if properly chosen, can represent a very good diagnostic information and can be connected to strategies related to the machine mode and speed, in order to reduce machine speed. But, in case of a fault in one of the valves, nothing can be done, in order to change the steering angle of the faulty wheel.

The proposed solution is based on the paradigm of trying to use hydraulic solutions, adding electronics to solve the hydraulic leakage, or to make the hydraulics more flexible and easy to use, or more comfortable.

The proposed solution is then based on the slave cylinders architecture for rear wheels steering. In the hydraulic circuit, two pulled cylinders are added in parallel with the front axle steering actuation system; these two cylinders directly control the oil in the actuation cylinders of the rear wheels. This architecture can be affected by wheel misalignments, so a couple of load dump valves are added (one for each wheel), in order to correct a wrong wheel angle only dropping a small oil flow, and only if an electro-hydraulic on/off enable valve is powered.

![Figure 6. Concept of the Electro-Hydraulic Solution](image)

The solution is shown in Figure 6. In the solution the control valve is activated only when the hydraulic system that controls the rear wheels steering is affected by a steering angle error, in respect to the front axle of one or both rear wheels. If no errors are detected by the angle sensors the drop valve and the enable valve are not activated. So the functional safety of the electro-hydraulic component shall be performed calculating a reduced exposure, because the most of the steering actions are performed with a pure hydraulic system without any electronic controlled steering correction.

The controllability is much higher in respect to the fully electronic solution, because of the reduced oil flow of the dropping on/off valve, that only serves steering angle correction and not steering angle actuation.

Moreover a fault in a single valve doesn’t affect the steering angle if the enable valve is de-energized. Finally, the DC of this solution is at least equivalent with the previous one.

In order to avoid cavitation and to maintain the control pressure in all the cylinder sides, feeding valves are added, to avoid that the oil flow dropped from the control valves lead to low oil pressure conditions.
The solution presented some feeding oil problem and some asymmetry in the functionality, then it was replaced by the one in Figure 7, that is symmetric, with independent control for both sides of the actuation cylinder and that maintain the double valve series for the complete enable of the angle error correction control.

In the Figure it can be noted that the slave cylinders structure of the solution in Figure 6 is maintained, while the oil drop actuation system was totally changed, using a three way valve for each wheel, to load dump the oil flow to the tank, every time a correction in the steering angle of a rear wheel is needed.

The enable valve is in series to the drop valve, so a fault with an unintended activation of one of the control valves results in a no-hazard condition because of the enable valve.

Exactly as in the previous system, the steering actuation is performed by the hydraulic system and, in case of lack of errors, the electronic system doesn’t result in any action. The electronic correction system doesn’t perform any action if the rear wheels steering angle doesn’t present any error.

Then the functional safety analysis is performed not on a rear steering function but on a rear steering error correction function. This function will be active in the life of the machine considerably less time in respect to the rear steering, that is purely hydraulic. From the functional safety point of view, the resulting exposure is deeply reduced and the AgPLr is consequently lower.

The AgPLr can be also reduced because of the slow correction action performed by the electronic control system controlling the drop valve: the small flow resulting in a wrong actuation due to a fault, can be easily corrected by the operator controlling the steering wheel and reducing the machine speed.

Considering the exposure less than 1% of the total machine time per year and the controllability as “easy controllable”, the resulting AgPLr is b, reachable with a lower hardware category and a lower SRL.

5. THE MACHINE ELECTRONIC CONTROL SYSTEMS

The Talpa Machine is a fully electronically controlled machine, and the rear steering isn’t the only machine function to be compliant with functional safety aspects. In fact the cruise control, the speed control with joystick and different machine modes, that modify many machine functions, and enable the work of the machine in the field, need for a complex machine electronic control.
The machine control is managed by a distributed control system, performed using 7 connected Electronic Control Units (ECU). The main machine control ECU is the one called Machine Main Control Unit (MMCU). The MMCU acquires all the main signals directly or through the CAN network and sets the machine mode, enabling the machine functions. The CAN network is the main communication medium and it is considered critical for safety and for machine function, as already stated in [7]. For all these reasons, the CAN network is redounded, in order to be able to safely move the machine, in case of network fault. The machine is then provided by a powertrain SAE J 1939 network, where transmission control and engine control exchange the main signals, and by a machine control network, where the inputs from operators and commands to steering, machine setup and rear implement functions are exchanged. The network topology is presented in Figure 8.

The MMCU is a Category 2 (Figure 5, left) ECU capable to reach the AgPL=C, that is required by other machine functions, while the SRL required level is 1 (following the table in Figure 2), easy to be reached under a proper development lifecycle control. The unit was designed in accordance in basic safety principles already published in [9], while the language chosen was the C language, that is a Full Variability language (FVL), instead of a Limited Variability Language (LVL), like Codesys, that could provide a higher easiness in SRL = 1 certification, as explained in [8] and confidence in testing.

The networked control is ensured in the most important and safety related units by a redundant CAN network, the Main Control Unit and its peripherals are designed to comply at least Diagnostic Coverage Level = Medium.

In order to safely acquire the steering angle, both of the front axle and of the rear wheels, three redundant double angle sensors are installed respectively in the articulated joint (Figure 9), in the machine front, and in each rear wheel (Figure 10). In that way not only the position of each part responsible for steering is acquired, but also it is diagnosed using the complete redundancy of input signals of angle sensors.

From the actuation point of view, the Steering correction of the rear wheels is enabled by a load dump central valve and from the single wheels oil dump control valves in series configuration (AND configuration). Only when both valves are enabled the steering correction can be performed. Then a single fault on a valve can't affect the steering function.
All valves, even are if only ON/FF valves, are controlled through a double electronic power stage, composed by an High Side Driver and a Low side driver in asymmetrical half bridge configuration, as shown in Figure 11. Then a single short fault on electronic actuation stage, can’t unintentionally activate the valves, and the consistency of command action are ensured by the main enable command, that is activated by both microcontrollers in AND configuration. MMC is the Main Microcontroller in Figure 11, while SMC is the Safety Microcontroller. The High side is used for Valve control command, while the low side driver is kept active and is de-energized only in case of fault. Both drivers provide a feedback to the microcontroller, in order to recognize the faults and activate fault recovery strategies. The consistency of control, from the logical and functionality point of view, is ensured by an SPI communication between the two microcontrollers.

Both enable valve and the rear steering correction valves, are controlled through the architecture presented in Figure 11. The Performance Level reached by the architecture presented is AgPL = c, higher than the level b required.

![Figure 10. The rear wheel steering system with the steer sensor and the cylinder for steer actuation](image)

![Figure 11. Main Machine Control Unit Valve Control Architecture](image)

The electronic system acts on rear wheels not for steering, but only to correct angle errors in the rear wheels. In order to increase the safety Performance Level an additional countermeasure was taken: in case of angle error the operator is warned by an acoustical and visual messages on the machine display, but also he is requested to acknowledge the angle correction angle if the error is recognized in travel mode. In fact when the machine is in travel in public road a special attention is taken on the steer angle. The operator must change the machine mode from travel to work, then, with a limited maximum speed, the electronic control system can start the steering angle correction. The operator is responsible to verify that the road conditions
are suitable to act steering correction. In order to regulate the size of the angle correction, an intervention threshold in angle degree can be set.

6. CONCLUSIONS

The Functional Safety analysis of electro-hydraulic systems under the new regulations, especially the Type C ones for Agricultural Machines ([5], [6]) and for Earthmoving Machines ([4] and the recalled Type B [3]), often results in a trouble for Performance Level required compliance. The Severity of hazards lead to high levels of Performance Level required, that need for complex electronics and a Software Quality very difficult to be reached by the most part of suppliers today.

A proper analysis of the system, the consciousness that normally hydraulic components are normally classified "well tried", then reliable from the Safety point of view, can help designers to find a compromise between function required and Functional Safety performance Level required.

This paper shows a way to obtain an acceptable AgPLr maintaining the full function required by the machine specifications. Moreover the paper presents a solution that not only allow the main machine function even in case of fault, but also provide a recovery strategy to increase the controllability level in the functional safety hazard analysis. The way in this case is the change of electronically controlled function, in order to reduce the impact of the function from the functional safety point of view.

The analysis performed, emphasizes that a big effort is requested to companies involved in electro-hydraulic systems design for mobile applications, in order to synthesize a development lifecycle for requirements, software, system and software tests, all based on formal or semi-formal design methods, in order to prove the consistency of statements and software, in a fully comprehensive environment.

REFERENCES

[5] ISO 25119 "Tractors and machinery for agriculture and forestry - Safety-related parts of control systems".
EFFICIENCY OF DIRECT DRIVEN HYDRAULIC SETUP IN ARCTIC CONDITIONS

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ABSTRACT

This paper focuses on investigation of directly driven hydraulic setup (DDH) for non-road mobile machinery (NRMM) application in arctic conditions. The control of the system is implemented directly with a servo motor drive without conventional hydraulic control valves. Speed of the double-acting cylinder is determined by incoming oil flow from the pump, out-coming flow to the hydraulic motor and angular speed of the electric motor. Efficiency measurements of DDH setup performed in different operating conditions (speed, payload, duty cycle) and environmental conditions (temperature, humidity). Influence of temperature on results is studied in range of +22 to -10 °C. Thermal images across the pump and electrical machine demonstrate relative efficiency of the components. Change in fluid temperature (delta T) across the pump indicates volumetric/friction loss within the pump and demonstrates its behavior during cold start-up. Sankey diagrams shows reduced pumping and system efficiency as heat generation results in energy consumption and heat load within the system during lifting.

KEYWORDS: Arctic conditions, low temperature testing, energy efficiency, fluid power, direct driven hydraulics, non-road mobile machinery, displacement controlled systems.

1. INTRODUCTION

Recent year’s tendency in industry for compact electro-hydraulic systems which deliver powerful, linear movement with valve-controlled or pump-controlled systems is observed. These technological steps are considered important steps towards a solution for “old problems”, such as reduction of CO2 emissions and improved performance / productivity / reliability / controllability [1, 2] as well as good options in forcing a diesel engine to operate within its optimum efficiency area. As in general, the diesel engines of non-road mobile machines (NRMM) operate far from their optimum efficiency range [3].

Today, as the search for natural resources expands to take in areas previously considered too remote and environmentally hostile to support viable extraction and processing operations, research all around the world are facing the challenge of establishing and maintaining industrial activities in very cold environments, such as the Arctic region. Similar conditions exist in both Polar Regions, and are can be found in other places such as Finland, Norway, Alaska (USA), Russia, Sweden and Canada.

Considering continuous operation in arctic environment, problems with the lubricants have not been observed. In fact, the majority of the overall heat production in the powertrain NRMM is the combustion heat created by the engine and the heat from the friction of moving parts in the transmission and hydraulic system.
and hydraulic throttling in the valves. The heat generated from the equipment operation keeps the all fluids warm and enables them to circulate properly. Problems come at start-up, when everything has cooled down to low temperatures. Under these conditions, a conventional hydraulic fluid will solidified, so when the NRMM starts up, there is no fluid circulating to protect the pump and other components from extreme wear. At start-up, there will be no time for the combustion heat to transfer to other media such as the lubrication and hydraulic oils of the system, and the metal parts of the powertrain itself. For successful operation, a quick start-up is crucial and the risk of equipment failure is unacceptable in remote locations.

Currently, research on the cold operation of NRMM is limited mostly to cold-start characteristics of engines [4] and to the development of new components and hydraulic oils specifically for arctic conditions [5-9]. It was reported in [10] that the engine start-up procedure varies significantly depending on the operational temperature. Various solutions exist to shorten the heating time of engine, for example, by using an external heater [11] or by using exhaust gas heat [12] or even use of energy buffer of the hybrid powertrain, or by extending the design space of the Energy Management Strategy with an additional temperature state [13,14]. In the existing literature, the analyses of the electric parts of NRMM are mainly focused on the investigation of extending life spam of the energy storage [15] and on the analyses of electrical machine in cold conditions [16]. Heat generation and transfer in oil-hydraulic systems is discussed and researched from overheating point of view only [17] or predicting accurate temperature [18,19]. For example, in [20, 21] thermo-dynamical behavior are studied in typical working temperatures (above 0 °C) and cycles. In [22] study in room temperature of energy efficiency and thermo-energetic behavior of compact drives using ITI simulation combined model of hydraulics and thermodynamics. Only few available extensive experimental research series in arctic conditions was done. In Finland in end of 1980s the research included testing different pumps [23], hydraulic motors [24], cylinder seals [25], O-ring [26] and hose assembly [27] in outside environment with the temperature range of -20°C to -50°C. In [28, 29] another independent experimental studies from Cracow university showed that operation in low temperatures require proper selection of hydraulic fluids and careful design of pump suction system. Therefore, it appears that detail modelling analysis of hydraulic components of NRMM in cold environment and especially cold start phenomena have been ignored. It is well-known very cold temperatures are as dangerous as overheating and extreme cold affects the viscosity of lubricating and hydraulic oils. Increased resistance to flow at low temperatures interferes with the lubrication of vital parts and slows the operation of hydraulic systems. Too high viscosity could cause catastrophic failure of the pump at start-up, while too low viscosity could accelerate the pump wearing because it may compromise the thickness of the protective oil films causing metal to metal contact. In conventional systems, excessive temperatures will oxidize the oil and lead to sludge deposits in the system. On the other hand, running the temperature too low will allow condensation in the reservoir and increase the likelihood of pump cavitation. Some serious problems affecting cold weather operations are a direct result of the interaction of cold and moisture. Also possible thermal shocks resulting from a too rapid increase or decrease in temperature exposes frame components to severe stress that can lead to sudden damage or reduced service life.

New configurations for NRMM, like for example hybrid solutions without the traditional source of heat and constantly running engine are coming. Taking in account the above mentioned configurations, the following questions are naturally born: will NRMM have problems if there is no enormous heat source, i.e. the combustion engine, in proximity? What kind of future NRMM’s would work best in arctic conditions? Will standard procedure to winterized hydraulic systems be enough? How will the electro-hydraulic systems behave in extreme cold temperatures? What kind efficiency can be expected? Will having very high efficiency mean no extra heat available?

To discuss these questions this research is conducted. Therefore, this paper focuses on investigation of directly driven hydraulic setup (DDH) for non-road mobile machinery (NRMM) application in arctic conditions. The remainder of this paper is organized as follows: Section 2 introduces a test setup. Measurement results and analysis are described in Section 3. Section 4 and 5 contain discussion and concluding remarks.
2. TEST SETUP

The experimental setup uses a speed-controlled electric servo motor drive rotating two hydraulic pumps to directly control the amount of hydraulic oil pumped directly to the double-acting cylinder. The experimental test setup is illustrated in Figure 1. For detail information regarding components refers to [30, 31, 32].

In this case, oil is circulated only when movement of cylinder is needed and it is observed that in practice oil has only a minor tendency to heating up. The hydraulic pump creates a flow depending on the rotating speed of the servo motor. The oil pressure rises to the required level as determined by the payload. During lowering motion of payload, the potential energy of the payload creates a flow that rotates the hydraulic machine as a motor, and the mechanically connected electric motor acts as a generator, which is controlled by the frequency converter. The speed-controlled generator controls the amount of fluid flow and the position of the payload. A program for the electric drive controls both the electrical and hydraulic sides of the system as there is no valve control.

Cold start up conditions is investigated. The cold start conditions represent the situation when the device has been stopped for a few hours, so the powertrain of NRMM is at its ambient temperature. Each cold start condition was performed for representative single lifting-lowering driving cycles from ground level to middle stroke lengths and back with motor speed 300 rpm and 175 kg payload. The locations of heat losses were captured with a thermal camera. This setup is tested in a cold chamber. Cold chamber specification is -50°C to +30° C. Essential parameters like pressure, torque, speed, cylinder position and velocity, electric motor voltage, current, energy consumption and energy efficiency will be either directly obtained from the setup with sensors or can be calculated. Figure 1 b illustrates schematics of setup and location of sensors.

![Figure 1. A) overview of DDH setup in cold chamber, B) schematic of DDH setup: a) double acting cylinder, b) wire-actuated encoder, c) pressure sensor, d) reversible gear pump/motor_1, e) PMSM motor/generator, f) pressure sensor, g) reversible gear pump/motor_2, h) pressure sensor in tank line, i) pressure sensor in tank line and j) oil tank.](image)

3. EXPERIMENTAL INVESTIGATION: RESULTS AND ANALYSIS

It is well-known fact that temperature is a determinant parameter in terms of performance, lifespan and safety working. The DDH test setup was investigated in a cold chamber with change environment temperature from -10° to +22° C.
The experimental efficiencies and energies of the DDH test setup system for lifting movements were calculated as shown below:

\[ \eta_{\text{up,net}} = \frac{E_{\text{net}}}{E_{\text{mech}}}, \]  

where \( E_{\text{pot}} \) is the potential energy of the payload

\[ E_{\text{pot}} = mgH, \]  

where \( m \) is mass of payload in kg, \( g=9.81 \) is gravitational constant in m/s\(^2\), \( H \) is position of cylinder’s piston in m.

\( E_{\text{mech}} \) is energy of the shaft and calculated as the integral of the power at the shaft \( (P_{\text{shaft}}) \).

\[ P_{\text{shaft}} = T\Omega, \]  

where \( P_{\text{shaft}} \) is output energy of shaft in W, \( T \) is motor torque in Nm and \( \Omega \) is angular speed in rad/sec. Angular speed measured and motor torque estimated by motor control algorithm.

Output energy of hydraulic part \( E_{\text{hydr}} \) is calculated as the integral of output hydraulic power:

\[ P_{\text{hydr}} = pv_c A, \]  

where \( p \) is pressure in Pa, \( v_c \) is velocity of cylinder piston m/s and \( A \) is cross area of cylinder piston in m\(^2\). Pressure measured by Gems 3100R0400S pressure transducers. Velocity of cylinder piston measured by a wire-actuated encoder SGW/SGI by SIKO.

Depending on the operating point of a hydraulic pump/motor unit in its performance curve, the relationship between the flow and hydraulic losses in a system varies significantly. During lifting, hydraulic pump/motor unit operates as a pump. Input energy is mechanical energy and output is hydraulic energy. Hydro-mechanical losses contains shaft and pump losses. Hydraulic losses in DDH systems are composed of pipe friction losses, elbows and other fittings, entrance and exit losses, and losses from changes in the pipe size by a reduction in the diameter and cylinder losses. A double-acting lifting cylinder is used in the DDH test setup. The overall cylinder efficiency is mostly dependent on the frictional losses encountered by the piston and the rod during its stroke. Frictional losses depend on the pressure difference across the seal, sliding velocity, seal material, temperature, time, wear, and direction of the movement.

Following section presents Sankey diagrams for measured losses/energies and efficiencies of the DDH system for temperatures in range of +22 °C and -10 °C discuss measurement results and their analysis.

Figures 2 represent reference Sankey diagrams for measured losses and efficiencies of the DDH system in ambient temperature 22°C.
Figure 2. Reference Sankey diagram of the system during lifting with motor speed 300 rpm and payload of 175 kg in ambient temperature 22°C.

Figures 3 represent Sankey diagrams for measured losses and efficiencies of the DDH system in ambient temperature -6°C. It can be seen with dropping temperature by 28°C, energy consumption increased over 100 % compare to reference 22 °C Sankey diagram. Low temperature increased hydraulic losses from 20.9 to 50.4 % and hydro-mechanical losses from 14.4 to 25.5 % in DDH setup.

Figure 3. Sankey diagram of the system during lifting with motor speed 300 rpm and payload of 175 kg in ambient temperature -6°C.

Figures 4 illustrate Sankey diagrams for measured losses and efficiencies of the DDH system in ambient temperature -10°C.
It can be seen there are no significant difference between Figures 4 and 5, this similarity in energy consumption can be explain due to torque limitations. Electric motor is undersize for performing required speed of 300 rpm with 175 kg payload in temperature below zero. This also can be seen in Figure 5, when torque saturates to level 16 Nm.

Figure 5 shows impact of the temperature on start conditions on the direct drive even over this small temperature range. The drop of the temperature start torque for the DDH test setup grows by 100 % (from 5 Nm to aprox.10 Nm); it is similar as with energy consumption.

Figure 6 illustrates that thermal image of the electric motor during start procedure in ambient temperature of 10 °C. It can be seen that electric motor heats up faster during start up meanwhile hydraulics stays cool. But it is worthwhile to notice that waste heat from electric motor is small even though it makes colorful pictures.

Figure 4. Sankey diagram of the system during lifting with motor speed 300 rpm and payload of 175 kg in ambient temperature -10°C.

Figure 5. The impact of the cold start conditions on the energy consumption of the DDH test setup, evaluated for driving cycle. Torque versus time.
is unlikely it could be used as source of heat like is case in combustion engines exhaust gas or cooling water. Electric motor controller has also tendency to generate waste heat but there is similar situation. If controller has proper parameters it is unlikely to have waste heat enough to heat up hydraulic oil. Of course having separate heating parameters amount of waste heat could be increased but it is likely to have is easier by direct resistive load or having proper oil for low temperature operations.

![Figure 6. Thermal image of setup in 10 °C ambient temperature](image)

4. DISCUSSION

Experiments showed that with dropping temperature below zero, energy consumption increased more than twice compare to reference room temperature. Lower temperature dramatically increased hydraulic losses from 20 to 50 % and hydro-mechanical losses from 14 to 26 % in DDH setup in all showed results. It is known and worth to remember very cold temperatures are as dangerous as overheat and extreme cold affects the viscosity of lubricating and hydraulic oils. Increased resistance to flow at low temperatures interferes with the lubrication of vital parts and slows the operation of hydraulic systems. Too high viscosity in cold could cause catastrophic failure of the pump at start-up, while too low viscosity in high temperature could accelerate the pump wearing because it may compromise the thickness of the protective oil films causing metal to metal contact. In conventional systems, excessive temperatures will oxidize the oil and lead to sludge deposits in the system. On the other hand, running the temperature too low will allow condensation in the reservoir and increase the likelihood of pump cavitation. Some serious problems affecting cold weather operations are a direct result of the interaction of cold and moisture. Also possible thermal shocks resulting from a too-rapid increase or decrease in temperature exposes frame components to severe stress that can lead to damage or reduced service life.

Results of presented experiments can be applied to most non-road mobile machinery for all Scandinavian and Nordic countries (Russia, Canada, USA (Alaska)), as cold conditions are common in these regions and expansion of search for natural resources is growing. When engineering and making the choice of the oil for hydraulics systems to be used in low temperatures there demand to consider matters related to start up or heat up. These results can help to create future NRMM suitable for extreme climate conditions. Further research for creating models of DDH in extreme cold conditions is required.

Thus, combined with the results from additional tests, several modifications and enhancements are expected to be made during the further part of the project after extensive testing. To investigate potential aging effects of temperature cycles on the selected components, long term tests need to be designed and run.

5. CONCLUSION

This document mainly describes knowledge gained from experimental study of the first prototype of the direct driven hydraulic setup for NRMM application. In this study efficiencies of the components of the directly driven hydraulic test setup without control valves were determined with help of Sankey diagrams for different
ambient temperatures. The tests showed that the energy consumption varies with the change of the outside temperature even when change of temperature is not a major one. As Sankey diagrams showed, the hydraulic losses dominate in the system. Furthermore, the investigation of the effects of temperature on the DDH is deepened. To investigate potential aging effects of temperature cycles on the selected components, long term tests need to be designed and perform.

ACKNOWLEDGMENT

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MODULAR SOFTWARE DESIGN OF SAFETY RELATED SYSTEMS
FOR MOBILE MACHINERY
- RELIABILITY - TESTABILITY - SIMULATION -

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ABSTRACT

This paper identifies the challenges during the development of control software for safety critical electro-hydraulic systems which shall be implemented in a wide-range of mobile machinery. The approach is based on a modular software design which provides a maximum of reliability and reusability of software components without reducing the flexibility or functionality of the system.

An inevitable prerequisite in order to provide functional safe software is the application of directives and standards for safety related software development [1], [2], [3], [4]. This paper will mainly focus on ISO 13849 [1] but general conclusions will be drawn extending beyond this specific standard. Due to the fact that safety requirements for software are entirely process related, the testability and documentation of safety related software is of key importance. In order to achieve testable software test features have to be implemented into the software by designing testable function modules, limiting the logic combination between functions, implementing test interfaces and entry points into the code. Therefore the testability of the software has to be defined at the very beginning of the software development process.

The focus of this paper will be laid on identifying the safety requirements for software as mandated by the standards for safety related software development. Furthermore it will highlight how the chosen approach of modular software design will help to fulfill these requirements.

KEYWORDS: Functional Safety, Modular Software Design, Software Test, Simulation

1. NORMATIVE REQUIREMENTS FOR SAFETY RELATED APPLICATION SOFTWARE (SRASW)

As a starting point and in order to understand the normative requirements this chapter transcripts some of the most important recommended measures for the development of safety related application software according to ISO 13849 [1]. It has to be noted that the requirements which are set by the ISO 13849 [1] differ slightly compared to the requirements set by other software development standards especially the IEC 61508 [2]. Thus the following list cannot be considered complete or sufficient for a specific software development. The mere intention of this list is to explain the basic idea of process related safety measures for software development.

The following safety measures shall be applied during safety related application software development:

- Software safety lifecycle with verification and validation activities (compare V-Model Fig. 2)
- Documentation and specification of design
• Modular and structured software design and coding
• Control of systematic failures during development process
• Verification of correct implementation of error management for random hardware failures
• Appropriate software safety lifecycle activities after modifications

Selecting tools, libraries, languages according to ISO 13849 [1]:
• Suitable tools with confidence from use (…)
• (…), validated function block (FB) libraries should be used
• (…) limited variability language (LVL)-subset suitable for a modular approach should be used, (…)

Software design according to ISO 13849 [1]:
• Semi-formal methods to describe data and control flow, e.g. state diagram or program flow chart.
• Modular and structured programming (…) safety-related validated function block libraries.
• Function blocks of limited size of coding.
• Code execution inside function block which should have one entry and one exit point.
• Architecture model of three stages, Inputs ⇒ Processing ⇒ Outputs (...).
• Assignment of a safety output at only one program location.
• (…) detection of external failure and for defensive programming (…) which lead to safe state.

Software implementation/coding according to ISO 13849 [1]:
• code shall be readable, understandable and testable and, (...);
• justified or accepted coding guidelines shall be used (…);
• data integrity and plausibility checks (…) available on application layer (…);
• code should be tested by simulation;
• verification should be by control and data flow analysis for PL = d or e.

Testing according to ISO 13849 [1]:
• (…) validation method is black-box testing of functional behavior and performance criteria (…);
• for PL = d or e, test case execution from boundary value analysis is recommended;
• test planning (…) should include test cases with completion criteria and required tools;
• I/O testing shall ensure that safety-related signals are correctly used within SRASW.

2. SAFETY RELATED SOFTWARE DEVELOPMENT

Development of safety related software causes an increased effort in testing and documentation. Testing, documentation and development tasks are almost equally time-consuming which makes it essential to reduce the effort and provide more effective approaches to all three development steps.

The development process according to the V-Modell (compare figure below) requires following the sequence of specifying the requirements, coding the software functionality, validation of the safety requirements as well as documentation of software design and test activities. The software validation and verification in particular has to be accounted for from the very beginning of the software development.
3. VERIFICATION AND VALIDATION OF SAFETY RELATED SOFTWARE

Software testing covers systematic failures only, the software itself is considered to be free of random failures. Random failures are hardware related, they will be identified by hardware endurance testing and hardware failure probability calculations which will not be covered by this paper.

The Validation & Verification efforts can be grouped into process related quality management type efforts to reduce failures before they actually happen on the one hand and into actual test setups to identify systematic failures on the other hand. Within the standards the definition for V&V is the following:

**Validation** is the task of identifying if all safety requirements have been accounted for. The question is “what” has been done to cover all requirements? The way to prove this is usually a high level functional and failure mode test.

**Verification** checks if all specified features and mechanism have been implemented correctly. The question is “how” have the safety features been realized. Do they fulfill the specification? These questions are typically answered by analysis the first step would be “on paper” which is usually sufficient for all hardware related questions. For software verification there are several steps of actual software testing involved, the range of which depend on the complexity of the system. These software tests can be executed as black box or rather grey box tests. The basic rule of thumb states that the higher the complexity level of the software the wider the scope of testing has to be.

**Validation of safety-related software according to ISO 13849 [1]**
- The validation of both safety-related embedded software (SRESW) and safety-related application software (SRASW) shall include
  a. **Operational aspects**: the specified functional behavior and performance criteria (e.g. timing performance) of the software when executed on the target hardware,
b. **Analysis:** verification that the software measures are sufficient for the specified PLr of the safety function, and

c. **Quality management:** measures and activities taken during software development to avoid systematic software faults.

- Review of documentation for the specification and design of the safety related software for:
  - Completeness, absence of erroneous interpretations, omissions or inconsistencies.

- Depending on the PLr (for SRESW, SRASW) the tests should include:
  a. **Functional test = Operational aspects:**
     black-box or grey-box testing of functional behavior and performance,
  b. **Limit value analyses:**
     additional extended test cases based upon limit value analyses, recommended PL d or e,
  c. **I/O tests:** including inspection of data communication interface
to ensure that the safety-related input and output signals are used properly, and
  d. **Fault injection test:**
     test cases which **simulate faults** determined analytically beforehand, together with the expected response, in order to evaluate the adequacy of the software-based measures for control of failures.

- Software module integration tests:
  Individual software functions which have already been validated do not need to be validated again. Where a number of such safety function blocks are combined for a specific project, however, the resulting total safety function shall be validated.

4. **MODULAR SOFTWARE DEVELOPMENT**

Within the development of safety related software a good estimation is that only 30% of the development effort actually relates to machine specific functionality of the application software. The rest are recurring functionalities which could be standardised and provided as basic services. Providing a thoroughly tested software layer with pre-installed basic services would serve two purposes: it significantly reduces development efforts and furthermore it complies with the safety standards (1.2).

*Figure 2. Development efforts for safety related application software*
In order to comply with the standards [1] the software design shall be modular and structured. The implementation of the hardware error management needs to be verified, tools shall be proven in use, validated function block libraries and limited variability language subsets are recommended. The chosen software architecture will distinguish three different layers, tested and provided by a software development environment (MATCH): the standard core software functionalities, re-usable function blocks from a library and an application specific software layer with application specific logic and functions coded in C or CoDeSys. See Fig.4 for the software layer model.

Basic software functionalities like failure control mechanisms, NvMem management, diagnostic and communication interface or hardware API wrapper have been integrated into a “Middleware” provided by the development environment.

Re-usable functional blocks cover functions which are typical for mobile machinery. The goal is that they can be parameterized and configured to adapt to different machines but don’t need to be programmed from scratch. Thus the testing of the software module stays valid for multiple applications. Examples are i.e. ground drive control module, proportional EH control axis (Lift-Lower), variable fan drive, steering module, suspension module.

The third layer consists of the machine-specific software components, these cover individual logic and functional requirements which have to be coded from scratch by the software developer.

Layer one and two are part of the “Middleware” which will be TÜV-certified to PLd [1], AgPLd [3], SIL2 [2]. Hence the test effort is reduced for the application software as all functions which are covered by the Middleware will comply with the software design requirement (1.2) of “function blocks deriving from safety-related validated function block libraries”.

The software development environment MATCH incorporates an auto-code-builder which provides a software frame with layer one and two and provides the frame for each preconfigured software module including the header and the I/O drivers. That means the programmer can focus completely on the functional logic of the new block.
6. SOFTWARE MODULES

Reusable software functions improve the robustness of the software as these functions will be proven in use on many different applications. They also reduce the test effort of the individual application project as once tested the function does not need to be tested as a software module but only on software integration level and on validation level.

The challenge is to cover different machine types, several scopes of functions and options, different user interfaces with one software approach. In order to do this one has to identify re-usable elements, define an interface for hardware independent software development, including multiple PinOuts of same controller hardware or integration of additional IO-modules.

In order to standardize software functions for maximum testability, reusability and flexibility they need to be integrated in modular software architecture with:

- Well-defined data links - No logical combination between functions – to allow rearranging software functions in various combinations.
- Configurable and parameterisable functions - in order to work for different applications with varying controller platforms and/or components without changing the code.

How can software be designed into modules? What should be standardized – what needs to stay flexible?

- Machine functions – vehicle level
  a. Different types of machines featuring multiple sets of functionalities
  b. Varying sizes and differentiation of the related components (i.e. higher control current for bigger valve for higher flow rates)
- Software functions – configuration level
  a. Selectable options of machine type and installed functionalities
  b. Parameterization of characteristics according to size, components or operator preference
- Safety functions
  a. Individual judgment of safety requirements according to machine type and application
  b. Adapt thresholds and failure reaction according to machine type, size and/or component
- Error management and diagnosis
  a. Standardize trouble code labeling and diagnostic concept (i.e. DTC by J1939)
  b. BUT: Differentiation in usage and operating strategies
The figure below shows a typical proportional electrohydraulic control function. It consists of a two directional proportional axis, here a "Lift-Lower" function of a loader. The goal is to design a reusable software function block with configurable characteristics for inputs, outputs and ramping/damping.

The structure shall comply with the three staged “Input-Logic-Output” approach as required by the standard [1], which leads to pre-defined input and output driver blocks which feature their own failure management. And to a Software function block which incorporates the functional logic.

If we zoom in onto the software function block beyond the logic of this function there needs to be a defined input and output interface connecting to the corresponding driver blocks. Additionally a configuration set has to be defined describing all parameters by name and size. These parameters can be modified later on during run time of the software. If all this information is condensed into one software function block, a new software module is created.
After testing this software module and the integration of the software module with the software environment a library block can be created. This example of a library block combines at least 6 individual software modules and the specific functional logic into only one new software module.

All consecutive application software projects can make use of this library block. Different hardware components can be connected externally by modifying the characteristics and operator preferences can be accounted for by selecting different ramping damping sets. All of this leaves the core software of the block untouched and therefore it is not necessary to test this software module again.

7. HARDWARE RELATION AND SUMMARY

The software development approach described in this paper is based on a system architecture design which is capable of multiplication on different machine types. The hardware layout typically consists of a main controller which covers the basic (safety) functionality and add-on modules which allow for functional extension and differentiation of the hardware topology.

The described software development approach makes use of the mobile application tool chain “MATCH”, which specializes in the development of control system software for mobile machinery. With MATCH the entire development activities are supported (compare figure below) - ranging from the system layout, to the software design and software testing, as well as service/diagnostic tasks and documentation.

This software development approach complies with current safety standards for software development. The software frame incorporates a pre-tested and certified Middleware providing basic core functionalities like: functional library blocks, error and NvMem management or communication interfaces. With this approach it was possible to significantly reduce the development efforts and especially the range of required testing.

Figure 7. Development steps during application software development

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   ISO 25119-3:2010(E) Part 3: Series development, hardware and software
   ISO 25119-4:2010(E) Part 4: Production, operation, modification and supporting processes
ISO 25119 will be succeeded by DIN EN 16590 currently in draft status.

ISO 15998 will be succeeded by ISO/NP 19014: Earth-moving machinery -- Control system safety -- Risk assessment and determination of performance level
DIGITAL HYDRAULICS ON RAILS – PILOT PROJECT OF IMPROVING RELIABILITY ON RAILWAY ROLLING STOCK BY UTILIZING DIGITAL VALVE SYSTEM

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ABSTRACT

Finnish Railways (VR) is a state-owned railway company in Finland. It has 18 six-carriage high-speed body-tilting Pendolino trains (“Pendolino”) which have been built in years 1995-2006. The Finnish Pendolino train is based on the FIAT Ferroviaria’s ETR 460 model but it is adapted to Finnish environmental conditions. The motivation for the research work was to improve the reliability of the tilting system and to decrease the life cycle costs. The Pendolino’s tilting system and retrofit options were studied by Tampere University of Technology (TUT). The most promising option found was substituting unreliable and costs generating servo valves with a robust Digital Valve System (DVS). In early phase, one tilting module was modified at TUT and the concept level testing was carried out. Functionality was proven and Bosch Rexroth developed and designed a market-proof version of a digital hydraulic tilting system for Pendolino train. New systems were tested on both at stand still depot tests and on rails at normal operation velocities. Since the successful verification tests, two cars with the digital hydraulic tilting system have been in line service for more than two years with excellent reliability. The new DVS based tilting system is now replacing the original system in the entire Finnish Pendolino fleet.

KEYWORDS: Digital Fluid Power, Rolling stock, Pendolino train, Tilting system, Digital Valve System, Reliability

1. INTRODUCTION

Short journey time is playing a major role in the competition of the customers in public transportation. The Finnish train operator VR has answered to that demand by using tilting trains, which has given a possibility to increase operational speed of the trains without big investments into the Finnish rail infrastructure.
The Finnish Pendolino train (Figure 1) is based on the FIAT Ferroviaria’s ETR 460 model but it is adapted to Finnish environmental conditions and also for certain VR’s special demands. An active tilting system allows higher speed in service on tortuous traditional rails. The maximum velocity of the Finnish Pendolino train is 220 km/h.

Figure 1. Pendolino is based on the FIAT Ferroviaria’s ETR 460 model. On the right, a side view of the bogie after the first test run.

A train with tilting cars can run through the curves with up to 35% higher speeds compared to conventional trains. Non-compensated lateral acceleration at bogie level can be 2 m/s², but the tilting compensates it down to 0.65 m/s² at passenger level. If the lateral acceleration values are too high, passenger may get motion sickness symptoms. [1]

The tilting system has some portion of the train purchase price but it accumulates even more costs during the 40 year’s lifetime of the train. As a part of the life cycle management of the Pendolino trains, VR started the R&D project first with the Tampere University of Technology (TUT) who had promising results of the utilisation of the Digital Valve Systems (DVS) in mobile machines, paper applications and other industries. Co-operation was started with a preliminary study where the original system was analysed. As a result, different ways to improve tilting system were proposed. One of them was substitution of servo valves with a DVS which could improve reliability and decrease maintenance costs.

In the next phase, tilting system with the DVS was modelled. Simulation results with the DVS were promising and laboratory tests were decided to start. VR was satisfied with the performance of the new solution and hence, project continued with the Pendolino tests. The first test runs with the prototype DVS were successfully completed and in the next phase, the project was extended with Bosch Rexroth who had the competence and capability to provide required technology in industrial well proven, state of the art products and system solutions.

2. DIGITAL HYdraulICS

The term “Digital hydraulic” refers to the use of fast switching hydraulic valves to achieve and even surpass the performance of today’s widely used proportional valves (servo valves or proportional valves), to provide continuous control of machine functions [2]. The valves switch so fast – via a triggered threshold approach – that the resultant flow response emulates proportional closed loop control capabilities, i.e. motion sequences and/or a desired force control profiles. The highly dynamic valve response provides functionality comparable to systems using conventional analogue command proportional valves and support the digitalization of the industry.

In the area of system sizing and configuration of hydraulic valve controlled axis and drive systems, there are two common approaches. Either the classic 3/2- or 4/3 directional valves, in which all control edges are mechanically linked via the valve spool, or individual single and multiple valves per control edge. For system architectures utilizing these so-called Autonomous Spools – fast switching 2/2 directional valves functioning...
as separate Meter in / Meter out (MI/_MO) elements — significant energy savings can be realized compared to conventional systems [3].

Increase of reliability by means of DVS is based on two aspects. Firstly, the digital base components are on/off-type, that is, simple. Simple components are often more reliable than complex ones. Also in case of hydraulic valves, oil cleanliness requirement are higher on proportional or servo valves than on/off-valves which is reliability issue if proper filtration level is difficult to achieve. Secondly, digital valve system is considered reliable because of parallel connection on/off-valves [Figure 2]. A single bit fault can be compensated by intelligent controller which detects faults [4].

With digital hydraulics the smallest unit of flow is through the 1st bit. Through multiplying parallel architecture of the single bit valves an extremely digitalized modular structure is possible. Recent fast switching valve hardware development provided the groundwork for a wide variety of solutions and different applications to be achieved and optimized. Even when modern applications put the emphasis on mostly so-called “single bit circuitry”, the examples presented prove that “multi bit circuitry” (parallel structures) can provide solutions with further unique advantages [5].

3. TILTING SYSTEM

The Tilting system is activated when the train velocity exceeds set speed of 80 km/h. There are in the market basically two different ways for the upper level control system of the train to achieve information of the incoming curve. A map based system utilizes stored track data and based on the positioning system, tilting is controlled. Another way, which is used in the Pendolino, is based on track measurements. In Finland, there is always inclination of the tracks in the curves which is detected in the train before the curve starts. The front car detects the inclination with the gyroscope and then in the curve, tilting is controlled based on the speed signals of the tachometer, acceleration transducers, gyroscope and tilt angle transducer. Total tilt of the car is sum of the track inclination and active tilting system output.

3.1. Original tilting system

Each car of a Pendolino train has own individually controlled tilting unit. Main components of the tilting system are introduced in Table 1 and a simplified hydraulic schema in the Figure 3. Tilting system has four...
single acting cylinders at the corners of a carriage and the cylinders are mechanically linked together in both bogies. The tilting mechanism is based on a four-bar-linkage.

Table 1. Key components of the tilting system

<table>
<thead>
<tr>
<th>Component</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric motor</td>
<td>7.5 kW</td>
</tr>
<tr>
<td>Variable displacement piston pump</td>
<td>20 liter/min @ 970 RPM, max working pressure 18.5 MPa.</td>
</tr>
<tr>
<td>4/3 directional servo-valve with mechanical position feedback</td>
<td>Flow capacity 150 liter/min at 7 MPa pressure differential</td>
</tr>
<tr>
<td>Cylinders</td>
<td>Diam. 90 mm, stroke 140 mm</td>
</tr>
<tr>
<td>Pressure accumulators</td>
<td>V_{\text{rom}} 2 x 18.1 liters</td>
</tr>
<tr>
<td>Filters</td>
<td>Pressure filter 3 μ</td>
</tr>
<tr>
<td></td>
<td>Return filter 5 μ</td>
</tr>
<tr>
<td>Pipes</td>
<td>30 x 3 mm steel pipes for each cylinder. Length can be even 8.5 meters between cylinder and HPU.</td>
</tr>
</tbody>
</table>

Figure 3. Simplified hydraulic schema of the original tilting system

All other hydraulic components than cylinders and pipes are in the hydraulic power unit which is located underneath a car almost in the middle. It is an own module which can be swiftly removed and installed because all hydraulic and electrical connectors are quick couplings. The tilting cylinders are in the bogies and there is up to 8.5 meters pipe and a couple of meters hose from the servo valve to the cylinders.

The original system has two 4/3 servo valves for reliability/redundancy reasons (see Figure 3). If one servo valve has malfunction, the other servo valve can be activated. However, this cannot be completed on the run. Malfunctions of the servo valves have occurred regularly and the main individual reason may have been based on oil contamination and ambient temperature. In addition, mechanical position feedback has zero drifting and it may cause unwanted tilting performance.
Accumulators have been sized according to requirement that four full tilting operations (8") must be able to complete even if the electric motor has accidentally lack of power. Original hydraulic system does not have an accumulator safety block, which is nowadays required.

3.2. New digital hydraulic tilting system

Taking into account the constraints of mounting space, installation requirements, connections, power, tilting function, etc, and of achieving the desired reliability enhancement and reduction of the maintenance costs, a hydraulic retrofit system kit has been developed (Figure 4). Guided by the principle of system simplification by consistent utilization of existing modular digital hydraulics and admirably achieves the design goals.

To create a compact and easy to maintain control manifold, 4/3 directional operated poppet valves were chosen, which are combined in a modular fashion, using standard valves all in parallel on a kind of bar manifold, easily accessible and easy to replace at any Finnish train service depot.

Original tilting system requires logic valves, but the utilization of poppet valves in the DVS and the new system architecture allows that additional logic valves are not needed anymore. This simplifies drastically the controller section of the power packs (Figure 5).

Figure 4. Simplified hydraulic schema for the new digital hydraulic system.

Figure 5. A CAD model and picture of the hydraulic power unit (HPU) including DVS.
The required improvement for reduced down time results from the inherent features of the poppet valves, which have, in contrast to the servo valves, no close tolerance parts and provides relatively large annular orifice clearance upon activation. Contamination, which find its way into hydraulic systems and/or because of thermal shock due to extreme rates of temperature change (especially in the winter seasons), cause no harm to these valves, as their design and construction are naturally fault tolerant.

The flow rate volume gradation of the valve bank is selected in such a way, that a single valve’s failure to operate properly will not affect nearby valves nor cause system failure, but will instead be automatically compensated for. The operation of the train will not be affected and the suspect valve (identified by the valve bank controller) valve could be replaced at the next possible service interval.

A part of the control scheme of fast switching valves is the so called “boost function”, in which the solenoid is safely overdriven for a very short time and in a well controlled manner, and thereby yields a very repeatable high dynamic response.

This alternative to sophisticated analog hydraulics using the simplification methods of digital electronics requires a - admittedly necessary - local controller to be used. This brings advantages, which includes not only the above mentioned sequencing and control functions, but also makes a system diagnostic structure possible, which wasn’t conceivable before. Such a controller structure could record and manipulate operating variables and alleviate problems (temperatures, pressure, valve switching errors, etc.), besides reporting/协调 responses with the train’s central control system.

4. MEASUREMENT SYSTEM

The maximum allowed track forces and passenger comfort factor determines the car tilting system behaviour. The factor takes into account a few things, but the most important in this case is horizontal acceleration, that is, “side way g-forces”. The ultimate safety related limitation is 2 m/s² which also determine maximum speed on the rail if tilting system is not functional for some reason. During operation the tilting system therefore compensates horizontal acceleration at maximum 1.5 m/s², which is achieved by tilting the car by ±8 degrees. To be able to compare the performance of the new and the original hydraulic power unit, several variables had to be measured. All the required signals were recorded with a dSpace 1103 measurement system, installed into a desktop computer.

Instrumentation was based on both separately installed and existing train instruments. Train tilt angle for instance was read from the high level train system but the accelerations (to 3 directions) and angular velocities (to 3 directions) of the car were measured by separate inertial measurement units (IMU). In addition, separate accelerometers with higher accuracy were applied during some of the tests. Bogie acceleration was measured by a transducer located at the center of a wheel. Locations of the IMUs are presented in Figure 6.
The hardware architecture of the data logging system is presented in Figure 7. The analogue voltage signals of the separate accelerometers, train speedometer, tilting angle sensor and the servovalve control signal were measured by the analogue input channels of the dSPACE system. A GPS receiver was connected to the serial port of the dSPACE system for linking the measurements to actual curves of the track. Both velocity and location messages were logged.

![Diagram of data logging system](image)

**Figure 7.** Data logging system. Upper row signals are read from the train system, GPS and IMUs are separate modules and Hydraulic signals are routed from the hydraulic unit via CAN cable.

5. RESULTS

The first measurements were done with servo valve system and the data was used for reference and also for model verification purposes. Also the project group gained experience of the Pendolino train, its control system and railways traditions and rules to be taken account in the project.

In Figure 8 behaviour of the tilting system when driving into curves is presented. The measured dataclip is from the track between Toijala and Viiala, about 30 km south of Tampere. As can be seen from the presented measurements, horizontal acceleration at the bogie is varying at much higher amplitude than in the car. Horizontal acceleration felt by a passenger remain smaller than $0.5 \text{ m/s}^2$. Theoretically, the system could compensate horizontal acceleration completely, but this would lead passengers to get "train sick". When tilting system was turned off, the Bogie and Car horizontal accelerations overlapped. Safety regulations give maximum allowed value for horizontal acceleration to be $2 \text{ m/s}^2$. At that high level items are sliding down from tables and sitting straight requires some muscle effort, but derailing would require much higher values.
Horizontal acceleration

Car tilting angle

Car tilting angular velocity

Servo valve command signal

Train speed & System pressures

Figure 8. Measurement with a servo valve system in the curves between Toijala and Viiala, about 30 km south from Tampere.
5.1. Depot measurements

Initial Bosch Rexroth solution was tested firstly in the depot. Two cars of the train were implemented with digital hydraulic tilting systems. Two cars were instrumented and a comparison between new digital valve system and original servo system was completed. The test signals were given to cars individually and references were manually triggered from the train control panel. The train system allows tilt testing from rest position at constant angular velocity to certain end position. Different velocities were tested. An example of the results is presented in Figure 9. The uneven delays are caused by manually given references. Behavior of the systems during motion is the point of interest.

![Figure 9. Depot measurement at the angular velocity of 3 °/s](image)

Depot tests do not offer general view of the system performance, but the conclusion was that the prototypes provided by Bosch Rexroth are functional and retrofittable.

5.2. Re-analysis of cooler requirement

Some of the Pendolino challenges are linked to winter conditions, but there have been also overheating issue in summer time. The leakage of the servo valves and resulting pumping to accumulators was estimated to be the reason for most of the energy consumption in the system. The digital valves are leak free and therefore it was expected that the new system would not require cooler at all. To get a confirmation for this hypothesis, a long term temperature measurements were done during summer.

Temperature measurements were made with the Elcolog “thermo buttons” which were installed into the power pack’s tank wall and insulated. Measurement period was two weeks and the sample time was 10 minutes. Tank temperature was measured in the car 1, 2 and 4. Cars 1 and 2 are equipped with the redesigned power pack with the digital valve system (DVS) whereas the car 4 has the original power pack.
with the servo valves and variable displacement pump. In the car 1 cooler was de-activated with a shut-off valve and in the car 2 cooler was operating normally. The power pack of the car 4 was kept in the original form with the cooler. Results of the temperature measurements are shown in Figure 10.

![Figure 10. Temperature measurements with Digital Valve System (DVS) and with the original servo valve. Measurements were completed in June 2013.](image)

During the measurement period shown in Figure 10 there was a hot season (highest day temperatures over 25°C) all over the Finland in the end of June. The results show that the cooling indeed lowers tank temperature. However, it is also clear that temperatures are still in the acceptable level even in the system without the cooling. Temperatures of the digital tilting system without the cooling are approximately in the same level as in the original servo system with the cooling.

6. CONCLUSIONS

The original tilting unit of a Pendolino train was replaced with a novel technology based prototype. The test runs proved that the prototype designed at Tampere University of Technology provided required level of travel comfort. In the next phase, market-proof tilting unit from Bosch Rexroth was tested on both at standstill in the depot and on rails.

Test results and practical experiences of the commercial Digital Valve System have proven that the substitution of servo valve with DVS is possible in described application. Controllability is good and it is expectable that the design based on the on/off-valves is more reliable than the original control system based on the sensitive servo valves. New tilting units from Bosch Rexroth was running on revenue service for about two years and during this period reliability problems concerning the tilting system have not been reported. The new DVS based tilting system is now replacing the original system in the entire Finnish Pendolino fleet.

The R&D project included a technology requirement, latest knowledge from academia and technology provider but also a supervising agency. VR required a more reliable technology for the current used tilting system; TUT had the idea of an innovative hydraulic solution and Bosch Rexroth had the competence and capability to provide required latest digital hydraulic technology in industrial well proven, state of the art products and advanced system solutions for the railway industry.
REFERENCES


HYBRID LOAD SENSING – DISPLACEMENT CONTROLLED ARCHITECTURE FOR EXCAVATORS

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ABSTRACT

Energy saving is very important due to the shortage of energy resources and the need to protect the global environment. Manufacturers of mobile machinery have responded to this need by developing more efficient hydraulic systems. Examples of such improved architectures are displacement controlled or transformer systems, which aim to minimize throttling losses. To further improve efficiency a new hybrid architecture, which combines the advantages of load sensing with displacement control, while at the same time eliminating their inherent disadvantages is proposed. The new hybrid architecture for hydraulic excavators is introduced in detail and the potential for efficiency improvement is shown by using simplified load models and a modelled digging cycle.

KEYWORDS: Energy saving, Excavator, New hybrid system design

1. INTRODUCTION

Energy saving becomes very important considering finite nature of energy resources and the fact that greenhouse gases need to be reduced. This also holds true for the hydraulic equipment industry, which faces considerably increasing market demands for energy saving. It is reported that hydraulic excavators are rather inefficient, they effectively convert only 10% of the consumed fuel's energy into mechanical work [1]. Under the circumstances, hydraulic excavators are showing improvements in energy efficiency; for example hydraulic excavators that use an electric swing drive instead of a hydraulic appeared recently on the market, and they achieve approximately 20% reduction of energy consumption [2]. In Europe and North America, in particular, research is actively conducted for energy efficiency improvements based on digital hydraulic systems [3], load-sensing systems, displacement pump controlled systems [4], transformer systems [5], etc.

The authors propose a new hybrid hydraulic architecture for excavators, which improves energy efficiency. As a combination of the different hydraulic systems mentioned above, it aims to utilize the advantages and eliminate the disadvantages of each system. This paper first shows the energy efficiency characteristics of a load-sensing system, displacement pump controlled system, and hybrid system by using a simplified load model compatible with both cases where the number of loads is one and two. Then, with a hybrid system designed for hydraulic excavators, a simulation is conducted to estimate the energy efficiency during actual digging work.
2. SIMULATION WITH SIMPLIFIED MODEL

2.1. Investigated system architectures

This paper shows comparisons of the energy efficiencies between the hybrid hydraulic system and the other two different hydraulic systems, which are mentioned in following sections, utilizing the simulation environment DSHplus. To predict the system efficiencies, a fuel consumption of diesel engine (left) and an efficiency map of variable displacement unit (right) are used, which are shown in Figure1. In the simulations, typical valves are modelled. The rated power of the diesel engine is 125 kW and the engine rotation speed is controlled to be constant at 1750 min$^{-1}$. Each variable displacement unit offers a displacement volume of 160 cm$^3$.

![Figure 1. Fuel consumption of Diesel engine at 1750 min$^{-1}$ and Efficiency map of variable displacement unit](image)

2.1.1. Load-sensing system

Figure 2 shows the load-sensing system (LS) [6]. It is a valve-controlled system, designed to provide improved compatibility with diversified attachments and to save energy, which is considered to have excellent controllability particularly in simultaneous operations compared with open-center type systems common in the Asian markets.

![Figure 2. Load-sensing system](image)
As Figure 2 shows, this system is equipped with a throttle valve (3) and a pressure compensator (2) before load valve (5). The pressure compensator keeps the pressure difference $\Delta p_{LS}$ constant. This differential pressure is referred to as the differential load-sensing pressure. Irrespective of the load level, the flow rate to the load valve is proportional to the opening area of the metering valve, resulting in excellent operational performance. The system sends the maximum load pressure detected at the shuttle valve (4) to the LS controller. This controller controls the pressure $p_p$ from the Load-sensing pump (1) to be $\Delta p_{LS}$ higher than the maximum load pressure. As $\Delta p_{LS}$, this paper uses 20 bar.

2.1.2. Displacement pump controlled system

One of the solutions for eliminating losses caused by the metering valves in series with the pressure compensator used in traditional hydraulic systems is a displacement pump controlled system (DC) in a closed-circuit configuration that uses variable displacement units to directly drive actuators [7]. See Figure 3. The variable displacement units (1) are directly connected to the rotation axis of the diesel engine. This system can directly control the flow direction of the fluid, and therefore requires no directional control valve. This is one of the advantages that the system offers. On the other hand, it uses single-rod cylinders in the closed circuit, causing an insufficient or excessive flow rate at the intake side of each variable displacement unit due to the difference in area between the piston and rod side of the cylinder. To avoid this matter, it requires pilot operated check valves (2). These valves are operated by the pressure of the high pressure line. When the pilot pressure exceeds the set value, the check valves are opened and the low pressure line is connected with the reservoir line, then the shortage of the flow rate is supplied and the excess flow rate is released. As all actuators are connected to the pumps, energy recuperation is enabled, for example, when the boom is lowered. The maximum pressure of the circuit is determined by the pressure relief valves (3). When the set value is exceeded, the system lets hydraulic oil escape to the reservoir line through the check valves (4) to protect the hydraulic circuit.

![Figure 3. Displacement pump controlled system](image)

2.1.3. Load-Sensing and Displacement pump controlled hybrid system

While offering advantages of producing no loss caused by throttle valves and providing high energy efficiency, the DC requires a high cost because it basically requires one variable displacement unit for each actuator. On the other hand, the LS is relatively low in cost because it can use a single pump to drive more than one actuator. For this reason, the authors propose a load-sensing and displacement pump controlled hybrid system (LS+DC). Figure 4 shows an example of an LS+DC. This system also uses more than one
variable displacement unit. According to their roles, the left and right variable displacement units are called the main pump (1) and secondary pump (2), respectively. In this simulation model one secondary pump is used, however the number of secondary pumps basically depends on the operating conditions and costs of the target machine.

In this simulation, when the number of loads is one, the hydraulic fluid to the load is supplied by the secondary pump, the system functions as a DC. At the same time the main pump charges the hydraulic accumulator (4) through the right position of the switching valve (3) and ensures the diesel engine is loaded at a higher torque and therefore in a region of improved efficiency. This operating condition is named DC mode.

When the accumulator is full, the clutch (5) between the diesel engine and secondary pump is disconnected. Subsequently, the main and secondary pumps temporarily produce a hydraulic transformer-like system, with the high pressure accumulator used instead of the diesel engine as the energy source. This operating mode is named TFL mode. When the energy stored in the accumulator becomes low, the system operates in DC mode again. Although it is possible to operate the engine with a lower rotation speed or completely stop it in TFL mode, the rotation speed was maintained at 1750 min\(^{-1}\) in the simulation. The fuel consumption rate for this operation mode is approximately 1.3 liter/h.

For two loads the system changes, the position of the valve (3) and at the same time the main pump functions as a Load-sensing pump to supply hydraulic fluid to the load. The right circuit acts as a DC. This condition is named LS mode.

![Figure 4. LS+DC Hybrid System](image)

2.2. Simulation results

To study the energy efficiency characteristics of each hydraulic system, simulations were performed with two assumptions: the first one where one actuator is activated and the second one where two actuators are activated at the same time. These simulations did not take into consideration the power required to activate the valves or all electric power to be used for control, for example.

2.2.1. Single Load

First, the static efficiency characteristics of each hydraulic system for a single load are analyzed, when the load pressure and flow rate are constant. The maximum load pressure \(\Delta p_{\text{max}}\) is 300 bar and the maximum
flow rate \( Q_{max} \) is 240 liter/min. The load pressure \( \Delta p_A \) and load flow rate \( Q_A \) of Load A is defined as the ratios to the respective maximum values by equations (1) and (2). The simulations are based on the assumption that both load pressure ratio \( \gamma_{\Delta p} \) and load flow rate ratio \( \gamma_Q \) are 25%, 50%, 75%, or 100%. If \( \gamma_{\Delta p} = 50\% \) and \( \gamma_Q = 50\% \), for example, then the load pressure is 150 bar and the load flow rate is 120 liter/min i.e. the output of the hydraulic system is 30 kW. This is equivalent to approximately 25% of the rated output of the diesel engine.

\[
\gamma_{\Delta p} = \frac{\Delta p_A}{\Delta p_{max}}
\]

(1)

\[
\gamma_Q = \frac{Q_A}{Q_{max}}
\]

(2)

\[\begin{array}{c}
\Delta p_A Q_A \\
\Delta p_B Q_B \\
p_p Q_p \\
\omega_WM_w \\
H_D Q_F \\
\end{array}\]

\[\begin{array}{c}
\eta_C \\
\eta_P \\
\eta_D \\
\eta_T = \eta_C \eta_P \eta_D
\end{array}\]

Figure 5. Definition of energy efficiency coefficients

As Figure 5 shows, the energy efficiency is determined by calculating the hydraulic circuit efficiency \( \eta_C \), pump efficiency \( \eta_P \), and diesel engine efficiency \( \eta_D \). These efficiencies are calculated using equations (3), (4), and (5). The overall efficiency \( \eta_T \) is calculated by equation (6).

\[
\eta_C = \frac{\int \Delta p_A Q_A + \int \Delta p_B Q_B}{\int p_p Q_p}
\]

(3)

\[
\eta_P = \frac{\int p_p Q_p}{\int \omega_WM_w}
\]

(4)

\[
\eta_D = \frac{\int \omega_WM_w}{\int H_D Q_F}
\]

(5)

\[
\eta_T = \eta_C \eta_P \eta_D
\]

(6)

In these equations, \( p_p, Q_p, \omega_w, M_w \) and \( H_D \) represent the pump pressure, pump flow rate, shaft rotation speed, shaft torque, and fuel calorific value (42.6 MJ/kg), respectively. \( Q_F \) represents the flow rate of the fuel supplied to the diesel engine.

Figure 6 shows the hydraulic circuit efficiency \( \eta_C \) of each system. From the left, they are for the LC, DC, and LS+DC. The horizontal and vertical axes of the graphs represent the load pressure ratio \( \gamma_{\Delta p} \) and hydraulic circuit efficiency \( \eta_C \), respectively. Irrespective of the load level, the DC and LS+DC systems deliver efficiencies of nearly 100% because the discharge pressures and flow rates of their variable displacement units are almost equal to the respective load pressures and load flow rates. On the other hand, the LS system delivers lower circuit efficiency when the load pressure ratio is low. This is largely because the differential load-sensing pressure \( \Delta p_{LS} \), which produces a loss when the fluid through the throttle valves, has a relatively large effect.

Figure 7 shows the pump efficiency \( \eta_P \) of each system. The LS is the best in pump efficiency. Generally, the higher the load is, the higher the pump efficiency is. The DC exhibits higher hydraulic circuit efficiency \( \eta_C \) than the LS and therefore the load on the variable displacement unit of the DC is lower than that of the LS, which seems to cause a lower efficiency. As shown in Figure 3, two secondary pumps are installed in DC,
however the idling losses of non-operational secondary pump were not considered in this result. The LS+DC is operated in TFL mode after the accumulator is charged fully. In this mode, the hydraulic energy inside the accumulator is sent to the load through the transformer-like system consisting of the main and secondary pumps. This process requires two energy form conversions, from the hydraulic energy to mechanical energy and then back to hydraulic energy, lowering the efficiency. In the low output area, in particular, the pump efficiency is as low as 50% or less.

Figure 8 shows the diesel engine efficiency $\eta_D$ of each system. The LS+DC exhibits the highest diesel engine efficiency. In DC mode, it operates with the almost maximum diesel engine efficiency. In TFL mode, it disconnects the clutch to idle the engine with the rotation speed maintained and therefore can keep a very high efficiency regardless of the load level. This result indicates that for the LS+DC, the performance of the variable displacement unit used in the system has a very large effect on the efficiency.

Figure 9 shows the overall efficiency $\eta_T$ of each system. Because it does not clearly show the differences between the systems, Figure 10 is used to show the efficiencies of the DC and LS+DC as ratios to that of the LS. The DC always exhibits a higher overall efficiency than the LS and this advantage decreases with an increase in load. On the other hand, the LS+DC exhibits the highest efficiency when it operates with about 25% or less of the maximum load and increases its advantage as the load becomes lower. However, at a high load of 65 kW or more, it shows the lowest efficiency among the three systems. This is because it cannot compensate for its low pump efficiency $\eta_P$ with the other efficiency advantages.
2.2.2. Double Loads

A typical hydraulic excavator has six actuators: cylinders for driving the boom, arm, and bucket; a motor for turning the upper structure; and right and left motors for driving the hydraulic excavator. When it is in action, basically two or more of them operate at the same time. For this reason, simulations are conducted that involve simultaneous operations by two actuators to estimate the energy efficiency characteristics of each system.

In addition to the hydraulic systems which are used for the single load simulations, this chapter adds a hybrid hydraulic architecture (LS+2DC), shown in Figure 11, is added for the simulations. This system is basically the same with the LS+DC except that it does not operate in LS mode when it supplies hydraulic fluid to the two loads because it is equipped with two secondary pumps, i.e. no losses occur, which are produced by the flow rate distribution at the LS-system. The simulations are intended to reproduce two different operating
conditions. Simulation 1 is for a case where the total output power of the actuators involved in the simultaneous operation is 25% or less of the rated power of the diesel engine, and Simulation 2 is for a case where the total output is between 20 to 80%.

For Simulation 1, the Load A pressure is \( \Delta P_A = 150 \text{ bar} \) and the total load flow rate (sum of \( q_A \) and \( q_B \)) \( \sum q = 120 \text{ liter/min} \) are used. For Simulation 2, \( \Delta P_A = 300 \text{ bar} \) and \( \sum q = 200 \text{ liter/min} \) are used. The ratio between the pressures \( \Delta P_B \) and \( \Delta P_A \) supplied to Load B is defined as \( \beta_{dp} \) through equation (7), and the ratio between the total flow rate and flow rate supplied to Load B is defined as \( \beta_q \) through equation (8). In these simulations, \( \beta_{dp} \) are 20%, 40%, 60%, or 80% and \( \beta_q \) are 25%, 50%, or 75% with \( \Delta p_B < \Delta p_A \).

\[
\beta_{dp} = \frac{\Delta P_B}{\Delta P_A} \quad (7)
\]

\[
\beta_q = \frac{Q_B}{\sum Q} \quad (8)
\]

Figure 12 shows the overall efficiencies of LS. The horizontal and vertical axes of the graphs represent \( \beta_{dp} \) and \( \eta_T \), respectively. The overall efficiencies of the DC, LS+DC and LS+2DC represented as ratios to the overall efficiency of the LS are shown in Figure 13. Figure 14 shows the results of Simulation 2.
Compared with the single-load case, each system exhibits much higher energy efficiency than the LS. This is because the LS causes a loss at the pressure compensator according to the pressure difference between the two loads and the distribution rate of each flow rate if the LS pump supplies hydraulic fluid to more than one load. If, for example, the load pressure difference is large and the flow rate at the low pressure side is large, then a very large loss is produced.

For Simulation 1, the LS+2DC exhibits the highest efficiency even if $\beta_{ap}$ and $\beta_q$ are changed, and its advantage increases as the load becomes lower. The LS+DC shows the lowest overall efficiency among the three systems but the difference compared with DC is small. For simulation 2, DC delivers the highest efficiency, followed by the LS+DC and then the LS+2DC.

The simulation results show the following characteristics of the LS+DC. In DC or TFL mode, it exhibits the best energy efficiency within the low-output area regardless of the number of loads. In LS mode, it provides a considerably better efficiency than the LS so long as the main pump is supplying hydraulic fluid to only a single load as a Load-sensing pump. However, if the main pump supplies hydraulic fluid to more than two loads, it may produce a large loss as the LS does.

3. SIMULATION OF DIGGING CYCLE

To estimate the characteristics of the LS+DC, a simulation is performed to simulate a case where actual digging work is done. The model of this simulation is 20 ton class hydraulic excavator. As with the simulations mentioned above, the diesel engine offers a rated power of 125 kW and the rotation speed is controlled to be constant at 1750 min$^{-1}$.

3.1. LS+DC architecture for excavators

First, the number of variable displacement unit for the LS+DC to be used for the hydraulic excavator is determined. Figure 16 (left) shows the ratio of the number of actuators that operate simultaneously per digging cycle based on typical digging cycle shown in Figure 15 [8]. It indicates that nearly half of the digging time simultaneous operations by two actuators occur. Figure 16 (right) shows the combinations of actuators for simultaneous operations by two actuators, indicating that the combination of the boom and swing actions dominates others.

![Figure 15. Digging cycle](image)

This study uses the main pump (1) and two secondary pumps (2) and (3) as shown in Figure 17. The pump efficiency map shown in Figure 1 is used for these three pumps. This system operates in DC or TFL mode when one or two actuators are operated at the same time. When three or more actuators are activated at the same time, the systems enters LS mode where the main pump functions as the Load-sensing pump. When four actuators are activated at the same time, the LS pump supplies hydraulic fluid to two actuators, which
may cause a very large loss. This study accepts this loss because its occurrence is only 14%. This system can use directional control valves (7) to switch the actuator to be connected to the secondary pump. In this architecture, the boom and bucket cylinders share a secondary pump and the swing motor and arm cylinder share the other secondary pump, with the boom cylinder and the swing motor given priority to use the secondary pump because they are expected to recuperate large energy. In DC or TFL mode, it uses the main pump to accumulate regenerative energy in the high pressure accumulator. In LS mode, the regenerative energy is basically used to support the diesel engine; however, when the power during the energy regeneration process is large, the system uses the charge pump (4) and the directional control valve (5) to accumulate energy in the high pressure accumulator.

*Figure 16. Rate of the simultaneous operation (left) & Combination of two simultaneous operated actuators (right)*

*Figure 17. LS+DC system configuration for excavator*
3.2. Simulation results

Figure 18 shows the stroke or angle for each actuator and input/output energy when the new hybrid system is used to perform digging. All actuators were operated almost within the target displacement, however they did not satisfy the target stroke when the operating mode has changed from DC or TFL to LS. This situation must be improved in the future.

It is not possible to accurately calculate the energy efficiency of the LS+DC without considering the energy balance within the high pressure accumulator. For this reason, this study defined one cycle as the duration for which excavation begins with the accumulator full and ends with the accumulator full again. One cycle time is approximately 13.5 seconds, which is 0.5 seconds longer than the digging cycle time.

In one cycle, the fuel consumed has a calorific value of 3094 kJ, the diesel engine output energy of 1176 kJ, the pumps output energy of 867 kJ in total, and the actuators consumed energy of 656 kJ in total. With this simulation, the efficiencies for excavation were determined by calculation as follows: hydraulic circuit efficiency $\eta_C = 75.7\%$, pump efficiency $\eta_P = 73.7\%$ and diesel engine efficiency $\eta_D = 38.0\%$. They resulted in an overall efficiency $\eta_T = 21.2\%$.

4. CONCLUSIONS AND OUTLOOK

For higher energy efficiency of hydraulic excavators, a new hybrid system is proposed. Using a simplified load model, the energy efficiency characteristics between a load-sensing system and displacement pump controlled system were compared. This comparison revealed that the hybrid system proposed, when operating in DC or TFL mode, is advantageous during low power output operation regardless of the number of loads. In LS mode, it provides a better efficiency than a standard LS system as long as the main pump is supplying hydraulic fluid to only one load.

Then, based on the results of the simulations using the simplified model and the measurement data from actual excavation work, a hybrid architecture for hydraulic excavators was designed.
A simulation model for that architecture was made to estimate the energy efficiency for digging work. The calculated overall efficiency $\eta_T$ is 21.2%.

For higher accuracy, future simulations will include loads such as electric generator and cooling fans driven by the diesel engine. In addition, simulation models will be made for a load-sensing system and displacement pump controlled system to compare the efficiencies.

References


HYBRID PUMP DRIVE

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ABSTRACT

The number of applications with rotational speed controlled fixed hydraulic pumps is increasing because they offer good controllability and efficiency compared to conventional fixed speed pump drives. Both electric motors and pumps are dimensioned according to maximum power/torque. Consequently, the electric motor is significantly larger than the hydraulic pump. This paper introduces a double pump system with connection to additional hydraulic supply creating a hybrid drive. The power/torque demand can be reduced making it possible to choose a smaller electric motor without sacrificing performance while increasing power density. The connection to two energy sources results in a flexible energy supply and makes it possible to recover either hydraulically or electrically. Simulated results show that the studied circuit enables electric motor size reduction as much as 50% when the bottom of the cylinder is the load side and the second pump is connected to the accumulator or pressure source. This is applicable in applications where the rod side loading is remarkably lower as well as in other applications that have a minor extending load.

KEYWORDS: Speed controlled cylinder drive, electric motor, hybrid drive, energy recovery

1. MOTIVATION

The hype and development of electric hybrids have increased interests in electric drives and there have been numerous discussions concerning the application of electric drives versus hydraulic drives during the last decade. The interest in the electric drives is the result of the demand for efficiency, controllability and, service-free operation and these features are usually connected to electric drives. As a result, electric drives have replaced hydraulic in some applications, such as the steering systems of passenger cars. However, the power density of the electric drive components is a weakness of those type drives, and that drawback is especially a concern in linear drives where hydraulic drives are still superior to electric linear drives.

Figure 1 shows the continuous power density of the final drive components (i.e. power density of an electric generator/motor and a hydraulic pump/motor without auxiliary devices).

The answer to the question as to whether an electric drive is better than hydraulic drive is depends upon the application. The applicability depends upon the availability of the particular power source or power grid. That is often the easiest way to use same technology that is already used in the application.
In industrial applications, an electric grid is available and the use of an electric drive is state-of-the-art. A hydraulic drive is only used if its real benefits can be utilized. In mobile machines the applicability of electric drives is restricted by the power limits of the electric grid because the standard mobile machine has either a 12 V or a 24 V grid and its power transfer capability is limited to a few kilowatts. However, if the machine has an electric hybrid drive the available voltage level is higher, up to 800 V, and the actuator power can be remarkably higher. Almost without exception, mobile machines always have hydraulic drives and a hydraulic circuit or “grid” in which the actuators are connected.

If both electric and hydraulic power grids are available, the actuator can use both energy sources, as the unit is hybrid driven. The speed controlled hydraulic pump is an example of a type of unit that can be connected to electric and hydraulic energy sources. The interesting question is how to divide the energy/power demand between these two energy/power sources.

### 2. SPEED CONTROLLED CYLINDER DRIVE

The most energy efficient way to control the hydraulic cylinder is to use displacement or speed control because there are no dissipating elements between power generation and usage and it enables energy recovery with the same components. In the displacement control the cylinder motion is directly controlled by the flow that is generated by the pump. The variable flow can be realized by a constant pump with variable speed or a variable displacement pump with constant speed or variable speed drive. Several circuits have been introduced and studied [1] – [8]. The three most widely known closed circuits for a speed controlled cylinder drive are shown in Figure 2.

In all three systems, the cylinder is in either a closed or semi-closed circuit that is different from a valve controlled cylinder drive in which open circuit is usually applied.

System A has two pumps and both are connected to the bottom of the cylinder. The right pump is situated between the rod and the bottom of the cylinder. The size of the pumps can be chosen in such a way that the ratio of the sum of the displacements to the displacement of the right pump is equal to the area ratio of the cylinder. If the area ratio of the cylinder is two, the nominal size of the pumps is the same. The left pump can be an open circuit pump, but the right pump should be a closed circuit pump.

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**Figure 1. Power density of electric motors and hydraulic units.**  
*Source Bosch Rexroth, Parker, and ABB. Continuous power is used for calculation.*

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<th>Basic electric motor</th>
<th>Liquid cooled permanent magnet motor</th>
<th>Constant displacement hydraulic unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>2</td>
<td>4</td>
<td>6</td>
</tr>
<tr>
<td>Variable displacement hydraulic unit</td>
<td>Constant displacement hydraulic unit</td>
<td>Basic electric motor</td>
</tr>
</tbody>
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<table>
<thead>
<tr>
<th>Power density kW/kg</th>
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<tr>
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</tr>
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<td>2</td>
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<td>3</td>
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<td>4</td>
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<td>5</td>
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<td>6</td>
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</tbody>
</table>
System B has two pumps that are driven by one electric motor. The left pump is connected to the bottom of the cylinder and the right pump is connected to the rod. The size of the pumps can be chosen so that their displacement ratio is equal to the area ratio of the cylinder. If the area ratio of the cylinder is two, then the displacement of the left pump (bottom) is double the displacement of the right pump. Both pumps can be open circuit pumps if the suction requirements are fulfilled at high rotational speeds.

System C has only one closed circuit pump and it is between the bottom of the cylinder and the rod. The flow excess or shortage is handled by the valves that connect the low pressure side to the supply pressure system. This requires either an accumulator or a supply pressure pump, the size of which depends upon the area ratio of the cylinder. If the area ratio of the cylinder is two, then the supply pressure pump has to be at least 50% the size of the main pump.

In systems A and B, the area ratio of the cylinder defines the size of the pumps that are used and the goal is that the displacement ratios are as close as possible to the area ratio of the cylinder. The available pump sizes restrict the realization, and in many cases there will be some differences. During operation, the actual flow of the pumps differs from the theoretical flow because leakages and compressibility cause a discrepancy between the cylinder and the pump flows. For this reason, all three circuits require some kind of supply pressure system to cover leakage losses and to compensate for the discrepancy between the pump sizes and the area ratio of the cylinder. Energy recovery is a built-in feature in all three circuits and the load pressure rotates the hydraulic unit, which acts as a motor, and the energy is transformed into mechanical energy and then into electricity.

For this study, system A was chosen because it enables the connection to a high pressure hydraulic line or accumulator and its total installed displacement volume is lowest of three systems.

3. DOUBLE PUMP DESIGN FOR THE CYLINDER DRIVE

In system B, shown in Figure 3, the cylinder motion control is realised through rotational speed control of the pumps.

The required torque for the electric motor is the result of the pressure difference across the hydraulic units

\[ T_e = (p_1 - p_a) \cdot V_{g1} + (p_1 - p_2) \cdot V_{g2} \]

in which \( p_a \) is the pressure of the accumulator or pressure of the pressure source of the left pump, \( p_1 \) is the pressure of the bottom of the cylinder, \( p_2 \) is the pressure of the rod side of the cylinder. \( V_{g1} \) is the displacement of the left pump and \( V_{g2} \) the displacement of the right pump. The supply pressure is assumed to be remarkably lower than the load pressure and it is therefore neglected in the calculations.
Let’s assume that the area ratio of the cylinder is $R$ and the displacements of the hydraulic units are chosen so that they fulfill the requirement

$$R = \frac{A_{\text{bottomside}}}{A_{\text{rodsi}de}} = \frac{V_{g1}+V_{g2}}{V_{g2}}$$

(2)

Now the torque demand is

$$T_e = (p_1 - p_a) V_{g1} + (p_1 - p_2) \frac{V_{g1}}{R-1} = \left( (p_1 - p_a) + (p_1 - p_2) \frac{1}{R-1} \right) V_{g1}$$

(3)

Moreover, the rated power of the electric motor should be equal to the continuous power required by the hydraulic pumps. The electric motor should work as a generator if the energy recovery function is needed. Figure 4 shows the operation quadrants of the system.

In the first quadrant (1Q) the load induces pressure on the rod side of the cylinder and the cylinder motion is outward. Fluid flows from the rod side towards the right pump and through it to the bottom side of the cylinder. The right unit acts as the motor generating the torque that is used to drive the left pump that also delivers flow to the bottom side of the cylinder. The torque that is not used by the left pump rotates the electric motor that acts as the generator and feeds electricity back to the grid.
In the second quadrant (2Q) the load induces pressure on the bottom side of the cylinder and the cylinder motion is outward. Both units are working as the pump and part of the fluid flows from the rod side through the right pump to the bottom side of the cylinder. The electric motor delivers the required torque.

In the third quadrant (3Q) the load induces pressure on the bottom side of the cylinder and the cylinder motion is inward. Both units are working as the motor and part of the fluid flows from the bottom side to the rod side of the cylinder through the right pump and part of the fluid goes through the left unit to the tank. The units generate torque that drives the electric motor that acts as a generator and feeds electricity back into the grid.

In the fourth quadrant (4Q) the load induces pressure on the rod side of the cylinder and the cylinder motion is inward. Now the right pump is acting as the pump and delivers flow from the low pressure side to the high pressure side and the left pump delivers the flow from the bottom side to the tank.

3.1. Energy Storage

When the accumulator is connected to the left pump replacing the tank connection, the system will be a hybrid. It has two energy sources and energy recovery is possible in two ways; electrically or hydraulically. The accumulator connections creates two closed circuits in which the left pump is between the accumulator and the bottom side of the cylinder and the right pump is between the bottom and rod side of the cylinder. The accumulator can be replaced by a connection to the hydraulic high pressure line, "grid".

The pressure \( p_a \) can be assumed to be constant if a large accumulator is connected to the left pump. The system requires separate supply pressure system, like the one in system C, because of both the hydraulic pumps are closed circuit pumps that require supply pressure.

Compared to the previous explanation for each of the quadrants, the torque demand for the electric motor changes because the pressure difference across the left pump is different depending on the pressure levels of the accumulator and the load.

In the first quadrant, the situation changes because the accumulator pressure, \( p_a \), is higher than the pressure \( p_l \), which is the low pressure side with a connection to the supply pressure. Both units are working as the motor generating torque, and the electric motor acts as generator feeding electricity back to the grid. This quadrant is not a favorable operation mode for this circuit because stored hydraulic energy cannot be used at actuator; it is transformed into electricity and losses.

In the second quadrant, the pressure level of the accumulator, \( p_a \), is reducing the energy demand because it reduces the pressure difference across the left pump. If the load pressure, \( p_l \), is below the accumulator pressure the left pump is in motoring mode, so torque is generated.

In the third quadrant, the accumulator pressure, \( p_a \), reduces the energy amount that is fed to the electric grid because part of the load energy is stored into the accumulator. If the load pressure, \( p_l \), is below the accumulator pressure the left pump is in pump mode, so torque is required.

In the fourth quadrant, both hydraulic units are working against pressure i.e. working as a pump. The left pump is delivering flow to the accumulator and the right pump is delivering flow to the load pressure side (rod side of the cylinder).

The size of the electric motor is defined by the maximum torque and the power of the pumps. A load spectrum has a great effect on the electric motor selection because the difference between the short-term loading and continuous loading is significant. The peak torque or power of an electric motor can be 2–3 times higher than the continuous torque or power. When the accumulator is added to the system, the maximum torque and power demand is reduced and a smaller electric motor can be chosen, which in turn improves the system’s power density. However, while the accumulator reduces the power density, its power density is still 10–50 times higher than the power density of an electric motor. The question is: To what extent can the size of the electric motor be reduced as a function of the accumulator pressure?
3.2. Power Ratio

When the flow comes from the accumulator (quadrants 1 and 2), the ratio between electric power and hydraulic power can be calculated as

$$\frac{P_e}{P_h} = \frac{(p_1 - p_a) + (p_1 - p_2)}{p_a} \frac{1}{R-1}$$  (4)

and when the flow is towards the accumulator (quadrants 3 and 4), the ratio between electric and hydraulic power can be calculated as

$$\frac{P_e}{P_h} = \frac{(p_1 - p_a) - (p_2 - p_1)}{p_a} \frac{1}{R-1}$$  (5)

The variations of the operating modes of the electric motor and the accumulator are different in each quadrant (Figure 5); therefore, the power demand depends upon the quadrant in which the system is operating.

![Figure 5. Operation modes of electric motor and hydraulic accumulator in different quadrants.](image)

Quadrant 1

For the load pressure effects on the rod side of the cylinder, let $r_p$ be the pressure ratio between load pressure $p_2$ and $p_a$, then $p_2 = r_p p_a$ and the ratio between electric and hydraulic power is

$$\frac{P_e}{P_h} = \left( \frac{p_1}{p_a} - 1 \right) + \frac{1}{R-1} \left( \frac{p_1}{p_a} - r_p \right)$$  (6)

Quadrant 2

For the load pressure effects on the bottom side of the cylinder and let $r_p$ be the pressure ratio between load pressure $p_1$ and $p_a$, then $p_1 = r_p p_a$ and the ratio between electric and hydraulic power is

$$\frac{P_e}{P_h} = \left( r_p - 1 \right) + \frac{1}{R-1} \left( r_p - \frac{p_2}{p_a} \right)$$  (7)

Quadrant 3

For the load pressure effects on the bottom side of the cylinder and let $r_p$ be the pressure ratio between load pressure $p_1$ and $p_a$, then $p_1 = r_p p_a$ and the ratio between electric and hydraulic power is

$$\frac{P_e}{P_h} = \left( r_p - 1 \right) - \frac{1}{R-1} \left( \frac{p_2}{p_a} - r_p \right)$$  (8)
Quadrant 4

For the load pressure effects on the rod side of the cylinder and let $r_p$ be the pressure ratio between load pressure $p_2$ and $p_a$, then $p_2 = r_p p_a$ and the ratio between electric and hydraulic power is

$$\frac{P_e}{P_h} = \left(\frac{p_1}{p_a} - 1\right) - \frac{1}{R-1} \left(r_p - \frac{p_1}{p_a}\right)$$

(9)

The results of the power ratio equations are the same for quadrants 1 and 4 and the results of the power ratio equations are also the same for quadrants for 2 and 3. Figures 6 and 7 show the power ratios as function of the area ratio of the cylinder and the pressure ratio (load pressure/accumulator pressure). The higher the power ratio is, the higher the electric power as compared to the hydraulic power. If the goal is to reduce the required electric motor power, the power ratio should be as low as possible; therefore, the area ratio should be as high as possible and the pressure ratio as low as possible. If the goal is to reduce the electric motor power level by 50% it means that the power ratio should be 1 or less.

Figure 6. Power ratio contours for the quadrants 1 and 4.

Figure 7. Power ratio contours for the quadrants 2 and 3.

The load direction has an effect on the power ratio behavior. In the extending load case, the power ratio is above 1 in all of the studied area ratios. In the retracting case, a 50% power demand reduction is possible in some parts of the studied area ratio.
Figures 8 – 10 illustrate the simulated cases for the system containing hydraulic pumps in which displacement are 8 cm³/rev and the cylinder size is 63/45 having area ratio of 1:2.04. In the simulations, the accumulator is replaced by a constant pressure source with pressure levels of 10 bar, 140 bar, and 260 bar. The external load generates a pressure level of 260 bar either at the bottom of the cylinder or the rod side of the cylinder. The cylinder is driven twice from the inner position to the outer position and back. During the first half (0–20 s) the external load is the retracting load (i.e., it generates pressure at the bottom of the cylinder). The first half covers the quadrants 2 and 3, in that order. During the second half of the simulation (20–40 s), the external load is the extending-type and it generates pressure at the rod side of the cylinder. During the second half, the operations in quadrants 1 and 4 are used in the same order. Between 19.5 and 20.5 s, the external load is changing from retracting mode to extending mode, resulting a ripple in power. The total efficiency map of the hydraulic pump/motors is shown in Appendix and the map is generated from the efficiency information obtained for two constant axial piston units. Figures 7 - 9 illustrate the input power for either the hydraulic power or the electric and the output power of the cylinder. The positive power indicates that the power flows from the source towards the load. Negative power indicates regeneration from the load to the hydraulic and electric grids.

![Figure 8. Power level when accumulator pressure is 10 bar.](image1)

![Figure 9. Power level when accumulator pressure is 140 bar.](image2)
When the accumulator pressure is 10 bar, the power demand for the electric motor is at its highest (15 kW) and the cylinder is moving outward against the load pressure (quadrant 2). When the accumulator pressure is 140 bar only 11 kW of electric power is needed and the demand is highest (quadrants 2 and 1). When the accumulator pressure is 260 bar and is equal to the load pressure, the highest power demand is 14 kW (quadrant 1).

The weak point of this circuit design is apparent and it is the 4th quadrant (30–40 s) when the accumulator pressure is greater than the supply pressure level because it increases the power demand. Both hydraulic units are working as a pump requiring power; the left pump is charging the accumulator and the right pump is working against the load pressure.

The simulated power ratio (electric power vs. hydraulic power) and the power ratio, according to equations 6–9, are comparable; this is shown by the dashed line in Figure 11, which represents the simulated power ratio without efficiencies. The efficiencies of the hydraulic units have an effect on the power ratio because they increase the power demand and reduce the amount of recovered energy. The lower the efficiency, the greater the effect on the power ratio.

The potential of reducing the size of the electric motor is dependent upon the operation quadrants and the ratio between the load and accumulator pressures. The potential is highest for operation quadrants 2 and 3.
when the bottom of the cylinder is the load side and the accumulator pressure is equal to the load pressure. In this case, a 50% reduction in the size of the electric motor is possible.

The operation in quadrant 4 is the worst case because both pumps are working against the pressure, maximizing torque demand; the lowest power demand is when the accumulator pressure is equal to the supply pressure level.

The operation in quadrant 1 is also problematic because the accumulator is discharged although the load pressure rotates the right pump and both hydraulic units are working as a motor and the electric motor is acting as the generator.

If the load pressure is at the same level (bottom and rod side), the potential of the required electric motor size reduction is defined by the operations in quadrants 1 and 4. When the accumulator pressure is optimized, the required size reduction is about 25%.

The reduction of the electric motor size is limited by controllability. The motion control of the load is done by the electric motor; therefore, it has to have enough torque in every case to overcome the torque demand. This fact can decrease size reduction of electric motor potential in some cases.

4. CONCLUSION

The desire to manufacture more efficient drives has led to an increase in the use of electric drives in different applications. However, in comparison to hydraulic drives the power density is still a weakness of electric drives and reduces applicability of electric drives in mobile applications. The speed controlled cylinder drive combines the benefits of electric and hydraulic drives. When the system is supplied by electric and hydraulic energy, it is a hybrid cylinder drive system that enables the reduction of the size of the electric motor and, this increases the system’s power density and decreases installation space.

The studied double pump circuit enables up to a 50% reduction in the size of electric motors when the bottom of the cylinder is the load side and the second pump is connected to the accumulator or pressure source. This is applicable in lifting applications where the rod side loading is remarkably lower as well as in other applications that have a minor extending load. When the load pressures are same for the bottom of the cylinder and the rod side of the cylinder, the size reduction potential is around 25%.

In conclusion, it can be stated that the size of an electric motor can be reduced by using a hybrid energy supply but the results depend upon the size of the cylinder, the pressure levels and especially the load pressure at the rod side.

5. REFERENCES

6. APPENDIX

Figure 12. AmeSim simulation model

Figure 13. Efficiency map for the hydraulic units


COMPARATIVE STUDY OF FUEL REDUCTION METHODS FOR HYBRID EXCAVATORS

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ABSTRACT

This paper presents a simulation study of three fuel reduction methods for hybrid excavators. The configuration of the studied hybrid excavator is a compound type. An optimal control method, dynamic programming, is used in the simulation to reduce the affection of non-optimal control strategy. Comparison of the three methods is first analyzed with the same work load to evaluate the effectiveness of these three methods on the fuel economy. Then a further study of the relationship of the three methods with different work loads and assist motor sizes is analyzed. Simulation results show that the three fuel reduction methods all have pros and cons. The fuel reduction method should be carefully selected for different work loads.

KEYWORDS: Hybrid excavator, fuel reduction, engine downsize, speed reduction, torque compensation

1. INTRODUCTION

Excavators are heavy construction equipment. They are widely used in infrastructure construction. However, they consume excessive energy and have worse pollution. Due to global warming and increasing price of the oil, new technologies for excavators are required to lower the fuel consumption and reduce the emission. Hybrid technology, which is first applied on vehicle successively, is a promising and effective way to reduce fuel consumption and emission. In recent years, hybrid technology has been introduced to construction equipment especially the excavators.[1-3] However, the structure of the excavators is more complicated than that of the vehicles. In addition, the work load of the excavators is also different from that of the vehicles. Since the fuel cost for running an excavator is much higher than that of a vehicle, development of fuel reduction method for the hybrid excavator is much more needed.

Several kinds of hybrid excavator are presented in [4, 5]. According to the structure of the drive train, they can be classified into three categories such as parallel, series and compound type. The compound-type hybrid excavator, which is researched most, has both the advantages of simple structure and good fuel improvement. In the compound-type hybrid excavator(Figure 1), the engine, the assist motor and the pump are directly connected to each other by the mechanical shaft, so the speed of these three components are just the same. The assist motor can work as a motor to provide additional power for the drive train or work as a generator to charge the energy storage system. An electric motor is used to replace the hydraulic motor for the swing motion in order to regenerate power of the upper structure of the excavator. Besides, the electric swing system can reduce the hydraulic loss of the traditional swing system due to the high efficiency of the electric motor. The energy storage system usually chooses the ultracapacitor for its fast dynamic, high specific power and long life.
Many studies can be found concerning the control strategy of hybrid vehicles, but those methods cannot be directly applied to the hybrid excavators due to totally different configurations and work loads. A few papers can be found in the literature on the fuel reduction method of the hybrid excavator.[2, 4, 6, 7] However, comparison of different fuel reduction methods and their effects with different work loads or component sizes are not analyzed.

This paper considers three fuel reduction methods for hybrid excavators (engine downsizing, speed reduction, torque compensation). Here we only focus on the power system, because the swing system cares more about drivability rather than fuel consumption.

2. METHOD

2.1. Fuel reduction methods

The engine operating points of the traditional excavator are present in a fuel consumption map from empirical data shown in Figure 2. The thin lines represent relative power-specific fuel consumption values at different speeds and loads. The thick line represents the maximum torque at different speeds. The engine speed of the excavator is always set to the maximum value in hard-work mode in order to utilize the engine capability up to its rated power and the work load usually distributes from very low power to very high power. These two reasons make the engine operating points strayed from the fuel efficiency region. The engine cannot be optimized for all the speed and load ranges under which it must operate. Based on this fact several fuel reduction methods are proposed. Here we concern the following three methods: engine downsizing, speed reduction and torque compensation.


Figure 2. Engine operating points of the traditional excavator

**Engine downsizing**

The energy saving mechanism of engine downsizing in hybrid excavators is the same as that of the hybrid vehicles. As we all know the engine fuel economy and emission are very bad at low loads but are improved at high loads. In hybrid excavator, the engine can be downsized due to the additional power of the assist motor. The assist motor can store the energy at low loads while use it at high loads. Therefore, a downsized engine can operate most of the time at the efficient region and thus reduce the fuel consumption and emission.

**Speed reduction & Torque compensation**

Speed reduction and torque compensation are two other fuel reduction methods aiming to regulate the engine operating points to the efficient region. From the engine fuel consumption map, we can see that the engine operating points are commonly strayed from the efficient region. Another fact is that the engine speed is always set constant to ensure the working performance while working, so to optimize the engine either reduce the speed or compensate the torque of the engine at the rated speed is needed.

2.2. Dynamic Programming

The purpose of this paper is to find out how well these three methods do in the fuel reduction job. In order to evaluate the potential capacity of these methods, an optimal energy management algorithm, dynamic programming (DP), is used in the simulation. DP can reduce the affection of non-optimal energy management strategy and ensure the final state of the ultracapacitor is the same as the initial value.

DP is a multi-stage optimization method based on Bellman’s optimality principle: An optimal policy has the property that, whatever the initial state and optimal first decision maybe, the remaining decisions constitute an optimal policy with regard to the state resulting from the first decision. [8] The optimal problem in this study is formulated as follows:

\[ \min_{u(t)} J = g_N(x_N) + \sum_{i=0}^{N-1} g_i(x_i, u_i) \]  

(1)

where \( J \) represents the cost function to be minimized. It is the total fuel consumption during a given operating cycle. \( x_i \) is the state variable (in this case, the SOC of the ultracapacitor), \( u_i \) is the input variable (assist motor torque), and \( g_i \) is the fuel consumption of the engine multiplied by the time step \( \Delta t \). The final cost \( g_N(N) \) is zero. \( u_i = \{u_0, u_1, \cdots, u_{N-1}\} \) is the input variable trajectory which is the assist motor torque strategy in this
problem. The optimal problem is to find the optimal assist motor torque strategy that minimizes the total fuel consumption.

2.3. System model

In order to apply DP algorithm, a quasi-static model is used due to its small computation time. The engine and electric motors dynamics are ignored. Only the ultracapacitor’s slow dynamics is considered. The excavator studied in this paper is a mid-sized 20-ton excavator.

The engine model uses a static look-up table that provides the fuel consumption with different engine torques and engine speeds.

\[ m_f = f(T_e, \omega_e) \]  

(2)

The electric motors are modelled using the efficiency map. The motor efficiency is described by

\[ \eta_m = g(T_m, \omega_m) \]  

(3)

The assist motor speed is the same as the engine speed. Assist motor efficiency map is shown in Figure 3.

![Figure 3. Assist motor efficiency map](image)

The ultracapacitor consists of several individual capacitor cells connected in series and parallel. The model of the capacitor cell can be represented by a capacitance in series with a resistor.

\[ P_{uc} = (u_c i_c - R i_c^2) n \]  

(4)

\[ \frac{du_c}{dt} = \frac{i_c}{C} \]  

(5)

\[ P_{uc} \] is ultracapacitor power, \( i_c \) is current in the capacitor cell, \( u_c \) is open circuit voltage of the capacitor cell, \( R \) is equivalent resistance, \( C \) is capacitor cell capacity and \( n \) is the number of the capacitor cells.

The power balance equations are as follows

\[ P_e + P_{am} - P_p - P_l = 0 \]  

(6)

\[ P_{uc} + P_{am} + P_{sm} = 0 \]  

(7)

where \( P_e \) is engine power, \( P_{am} \) is assist motor power, \( P_p \) is pump power, \( P_{sm} \) is swing motor power, \( P_{uc} \) is ultracapacitor power, and \( P_l \) is engine power loss. The pump power and swing motor power are got from the work cycle.
3. RESULTS AND DISCUSSION

The optimal energy management process is performed over a digging cycle which is a typical work task of the excavator. In this operation cycle, the excavator digs a load of earth or sand, rotates and unloads it into a truck. This operating cycle is usually used to evaluate the fuel economy of the excavators. Figure 4 shows the power of the pump and swing system and actuator positions in the digging cycle. The digging cycle is got from an experiment by an expert operator.

![Figure 4. Digging cycle and actuator positions](image)

3.1. Engine downsizing method

In engine downsizing case, we consider four engine sizes (82%, 87%, 91% and 95% of the baseline engine) so as to evaluate the effect of different engine sizes on the fuel consumption. Figure 5 shows the simulation results of the four engine sizes in terms of fuel consumption reduction. The ultracapacitor size is selected large enough to exclude its effect while the assist motor size is also fixed to 23kW for all engine sizes.
Figure 5. Fuel economy increase of engine downsizing method with different engine sizes

From Figure 4 we can see that engine downsizing can significantly reduce the fuel consumption. This is due to the fact that the downsized engine can operate most of the time at high efficient regions. Besides, smaller engine can get a relatively better fuel economy. Another benefit downsizing engine can achieve is cost effectiveness. Although the small engine always needs a large assist motor to meet the power demand, the additional component cost of the assist motor is much small compared to the reduced cost of the engine.

Although downsizing engine is an effective method to reduce the fuel consumption, the downsizing room is not that large compared to the hybrid vehicles because of the high load of the excavator. Therefore engine downsizing method fit well with the excavators doing light work tasks due to the large downsizing room.

3.2. Speed reduction method

The key parameter of the speed reduction method is the reduced engine speed. Different speed may bring different fuel consumption reduction. Therefore, three different speeds (1700rpm, 1800rpm, and 1900rpm) are studied here. The simulation results of these three speeds are shown in Figure 6. The assist motor size and ultracapacitor size are kept the same.

Figure 6. Fuel economy increase of speed reduction method with different engine speeds

It is evident from figure that reducing engine speed is an effective method to reduce the fuel consumption. For example, at 1800rpm the fuel economy increase reaches almost 4% which is larger than that of engine
downsizing method. The reason for this fact is that the engine efficiency around the reduced speed is higher than that of the rated speed. Another reason is that reducing speed can make the engine operating points more concentrated to the high load region which is more efficient. We can also see from figure that the fuel consumption is related to the reduced speed. The optimal reduced engine speed is 1800rpm because of the high efficient region around it.

3.3. Torque compensation method

The mechanism of the torque compensation method is redistributing the engine operating points at a constant speed so as to get a better fuel economy while fully use the engine capacity. From the engine fuel efficiency map, we can see that the changes of the engine efficiency at different engine speeds are not the same. Hence, the effect of the torque compensation method is related to the engine working speed. The fuel consumption of the torque compensation method with different engine speeds is shown in Table 1.

<table>
<thead>
<tr>
<th>Engine speed</th>
<th>1700rpm</th>
<th>1800rpm</th>
<th>1900rpm</th>
<th>2000rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel economy increase</td>
<td>0.20%</td>
<td>1.62%</td>
<td>1.01%</td>
<td>2.72%</td>
</tr>
</tbody>
</table>

The torque compensation method is effective at 2000rpm speed while for other speeds the effect is not that significant. Therefore, torque compensation method is suitable for work tasks with rated speed.

Note that the benefit that the torque compensation method brings is premised on a large assist motor. It is well understood that the large assist motor can provide high additional torque thus regulate more engine operating points to the efficient region. In order to make a further comparison, the torque compensation method with different assist motor size is studied. Figure 7 shows the fuel consumption of torque compensation method with different assist motor sizes. We can see the fuel consumption is quite sensitive to the assist motor size. Therefore, we recommend large assist motor size in torque compensation method so as to get a good fuel economy.

![Figure 7. Fuel economy increase of torque compensation method with different assist motor sizes](image)

From the previous analysis of the three fuel reduction methods, we can see for standard digging cycle the speed reduction method can get the best fuel economy while the torque compensation method is less effective. A drawback of the speed reduction method is that the pump control strategy should be modified so as to keep the flow rate constant since the speed is reduced. A larger pump is also needed to ensure enough output power. Engine downsizing method can get a comparative fuel economy. Meanwhile, it also reduces the component cost. However, it cannot increase the work efficiency compared to the torque compensation method.
3.4. Sensitivity to work loads and assist motor sizes

The characteristic of the work loads can be represented by the load ratio (average load power/maximum load power). In order to evaluate the effectiveness of the three methods with different work loads, a grading cycle which has high load ratio is used here (Figure 8). In this work cycle there is always no swing motion involved. Therefore, the load ratio is bigger than that of the digging cycle. The load ratio of grading cycle is 86% which is much higher compared to the digging cycle’s 72%. The simulation results of the three methods with the grading cycle are presented in Table 2.

![Figure 8. Grading cycle](image)

We can see the speed reduction method is more adept at handling the high ratio load, while the other two methods are less effective. This is because in high ratio load most operating points are already around the efficient region. Therefore the room for regulating operating points is much small. We recommend using speed reduction method in high ratio load.

<table>
<thead>
<tr>
<th>Methods</th>
<th>Specification</th>
<th>Fuel economy increase</th>
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<tbody>
<tr>
<td>Engine downsizing</td>
<td>Engine size: 91%</td>
<td>3.01%</td>
</tr>
<tr>
<td>Speed reduction</td>
<td>Speed: 1800rpm</td>
<td>7.48%</td>
</tr>
<tr>
<td>Torque compensation</td>
<td>Assist motor: 42kW</td>
<td>2.55%</td>
</tr>
</tbody>
</table>

Table 2. Fuel economy increase of three methods with grading cycle

As we have analyzed before, the effectiveness of torque compensation method is sensitive to the assist motor size. However, whether the effectiveness of the other two methods is related to the assist motor size or not is yet to be researched. In this section, the sensitivity of the other two methods to the assist motor size is studied. The fuel consumption results are shown in Figure 9 and 10.
Both the speed reduction and engine downsizing method are not sensitive to the assist motor size. This fact can be explained from two aspects: First, the engine operating points are more concentrated due to the reduced speed and downsized engine. Second, the changes of engine efficiency in different region are not evident in downsized engine or reduced speed case. Hence, the effectiveness of regulating the engine operating points by the assist motor becomes small.

4. CONCLUSION

This paper has analyzed three fuel reduction methods for hybrid excavators. Each of the three fuel reduction method is analyzed respectively first with the standard digging cycle. Simulation result shows all the three methods can improve the fuel efficiency. But the energy saving effect of different methods is affected by different factors. For the engine downsizing method, the smaller the engine size is the more efficient the hybrid excavator is. While for the speed reduction method, there is an optimal speed at which the hybrid excavator is the most fuel efficient. When it comes to the torque compensation method, the fuel efficiency is sensitive to the engine speed.

The energy saving effect is also sensitive to work loads and assist motor sizes. Sensitivity simulation results show that speed reduction method is work-load-free, which means speed reduction method can do well in fuel reduction regardless of the work load while the other two are sensitive to work load. Especially in high work load, the performance of engine downsizing and torque compensation method is not good.

The torque compensation method is quite sensitive to assist motor size. However, the other two are not sensitive to the assist motor size. We recommend a large assist size in torque compensation method.

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HYDRAULIC HYBRID ACTUATOR: THEORETICAL ASPECTS AND SOLUTION ALTERNATIVES

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ABSTRACT

This paper presents and analyzes a hybrid solution, in which the hydraulic energy storage element is integrated to the hydraulic actuator. The approach results in a new system layout—a distributed hybrid system—in which only mean power is transmitted between the actuators and the high power peaks are handled locally. Three different implementations are discussed. A multi-actuator excavator load cycle is analyzed and dimensioning of the components is discussed. Limitations of the approach are also discussed.

KEYWORDS: hydraulic hybrids, hybrid actuator

1. INTRODUCTION

A traditional hydraulic system is based on a centralized pump unit that produces hydraulic power for all the actuators of the system. The main advantages of the solution are its simple layout and its low price. The main disadvantage is poor energy efficiency because one pump can produce only one pressure level for the system. Several approaches have been suggested to improve efficiency: hydraulic transformers [1, 2], pump controlled actuators [3], hydraulic hybrids [4, 5, 6], distributed valve systems [7], and the use of several pressure sources [8–11]. Electrohydraulic actuators are another strong trend; these are used routinely in modern airplanes and also in some industrial applications [12, 13]. The concept typically includes a variable speed electric motor, a fixed displacement pump, and a low-pressure accumulator as an oil reservoir and a cylinder. The drawback of this solution is the one-to-one connection between the electric power source and the hydraulic power. This means that the electric motor must be able to produce the peak power of the actuator, which results in either low maximum power or a bulky solution. The problem can be significantly reduced by integrating the electric motor and the hydraulic pump into a compact unit [14], but the electric motor and the hydraulic pump must still be able to produce the maximum power of the actuator.

Hydraulic hybrid solutions are already used in commercial machines. Komatsu’s hybrid excavator has an electric swing actuator, while others are traditional hydraulic actuators [4]. A supercapacitor is used as the energy storage and the estimated reduction in the fuel consumption is 25%. Caterpillar has presented a hydraulic hybrid excavator, which uses distributed control valves and a hydraulic pressure accumulator as the energy storage [5]. Only a swing drive has an energy recovery function and the estimated reduction in the fuel consumption is also 25%. In both cases, the reduction of the fuel consumption is moderate because only the swing actuator has an energy recovery function and because boom actuators use throttling control. Significantly larger fuel savings are possible if all the main actuators have an energy recovery function and if
they are controlled without throttling. Kobelco has introduced a solution in which both the swing drive and the boom lift actuator have energy recovery [6]. The throttle-free pump control is used and the impressive 40–60% reduction in the fuel consumption has been achieved. The pumps are driven by electric motor/generators and the system layout is an electric series hybrid. The drawback of the solution is that it is complex having three pumps, four speed variable electric motors, a supercapacitor and a Ni-MH battery. A solution based on hydraulic transformers has been theoretically analyzed in [2] and the simulation results showed a fuel consumption reduction of up to 50% for the wheel loader. However, the hydraulic transformers are not commercially available. Hybrid solutions based on the utilization of three pressure levels have been presented by Lumkes and Andruch [8], Erkkilä et al. [9], Vukovic et al. [10] and Ketonen et al. [11].

The common feature of the previous hybrid solutions is that they have centralized energy storage. The benefits of the approach are a simple layout and the possibility to transfer energy from one actuator to another. The drawback is that big power must be transferred between the energy storage and the actuators. An alternative approach is studied in this paper. The hydraulic accumulator is integrated into the actuator and it is used to cover the peak power of the actuator. Only the mean power is transmitted into the actuator while the power peaks are handled locally at the actuator. Although the idea is simple, the authors have not been able to find any related publications.

2. THEORETICAL ANALYSIS

2.1. System Layout

The two possible system layouts are shown in Figure 1. Both alternatives are based on the idea that big power is kept at the actuators while mean power is transmitted to the actuators via a low power hydraulic line (version (a)) or an electric line (version (b)). The latter version requires an integrated pump and a low-power electric motor at the actuator. In both cases, following power flows are needed:

1) From the mean power source to the accumulator (accumulator charging)
2) From the accumulator to the actuator (satisfaction of power peak demands)
3) From the actuator to the accumulator (energy recovery)
4) Optionally, from the mean power source to the actuator

It is assumed that there is no need for the power from the actuator or the accumulator to be fed back to the mean power line, which simplifies the mean power supply system.
2.2. Lossless Power Analysis

Consider a hydraulic cylinder with piston areas $A_A$ and $A_B$, stroke $x_{\text{max}}$ and maximum pressure $p_{\text{max}}$. The maximum power of the actuator is:

$$P_{\text{max}} = F_{\text{max}} v_{\text{max}} = p_{\text{max}} A_A v_{\text{max}}$$

(1)

The maximum power available from the MP line is:

$$P_{\text{MP, max}} = Q_{\text{MP, max}} p_{\text{MP}} = \varepsilon P_{\text{max}}$$

(2)

where the design parameter $\varepsilon$ tells how big a part of the actuator power the MP line can produce. The power needed from the accumulator is:

$$P_{\text{Acc, max}} = (1 - \varepsilon) P_{\text{max}}$$

(3)

The average power needed by the actuator is:

$$P_{\text{avg}} = \frac{\int_0^{t_C} F v dt}{t_C}$$

(4)

where the integration interval $t_C$ covers the complete load cycle of the actuator. The MP line must cover the average power of the actuator, and the lower limit for $\varepsilon$ is thus:

$$\varepsilon \geq \frac{P_{\text{avg}}}{P_{\text{max}}}$$

(5)
2.3. Accumulator Sizing

If ideal pressure transformation between the accumulator and the actuator is assumed, the accumulator sizing can be made according to its energy balance. Consider the work cycle shown in Figure 2. The maximum power is needed to extend the piston and the same power is recuperated during the retracting movement. The average power required is zero; thus, it is impossible to charge the accumulator from the MP line. If MP line were used, the energy would cumulate to the accumulator. All the energy is first taken from the accumulator and then recuperated; this cycle represents the worst case. The energy needed in the first half of the work cycle is:

\[ E_{\text{max}} = F_{\text{max}} x_{\text{max}} = A_p p_{\text{max}} x_{\text{max}} \]  

(6)

and this is the worst case scenario for the accumulator capacity. In reality, the situation is much better because actuators do not work against the maximum force for the whole stroke and because the MP line can be used to produce part of the energy needed. If the load cycle is such that full capacity of the MP line can be used, the amount of energy needed from the accumulator is reduced to:

\[ E_{\text{max}} = (1 - \varepsilon) F_{\text{max}} x_{\text{max}} \]  

(7)

Figure 2. The worst case load cycle.

The typical actuator of a medium size excavator has a diameter of 130 mm and a stroke of 1 m. This results in the worst case accumulator energy of 465 kJ, if the maximum pressure is assumed to be 35 MPa. This kind of energy storage capacity is achieved with about 45 l accumulator \((p_0 = 11 \text{ MPa}, p_{\text{max}} = 35 \text{ MPa})\). The oil volume of the A-chamber of the cylinder is 13 l and a rule of thumb is that the accumulator volume is 3.5-times the volume of the cylinder chamber. This means that the physical size of the accumulator is significantly bigger than the size of the actuator in the worst case scenario.

3. IMPLEMENTATION OPTIONS

3.1. Implementation with a Speed Variable Pump-motor

One way to implement the hybrid actuator is to use the speed variable pump-motor. A possible implementation is shown in Figure 3. The system has a 2:1 differential cylinder and it is controlled by the speed variable pump-motor. The inlet of the pump-motor is connected to either the tank line or to the accumulator line, which allows the pressure boost without increasing the size of the electric motor.
Figure 3. Hybrid actuator with a speed variable pump-motor.

Figure 4 shows the different control modes of the system together with the pressure ranges. It is assumed that accumulator pressure is 10 MPa and the maximum pressure is 20 MPa. The logic valves are not shown in order to simplify the pictures. It is seen that the only difficult situation is the large overrunning load with the extending direction of the movement. This kind of situation occurs rarely and it could be handled with the proportional valve between the B-chamber and the accumulator.

3.2. Implementation with a Multi-chamber Cylinder

The drawback of the solution presented in Section 3.1 is that it requires an electric servomotor with a peak power that is about half of the maximum power of the actuator. This makes the solution expensive and
unnecessarily bulky, especially if the mean power is much smaller than the peak power. A secondary controlled multi-chamber cylinder is one approach that can be used to solve the problem [15]. The idea is to have a cylinder with four piston areas in ratios 8:4:2:1 and to connect these chambers to either the high-pressure line or the low-pressure line via low resistance logic valves, as depicted in Figure 5. The result is an actuator with 16 different equally spaced output forces, which can be utilized in the secondary control. The experimental results in [15] have shown that it is possible to implement energy efficient motion with a high ratio between the peak and mean powers. The challenge of the approach is controllability with small inertia loads, which can be solved by using slight resistance control [16]. The difference between the original publication [15] and Figure 5 is subtle: the high-pressure accumulators are now integrated on the actuator and, thus, the high power is handled locally.

![Figure 5. Hybrid actuator implemented using a four-chamber cylinder [15].](image)

### 3.3. Compact Implementation of a Constant Pressure Accumulator

The solutions presented in Sections 3.1 and 3.2 require a constant pressure accumulator. As the energy storage capacity of the accumulator depends on the ratio between its maximum and minimum pressure, very big accumulator is needed, which negates the compact hybrid actuator idea. One solution to this problem has been presented in [17]. That solution is a piston accumulator in which the piston has four areas as shown in Figure 6. This allows almost full utilization of the energy storing capacity of the gas volume.
3.4. Multi-pressure Implementation

An alternative to the multi-chamber cylinder is the combination of the standard cylinder and multiple pressure sources. Again, the straightforward implementation with multiple accumulators and a loading pump yields a bulky solution so some kind of integrated solution is needed. One such sketch is shown in Figure 7 in which only one gas volume is used together with several independent pistons. The pressures have fixed ratios and the usage of the accumulator changes all the pressure levels. The loading of the accumulator is made to the highest pressure chamber only. The positions of the other pistons are controlled by active selection of the control mode of the system. The proportional valves shown in figure are optional and they may be needed in small inertia systems. The solution resembles the three pressure hybrid systems [8–11], but the principle is different. The local energy storages are used and intermediate pressures are also generated locally.
4. EXCAVATOR LOAD CYCLE ANALYSIS

4.1. Introduction

The analysis presented in Section 2.3 shows that the limited accumulator energy storing capacity is a big challenge. The analysis is based on the worst case load cycle scenario and the situation may be much better in practice. This is why an excavator load cycle is analyzed. A 21 ton wheeled type excavator equipped with mobile proportional valves was used. The output powers of the actuators are calculated as a product of piston force and velocity, and the piston force is calculated from the measured chamber pressures.

4.2. Load Cycle

The load cycle is medium speed digging of macadam from the outermost position of the bucket. The unloading position is about 2.5 metres above the ground level and close to the machine. Figure 8 shows the measured actuator positions and the output powers of the load cycle. The mean powers of the boom, arm, bucket, and swing actuators are 1.21 kW, 3.79 kW, 2.95 kW, and 0.78 kW, respectively. These are very small values when compared to the peak powers, which are 64.4 kW, 27.7 kW, 48.0 kW, and 22.1 kW.

![Figure 8. Measured actuator positions and output powers of the digging cycle.](image)

4.3. MP Power and Accumulator Sizes

The minimum value for the MP power is the sum of the mean powers of the actuators, which is only 7.9 kW. However, the MP power is selected to be 20 kW, which takes into account the fact that the load cycle is not too aggressive and that losses occur in the system. Assuming that 5 kW is lost, 15 kW from the MP line is
still available to the actuators. The power curves show that the boom actuator requires the most energy and the energy used for lifting is 210 kJ. The lift movement takes 7 s and if half of the MP power is available for the boom actuator, the energy taken from the MP line is 52.5 kJ. The boom accumulator energy storing capacity should thus be about 160 kJ. However, the starting point of the digging is ground level and in practice longer lifting movements are possible. This argument yields to an accumulator that is about doubled in size, and it can be concluded that an energy storing capacity of 320 kJ should be enough. The corresponding accumulator size is about 30 l. The worst case energy storing capacity of Eq. 6 is much larger, 1040 kJ (two 130 mm cylinders, $x_{\text{max}} = 1.12 \text{ m}$, $p_{\text{max}} = 35 \text{ MPa}$).

5. LIMITATIONS OF THE APPROACH

5.1. The Energy and Power Capacity of the Accumulator

The worst case analysis presented in Section 2.3 shows that the physical size of the accumulator is clear limitation with some types of load cycles. On the other hand, the excavator load cycle analysis shows that the situation is much better in practice. Wide pressure variation is required in order to utilize the full capacity of the accumulator, which calls for some kind of pressure transformation between the accumulator and the actuator. The multi-area accumulator [17] is an interesting alternative for implementing this. The power density of hydraulic accumulators is good, but the challenge is how to guarantee good efficiency with rapid loading and unloading of the accumulator. Some kind of insulation or heat regeneration solutions are probably needed in order to guarantee proper and efficient operation of the accumulator [18, 19].

5.2. Control of the Energy Balance: Under-power and Stagnation

The proper control of the energy balance of the accumulator is essential for the correct operation of the hybrid actuator. If the energy level is too low, it is possible that the actuator might be temporarily unable to produce enough power and under-power situation could occur. Conversely, if the energy level is too high and the accumulator cannot receive additional power from the actuator, stagnation occurs. Predictive and learning control are possible solutions for these challenges.

5.3. Temperature

The amount of oil in the hybrid actuator is small and it is easy to overheat the system if its efficiency is not good enough. If the mean power is 10 kW and efficiency is 80%, there is still 2 kW heat generation, which might be difficult to dissipate without active cooling. Thus, the highly efficient solutions are thus needed.

5.4. Controllability

All the solutions presented in this paper are based on the concept of different control modes. The switching between control modes must happen smoothly, which is challenging. Fast components and proper control methods are needed in order to guarantee good controllability. Fortunately, a new fast and high capacity on/off valve has come into the market [20].

6. CONCLUSIONS

A hybrid solution has been presented in which the energy storage elements are integrated into actuators. The feasibility of the approach has been analyzed and three different solution alternatives are given. The
energy storing capacity of the hydraulic accumulators is the biggest challenge in order to make the solution compact enough. The size of the accumulator depends strongly on the load cycle of the actuator and it can be concluded that the approach has potential in many applications.

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ABSTRACT

This paper aims to compare three solutions of potential energy recuperation on a mobile harbour machine designed to carry and stack containers. The boom actuated by two hydraulic cylinders can lift and lower loads up to 45 tons. The current system dissipates the energy during boom lowering through a flow control valve.

The three presented systems use hydropneumatic accumulators to temporarily store the energy and then release it during a boom lifting or any power demand. The first system uses only a flow control valve to reach the pressure imposed by the hydropneumatic accumulator. The second system uses a transformer based configuration directly coupled to the internal combustion engine (ICE). This layout allows the stored energy to be easily released but also to be recovered regardless of the pressure difference between the hydraulic circuit and the hydropneumatic accumulator. The last presented system is also composed of a pressure transformer which is not attached to the ICE. This solution enables the recuperation devices to have a rotation speed independent of the ICE but also to avoid the engine braking.

The solutions have been modelled and simulated for different initial and final positions of the container in terms of height and depth. A duty cycle has been performed giving for each solution a global view of the fuel savings. The system without transformer showed an amount of energy recovered lower than 37% because of the limitation of the accumulator volume. The second solution directly coupled to the engine shaft demonstrated better recuperation performances. However between 15% and 20% of the recoverable energy is dissipated by the engine braking. The recuperation motor drives indeed the ICE during the boom lowering generating a parasitic negative torque. Besides, the relatively low speed imposed by the ICE induces the need for high displacement units to achieve the rod retraction speed specification. The last architecture showed fuel consumption economy of up to 16%. The great advantages of this solution is its independence of the ICE speed and the engine braking but also the possibility to recover energy even when the lifting actuator pressure is low, for instance when an empty container is lowered.

KEYWORDS: hydraulic hybrid, potential energy recovery, reach stacker, off-highway vehicle
1. INTRODUCTION

Since the end of the 1990’s off-highway vehicles are subjected to more and more restrictive norms concerning pollutant emissions. We can mention for instance the US Tier IV Final standards or the EU Stage IV. Moreover owners of heavy machines are trying to reduce their operating costs to keep competitive. Since fuel costs related to the use of machines are an important part of the operational charges, the reduction of the consumption is a key topic. Lot of work has been done recently to find solutions to save fuel like improving the efficiency of already existing components or finding new architectures to transmit energy in a more efficient way [1] [2]. It is also sometime possible to recover the energy which is lost in traditional machines.

The case study focuses on an off-highway vehicle working in harbours or transport hubs whose task is to carry containers and stack them on the appropriate area. Lifted heavy containers represent a large amount of potential energy which is currently transformed into heat by the meter-out valve of the hydraulic circuit. Recovering the potential energy and use is later when the rest of the system has a power demand would result in a decrease of the required power of the internal combustion engine (ICE) and consequently fuel savings. In this paper, three different architectures of potential energy recovery systems are presented and compared by the mean of simulation. The three solutions are based on hydraulic components for different reasons. Firstly the high power density of hydraulics is suitable for recovering a important amount of energy during a short time [3]. Secondly working in the same energy domain as the rest of the system is a good way to limit energy transformations which induce automatically losses for each change of physical domain. Finally it is much simpler for the owner of the machine to maintain a system with similar components.

The part 2 of this paper describes the operating principle of the reference machine. The layouts and the modelling of the three solutions are presented in part 3 and the last part deals with simulation results and the comparison between each architecture. Future work will focus on the implementation of the selected solution on a real machine.

2. OPERATING PRINCIPLE OF THE MACHINE

2.1. Description of the reach stacker

Reach stackers are mobile machines capable of carrying containers up to 45 tons to a height of five standard containers and 35 tons in sixth height. Figure 1 shows a typical maximal load range of container stacking. The machine has an Internal Combustion Engine (ICE) as primary energy source providing the necessary power to several actuating lines permitting the machine to translate, lock and lift the container in order to move it to its next place. We can mention the powertrain composed of a torque converter and an automatic gearbox driving the energy to the front axle. The ICE also drives two hydraulic pumps both equipped with a Load Sensing (LS) system. The hydraulic fluid is supplied to the lifting and telescoping cylinders via two proportional valves controlling the flow rate. The two lifting cylinders and the telescoping cylinder are the main actuators as they require the highest flow rates. Table 1 presents the main features of the ICE and hydraulic circuit. Finally the spreader interfaces the machine with the container via different smaller actuators and a twist-lock system.

Auxiliary components like steering system, cooling circuit, braking circuit or spreader actuators are neglected insofar as their energy consumption is much lower than the other actuator needs. Thus are selected the powertrain, the lifting and telescoping system to be modelled. The powertrain is considered in order to simulate the functioning of the machine on a global duty cycle including translations and hydraulic movements. Finally a multi-body dynamic model is also established in order to represent accurately the behaviour of the machine.
Table 1. ICE and hydraulic features

<table>
<thead>
<tr>
<th>Element</th>
<th>Feature</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>Maximum power</td>
<td>235 kW</td>
</tr>
<tr>
<td></td>
<td>Maximum torque</td>
<td>1580 N.m at 1260 rpm</td>
</tr>
<tr>
<td>Pump 1</td>
<td>Maximum displacement</td>
<td>145 cm³</td>
</tr>
<tr>
<td>Pump 2</td>
<td>Maximum displacement</td>
<td>145 cm³</td>
</tr>
<tr>
<td>Lifting cylinders</td>
<td>Maximum pressure</td>
<td>420 bar</td>
</tr>
<tr>
<td></td>
<td>Length of stroke</td>
<td>2.95 m</td>
</tr>
<tr>
<td>Telescoping cylinder</td>
<td>Maximum pressure</td>
<td>350 bar</td>
</tr>
<tr>
<td></td>
<td>Length of stroke</td>
<td>8.3 m</td>
</tr>
</tbody>
</table>

2.2. Baseline model

In [4] a dynamic model of the baseline machine has been established and compared with measurements. A simplified outline of the machine is shown on Figure 1. This physical model which is divided into three connected submodels, namely the multibody, the hydraulic and the powertrain submodels has been used as a reference.

The dynamics of the ICE includes its inertia and shaft stiffness but the torque is deduced from the speed-torque characteristic curve given by the manufacturer. Concerning the powertrain the torque converter has been dynamically modelled and the gearbox behaviour like transmission shift points and internal clutches engagement has been taken from measurements. Those measurements also permitted to define the rolling resistance and the losses along the powertrain.

Pump inertias and fluid compressibility are considered in the model and pressure drops for each valves have been correlated with the reality. The low pressure circuit dedicated to spools actuation as well as the load

Figure 1. Baseline architecture of the reach stacker
sensing system are neglected because of their low energy consumption. Both are replaced by non powered lines transmitting only information.

In this paper the powertrain and the telescoping circuit are not subjected to any modification of their components. The lifting circuit on its side is modified to recover the lost energy during boom-downs. The actual system is based on two LS pumps (Figure 1 (1)) providing the desired flow to the actuators (5) via a proportional valve (2). We can notice that both lifting and telescoping circuit have regeneration valves (3) which are only used when unloaded. Oil outgoing from the rod chamber is added to the pump flow resulting in a faster extension than in normal operating. In loaded phases the reached pressure in the piston chambers would be too high that is why the rod chamber is in this case connected to the tank. During boom down the flow limiter (4) added to the lifting circuit avoids lowering overspeeds especially in loaded case. The hydraulic pressure is dissipated in this valve to maintain a maximal predefined lowering speed. The rest of the hydraulic energy is lost inside the proportional valve.

2.3. Work cycle

Reach stackers are working mostly on container storage areas and their task which is repetitive consists of unstacking and stacking containers from one place to another like unloading a truck and stacking the container in the storage area or the contrary. The duty cycle employed to simulate the functioning of the machine is divided into two parts. The first part represents an unstacking stage followed by a truck loading and the second part is exactly the contrary. This configuration enables the machine to face all different situations of rolling, lifting and telescoping at loaded and unloaded cases.

The translation speed profile remains always the same whereas the hydraulic side is highly variable. The initial position of the container (row, height) changes the required length and angle of the boom to handle the container so that the energy distribution between lifting and telescoping will be different for each case. Figure 2 shows an example of a working cycle in terms of vehicle speed, boom angle, boom length and load for a given initial position of the container. A change of this initial position modifies the cycle except the vehicle speed profile meaning that there are as many different duty cycle as container positions. Finally the work cycle also includes an obligatory boom position for translation. Indeed, the telescope must be completely retracted and the boom angle must be approximately 45° to avoid any unbalance situation which could be dangerous during steering or braking phases. Thus if a container is picked up at the second height the driver will be obliged to lift the boom until 45° in order to provide sufficient visibility to drive. As the boom is necessarily elevated for each cycle the interest in a regeneration system on the lifting circuit increases.

![Figure 2. Duty cycle](image)
3. STUDIED ARCHITECTURES

3.1. Presentation of the solutions

3.1.1. Solution 1: Meter-out control valve based system

The first solution illustrated in Figure 3 uses the inherent advantage of the lifting system. Indeed, when the boom angle decreases the necessary force to hold the load rises resulting in a higher pressure in the lifting cylinders. This intrinsic feature is suitable for the implementation of this solution because the pressure of the cylinders will rise together with the accumulator pressure, enabling the system to continue recovering a part of the energy unless the maximum pressure of the storage is reached.

A similar concept presented in [5] shows a regenerative system connected to the piston chamber of the lifting cylinders of an excavator. While boom-down a proportional valve and a pressure compensator adjusts the desired flow and transmits it either to an accumulator or to the suction line of the main pump so that its power demand for feeding other actuators is reduced. Thus the accumulator volume can be reduced because the major part of the regenerated energy is instantaneously reused by the pump. In our case the boom lowering is often carried out in a single operation. Moreover the volume of fluid to regenerate is much bigger than for an excavator. Those elements led to some modifications to adapt this system to our case. In [6] the authors present a similar architecture which demonstrated 10% fuel savings. However a digital valve package was used to achieve the energy recovery.

Here we use two piloted restrictors (6) and (7) to control the rod retraction speed. The flow goes from the lifting cylinders (5) to the accumulator (8) until its pressure is too high to keep the desired flow rate.

During a rod retraction, if the piston chamber pressure is higher than the accumulator pressure, the piloted flow restrictor (6) opens in such a way that it dissipates a part of the energy by creating a pressure difference in order to reach the accumulator pressure. The higher is the pressure difference between piston chamber and accumulator the smaller is the flow restrictor opening. While the valve (6) is not fully opened the restrictor (7) stay closed. A totally opened valve (6) means that the pressure difference is not high enough to hold the desired flow rate. At this time the valve (7) opens and dissipates the extra flow. As the cylinder pressure tends to rise during a rod retraction, the accumulator is still able to regenerate a part of the energy.

To reuse the previously stored energy, a variable displacement motor (10) is added to the engine shaft. The accumulator feeds the motor whose torque helps the ICE to drive the main pumps, the auxiliary components or the powertrain.
This system has the advantage of being very simple and the need of few extra components makes it a low cost solution. However the recovered energy might be low unless a huge accumulator is used.

3.1.2. Solution 2: ICE-depandanetransformer based system

Similar regeneration architectures to the second solution shown in Figure 4 have been introduced in the literature, however we can find some differences. In [7] a common shaft for all units is employed, nevertheless the authors are using the same unit to drive the actuator and to recover the potential energy thanks to a displacement controlled (DC) system. In this machine such a configuration would require too many DC units, therefore an independent motor-pump system (Figure 4 (8),(10)) is used to recuperate the energy and reuse it.

In [8] the same motor is used to recover and to reuse the energy and the main pump also has a double function, since it can either feed the actuators during normal operation or store the energy into accumulators during energy recovery phases. On the contrary the proposed layout permits to combine energy recovery of the lifting system with other actuations. Indeed, some functions like steering must remain available even during short periods. Thus the main pumps (2) and the recovery system (7-11) are completely independent. Finally the layout presented in [9] has an extra pump-motor attached to the ICE which can either store energy during low load actuations and release it in peak power operation or recuperate energy during overruns. This solution appears to be inappropriate as our case study is characterized by long periods of high power requirement.

During a boom lowering the on-off valve (7) opens to connect the piston chamber of the lifting cylinders to the motor. The torque produced is transmitted to the pump (10) which stores the energy in the accumulator (11). The motor can also assist the ICE (1) to drive the main units (2) if there is a power demand at the same time. The case of a boom lowering without any other power demand is disadvantageous because an important part of the motor torque is dissipated into the engine braking and the friction torque of the pumps. At low speed those unwanted parasitic torques are reduced and the recuperated energy is higher. Nevertheless the displacement of the motor and the pump will have to increase to absorb the entire flow rate.

Compared to the solution presented in 3.1.1 it is here possible to adjust the displacement of the recovery unit (10) permitting more energy to be recuperated even when the pressure at the cylinder side is different from the accumulator side.
3.1.3. Solution 3: ICE-independent transformer based system

The last solution studied (Figure 5) is also based on a motor-pump system (7-8) to recover the potential energy. In [10] the authors present a transformer based system and in [11], [12] a control strategy is developed but no energetic evaluation of the solution is proposed.

The idea is to reduce the displacement of the recuperation units by making them rotate faster. This is now possible thanks to the new location of the transformer which is directly integrated on the hydraulic line between the proportional valve (3) and the lifting cylinders (6). The recovery phases are independent from the engine braking and pump drag allowing more energy to be recuperated.

When the boom is lowered the pressure difference at each port of the motor creates a torque driving the variable displacement pump. If the accumulator pressure is not high enough to hold the load, then the pump displacement increases. The proportional valve (3) regulates the pressure difference at the motor side so that the load cannot drive the pump faster than desired. The stored energy is then released to the system by switching the valve (9). The accumulator (10) is discharged via an extra motor (11) whose variable displacement enables the assisting torque to be controlled.

3.1.4. Energy reuse strategy

For the solutions 1 and 3 an extra motor of 100 cm$^3$ is attached to the ICE shaft in order to help the engine to drive the main pumps or the powertrain. This small unit has several advantages: the first one is its low price which is a good point for an additional system. Secondly it has a low friction torque at zero displacement meaning that the impact when it is not used is low. Finally a high displacement unit would provide an important torque making the ICE work at low load conditions corresponding to bad efficiencies working points. The second solution meanwhile uses the recovery pump as a motor to achieve the same task.

3.2. Modelling and control

3.2.1. Meter-out control valve based system

As explained in section 3.1.1 the opening signal $u_6$ of the valve (6)(Eqn.3) depends on the pressure at each port but also on the joystick position $k_L$ establishing the cylinder speed instruction. The second valve (7) signal $u_7$ permits to keep the desired flow rate even if the accumulator pressure is too high (Eqn.8).
3.2.2. ICE-dependant transformer based system

The fixed displacement unit (8) makes its rotation speed target directly proportional to the rod retraction velocity. Equation (10) shows that the motor speed depends on different resistive torques due to the accessories, the engine braking and the main pumps. The displacement signal \( u_6 \) of the pump (10) is evaluated (Eqn.12) to control this rotation speed. Thus if the torque provided by the motor is entirely used by the main pumps, the accessories and the engine braking, then the recovery pump displacement is set to zero.

On the one hand the instantaneous reuse of the recovered energy by the accessories or the main pumps tends to downsize the storage capacity. But on the other hand the engine braking and the friction of the pumps at zero displacement induce parasitic torques which drastically reduce the energy savings.

\[
T_{m8} = \frac{V_{emb}(p_{A8} - p_T)}{20\pi}
\]

\[
J_{ICE} \frac{d\omega_{ICE}}{dt} = T_{m8} - T_{rp10} - T_{ICE} - T_p2 - T_{acce}
\]

At constant speed, \( \frac{d\omega_{ICE}}{dt} = 0 \)

\[
T_{rp10} = T_{m8} - T_{ICE} - T_p2 - T_{acce}
\]

\[
u_{10} = \frac{T_{m8} - T_{ICE} - T_p2 - T_{acce}}{V_{erp10}(p_{accu11} - p_T)} \eta_{m, rp10} \cdot 20\pi
\]

3.2.3. ICE-independent transformer based system

The evaluation of the needed pump displacement (Eqn.14) to ensure a sufficient resistive torque for the hydraulic motor is similar to the previous section. Nevertheless no additional torque is present since the shaft is independent of the ICE. Thus the pump displacement controls alone the boom lowering speed unless the resistive torque is not high enough. In this case the signal \( u_3 \) controls the proportional valve opening (Eqn.16) making the back pressure of the motor higher resulting in a lower torque transmitted to the pump.
4. SIMULATION RESULTS

Table 2. Sizing of the hybrid systems

<table>
<thead>
<tr>
<th>Element</th>
<th>Solution 1</th>
<th>Solution 2</th>
<th>Solution 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Recovery motor disp. [cm(^3)]</td>
<td>-</td>
<td>220</td>
<td>160</td>
</tr>
<tr>
<td>Recovery pump disp. [cm(^3)]</td>
<td>-</td>
<td>145</td>
<td>190</td>
</tr>
<tr>
<td>Energy reuse motor disp. [cm(^3)]</td>
<td>100</td>
<td>-</td>
<td>100</td>
</tr>
<tr>
<td>Accumulator volume [L]</td>
<td>150</td>
<td>150</td>
<td>150</td>
</tr>
</tbody>
</table>

The sizing steps for each solution are not detailed but the Table 2 presents the displacements of the pumps and motors. The sizing of the accumulator volume depends on the amount of energy to be stored but also on the selected operating pressure range. Considering a 45t load lowered from the 5th height to the ground led to highly oversized storage capacities, that is why the average container weight and height (25t, 3rd height) were considered instead, resulting in a storage capacity of 150L. This volume corresponds also to the maximal available space for accumulators inside the machine. Finally the pressure operating points are adapted for each solution to optimize the amount of recovered energy.
The three models have then been simulated considering the duty cycle previously presented. For each initial position of the container an energetic analysis was conducted but the focus has been made on the two first rows and the three first heights of container stacks since they represent most cases. Figure 6 shows fuel savings for each initial position of the container and each considered solution. The first and second solutions achieve between 5% and 11% fuel saving whereas the third solution is capable of consumption reductions up to 16%.

Solution 1 shows poor energy recovery possibilities due to its architecture without transformer. We can see on Figure 7(a) that during a boom lowering the recovery can be split into two phases. Between the point A and B the pressure difference between the cylinder and the accumulator is high and almost all the flow is transferred to the accumulator (Figure 7(b)). From point B to C the restriction connected to the tank starts opening wider inducing a flow rate to the accumulator much smaller. The remaining pressure difference corresponds to the pressures losses through the first restriction and the check valve. The system dynamics can theoretically be identical to the traditional solution, however we can see on Figure 7(b) that during the transition between a full and a partial recovery (around point B) the flow rate tends to slow down. Here the control of the system should be improved to overcome this situation.

Figure 7. Solution 1: Pressures and flows during a recovery phase (boom angle from 45° to 10° and container of 25t)

The second solution achieves better recuperation performances but only in certain conditions. Indeed the system is able to recover between 40% and 50% of the potential energy when a heavy container is lowered. However the duty cycle also has boom-downs without container. In these conditions the lift cylinder pressure is too low to drive the hydraulic motor due to the numerous resistive torques. The potential energy cannot be recovered and a traditional throttling valve is used to dissipate this energy.

Figure 8 shows the torque distribution provided by the recuperation motor. From point D to E the desired lowering speed is not reached that is why the pump is not activated meaning that no energy is stored. Nevertheless about 30% of the torque is immediately reused by the accessories but the rest is dissipated through the pumps friction at zero displacement, the torque converter, the engine braking and the acceleration of the inertias. Furthermore a clearly identified drawback is the slow dynamics brought by uncontrolled resistive torques.

From point E to F the pump controls the lowering speed via its displacement and the energy storing begins. The recuperated energy on the Figure 6 is calculated by comparing the total recoverable potential energy with the stored energy plus the immediately reused energy. We can also see that the fuel savings are similar to the first solution despite of the better recuperation performances. In fact the friction losses at zero displacement generated by the additional motor and pump bring an extra torque to the ICE which consumes more fuel outside the recovery phases. This side effect decreases strongly the benefits brought by this system.
The last proposed architecture solves a part of the problem by using only a small unit attached to the ICE. The friction losses are much smaller compared to the solution 2. We can observe on Figure 9 (d) that the motor consumes few energy when it is not used. Between the point G and H a container of 25t is lowered generating a torque to the pump which fills the accumulator. The pump starts the recovery phase at full displacement as the accumulator pressure is low and decreases gradually until 30% since the cylinder pressure increases slower than the accumulator pressure. In a second time the previously stored energy is released via the motor to help the ICE to drive the main pumps for a further boom lifting. From point I to J the boom is lowered from the driving position (boom angle at about 45°) to the container release position and between the point J and K the boom is lowered without any container. We can see that it is still possible to recover energy thanks to the variable displacement pump. This feature together with the independence from the engine breaking and the other parasitic torques lead to better performances. We can see on Figure 6 that the rate of recovered energy is almost twice compared to the two first solutions.

Figure 8. Solution 2: Torques distribution during a recovery phase (boom angle from 45° to 10° and container of 25t)

Figure 9. Solution 3: (a) Boom angle, (b) Regeneration pump torque, (c) Accumulator pressure, (d) Helping motor torque
5. CONCLUSION AND FUTURE WORK

In this paper three systems dedicated to potential energy recovery have been modelled and simulated. Each solution was implemented on the physical model of a container handling machine whose high potential for energy recovery is suitable for hybridization.

The first architecture based on a meter-out valve demonstrated on average 30% of energy recuperated compared to the total recoverable energy and 8% fuel savings. In spite of poor energy performances, this solution requires few additional components and the control system remains relatively simple. The solution with a pressure converter coupled to the ICE showed better performances in terms of energy recuperation but the fuel savings are similar to the previous architecture. This is mainly due to the friction losses generated by the additional motor and pump continuously driven by the engine. Moreover the extra pump for energy storing induces a more complex system and a higher cost of the system whereas the fuel savings are low. The last studied solution is based on the decoupling of the regeneration system from the ICE. This architecture achieves on average 65% of recovered energy and the consumption reduction can reach 16%. The great advantage is the possibility to recover energy regardless of the container weight meaning that the fuel savings are more independent from the duty cycle.

In the near future strategies concerning power management and more particularly the energy reuse will be studied deeper to improve the entire system. Moreover the most promising solution with the independent transformer based recuperation system will be implemented on a machine in order to validate the simulation results. The dynamics of a boom lowering with energy recuperation will also be compared with the traditional system to confirm the promising performances of the hybrid machine.

NOMENCLATURE

\[ A_i \quad \text{Flow area of the valve (i)} \quad \text{[m}^2\text{]} \]
\[ A_{\text{max}, i} \quad \text{Maximum flow area of the valve (i)} \quad \text{[m}^2\text{]} \]
\[ C_{\text{qCV}} \quad \text{Check valve discharge coefficient} \quad [-] \]
\[ C_{\text{qL}} \quad \text{Discharge coefficient of the valve (i)} \quad [-] \]
\[ i_{\text{ICE}} \quad \text{Total inertia on the engine shaft} \quad \text{[kg/m}^2\text{]} \]
\[ i_7 \quad \text{Inertia of the recuperation motor (7)} \quad \text{[kg/m}^2\text{]} \]
\[ i_8 \quad \text{Inertia of the recuperation pump (8)} \quad \text{[kg/m}^2\text{]} \]
\[ k_L \quad \text{Joystick position} \quad [-] \]
\[ \eta_{\text{m,rp10}} \quad \text{Mechanical efficiency recuperation pump (10) system 2} \quad [-] \]
\[ \eta_{\text{m,rp8}} \quad \text{Mechanical efficiency recuperation pump (8) system 3} \quad [-] \]
\[ \eta_{\text{m,m8}} \quad \text{Mechanical efficiency recuperation motor (8) system 2} \quad [-] \]
\[ p_{\text{Ai}} \quad \text{Pressure at port A of the component (i)} \quad \text{[bar]} \]
\[ p_{\text{Bi}} \quad \text{Pressure at port B of the component (i)} \quad \text{[bar]} \]
\[ p_T \quad \text{Tank pressure} \quad \text{[bar]} \]
\[ \rho \quad \text{Hydraulic fluid density} \quad \text{[kg/m}^3\text{]} \]
\[ p_{\text{accu,i}} \quad \text{Pressure of the accumulator (i)} \quad \text{[bar]} \]
\[ q_{\text{accu,i}} \quad \text{Fow rate at the inlet port of the accumulator (i)} \quad \text{[L/min]} \]
\[ q_{\text{max}} \quad \text{Maximum lifting cylinders output flow rate} \quad \text{[L/min]} \]
\[ T_{\text{acc}} \quad \text{Accessories torque} \quad \text{[N.m]} \]
\[ T_{\text{ICE}} \quad \text{Engine torque} \quad \text{[N.m]} \]
\[ T_{\text{mi}} \quad \text{Torque of the motor (i)} \quad \text{[N.m]} \]
\[ T_{\text{pi}} \quad \text{Torque of the pump(s) (i)} \quad \text{[N.m]} \]
\[ T_{\text{rpi}} \quad \text{Torque of the recovery pump (i)} \quad \text{[N.m]} \]
\[ u_3 \quad \text{Valve (3) input signal (system 3)} \quad [-] \]
\[ u_6 \quad \text{Valve (6) input signal (system 1)} \quad [-] \]
\[ u_7 \quad \text{Valve (7) input signal (system 1)} \quad [-] \]
\[ u_8 \quad \text{Percentage of pump (10) displacement system 2} \quad [-] \]
REFERENCES


RESEARCH ON PULSATION DEMAND OF PUMP USED IN ELECTRO-HYDROSTATIC ACTUATOR

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ABSTRACT

Conventional centralized aircraft hydraulic system has very high demands on the flow output pulsation of the engine driven pump (EDP). However, few papers have mentioned the requirements on the high speed pump used in Electro-Hydrostatic Actuator (EHA). The performance of EHA is important in electro-hydraulic servo system. It is known the pump used in an EHA has a huge influence on its performance especially the flow pulsation of the pump lowers EHA performance. Therefore, the focus is to investigate the pulsation demand of the pump according to the performance of EHA. By using AMESim, a common EHA model is established to simulate the working conditions of EHA and the pump unit. In this paper, the working parameters of the pump are adjusted while the system performance characteristics are observed and analysed. And the pulsation demand mentioned can be concluded through reverse analysis. System performance of EHA such as position tracking error, dynamic response, dynamic stiffness and force loop are evaluated in simulations. Meanwhile, the acceptable range of pump flow pulsation is analysed. That is, the pulsation demand is given according to the EHA performance requested.

KEYWORDS: Flow Pulsation Demand, Electro-Hydrostatic Actuator, EHA Performance

1. INTRODUCTION

Axial piston pump is widely used in hydraulic system and it has an influence on the overall system performance. Because of the pump structure and working principle, its output flow fluctuates near the expected value, which is called flow pulsation and could have bad effect on the hydraulic system [1-2]. So the pulsation demand should be investigated according to the system performance. For example, in the conventional aircraft hydraulic system, the output flow pulsation of the engine driven pump (EDP) should be less than a certain value [3]. But few papers request the pump's pulsation used in EHA. Because of the power-by-wire layout, there is no complicate hydraulic pipeline in the EHA actuation system compared with conventional centralized hydraulic system. Therefore, the EHA actuation system has lots of advantages such as more simplified circuit and efficient motion control.

Compared with EDP, EHA pump has the characteristics of its own. In consider of the integrated structure design, EHA pump displacement is relatively lower. And in order to ensure that the pump has enough flow output, the EHA pump rotation rate will be higher. Cylinder of EHA is commonly bidirectional symmetry because the EHA is always bidirectional working. As EHA may run in complicate working condition, the pump rotation rate is always variable. The variable speed leads to variable frequency, so the flow pulsation is more complicate. All these EHA pump characteristics make the pump research meaningful.

And with more and more attention paid to further industrial applications of EHAs, it is also necessary to propose the pump pulsation demand of EHAs. So in this paper, the EHA pump flow pulsation and its influence on EHA performance is researched. First of all, a classic EHA model is established in AMESim. Then the reason and result of pump flow pulsation is analyzed and the flow pulsation is also simulated in AMESim. The flow pulsation is added into the classic EHA model to analyze its influence on the EHA
performance for example position control performance such as position tracking error, dynamic response
dynamic stiffness and force control performance such as static and dynamic force simulation. At the same
time, we have theoretically analyzed flow pulsation's influence on EHA performance and finally come to a
conclusion.

2. MODELING

In this section, a classic schematic diagram is given, then the operation principle is analyzed. And the classic
EHA model is established through AMESim according to the analysis [4-5]. All the necessary segments of
EHA have their corresponding sub model in the model, meanwhile the structure parameters of EHA
components are studied to ensure that the EHA has a certain level of performance. Next, the flow pulsation
of the pump is analyzed to prepare for the simulation. Then typical performance of EHA has been put
forward to study flow pulsation's influence on it.

2.1. EHA Modeling

A schematic diagram of the conventional EHA is shown in Figure 1. In the picture, the main frame of EHA
consists of a servo motor, a pump and a hydraulic cylinder [6]. Because linear reciprocating movement can
be accomplished by the three components, and the fluid compensating circuit, the bypass valve and the
relief valves added into the loop will improve the system performance and safety. The accumulator is used to
compensate for the fluid leakage of EHA and maintain a certain pressure of the circuit. When the pressure
difference of the two chambers is too high, one of the relief valves will open to keep a relatively low pressure
difference. The bypass valve can be switched in case of emergency. The controller receives the command
signal from computer, the position feedback signal of the cylinder. After dealing with these signals, the
controller then gives the servo motor a voltage control signal in order to achieve servo control for linear
motion or closed-loop pressure control.

![Figure 1. Schematic Diagram of EHA](image1)

According to the schematic diagram of EHA, a simulation model is established in AMESim as shown in
Figure 2. The EHA model can be divided into three parts: controller, hydraulic circuit and the load system
respectively drawn in three wire frames.

![Figure 2. EHA AMESim Model](image2)
After the EHA model established, all the structure parameters of EHA must be set up properly. Therefore, based on a certain type of EHA, the main simulation parameters of the model are listed as follows in Table 1.

<table>
<thead>
<tr>
<th>Table 1. Part of Simulation Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Hydraulic Pump</strong></td>
</tr>
<tr>
<td>Pump Displacement</td>
</tr>
<tr>
<td>Typical Speed of Pump</td>
</tr>
<tr>
<td>Check Valve</td>
</tr>
<tr>
<td>Flow Rate Pressure Gradient</td>
</tr>
<tr>
<td>Hysteresis for opening/closing</td>
</tr>
<tr>
<td>Hydraulic Orifice</td>
</tr>
<tr>
<td>Characteristic Flow Rate</td>
</tr>
<tr>
<td>Corresponding Pressure Drop</td>
</tr>
</tbody>
</table>

2.2. EHA Performance

The way of researching pump's pulsation demand of EHA is to request EHA's performance first. That is, what a flow pulsation range is allowed according to the expected performance of EHA. And we can research how pump's flow pulsation influences EHA's performance to get their change relation and put forward pulsation demand.

The most commonly used and important performance of EHA includes its position tracking error, dynamic response, dynamic stiffness of position control and force loop [7-8]:

1. Position tracking error: position control is widely used in various servo systems. Its control accuracy is just what we need, so the position tracking error directly determines systems' performance.
2. Dynamic response: sometimes EHA's response to various signals needs analyzing when we have fatigue test. Then the good dynamic response is expected.
3. Dynamic stiffness: it is the structure's ability to resist deformation in a certain dynamic interference. Cause the work environment of the EHA may be complex, not only the static stiffness but also the dynamic should be taken into consideration.
4. Force loop: although force loop is seldom used in aircraft actuation system, it is needed in many other civil fields. For example, force control is always used in loading system when having a structure test and structure test is widely used in industrial manufacture. In this working condition, loading test directly depend on force control accuracy. So it is necessary to take the force loop into consideration.

2.3. Pulsation Analysis

In section 2.1, a classic EHA AMESim model has been established. But the pump component of model is ideal. It is necessary to add a flow pulsation simulation segment into the EHA model to simulate the pump's real work condition. And we can adjust the flow pulsation situation according to test demand to see how it affects the performance of EHA talked in section 2.2.
First, we have to analyze the reason why pump's flow fluctuates. One of the reason is that the axial piston pump's special structure and working principle [9-10]. And it is called the pump's geometric flow pulsation. Figure 3 is the axial and radial sectional view of the pump. For one piston, it starts with $\phi=0$, when the cylinder has a rotation of $\phi$, the piston has a displacement of:

$$ s = AB - CD = R\tan \beta (1 - \cos \phi) $$

(1)

All the letters have been defined in Figure 3. $\beta$: swash plate angle.

Then the flow of the piston is:

$$ Q = A s' = m^2 R \omega \tan \beta \sin \phi $$

(2)

$A$: area of the piston section, $r$: radius of the section, $\omega$: rotation rate of the cylinder.

So the instantaneous flow of the odd piston pump is:

$$ Q = \pi r^2 R \omega \tan \beta \sum_{k=1}^{i} \sin(\phi + 2\pi (k - 1)/z) $$

$$ = \pi r^2 R \omega \tan \beta \cos \left[ \phi - \left( m + 1 \right) \pi / 2z \right] \csc(\pi / 2z) / 2 $$

while $m\pi / z \leq \phi < m\pi / z + \pi / z$

(3)

Where, $z$: number of the common odd pump's piston, $m$: natural number, $i$: number of the piston located in the flow out area of the valve plate.

When $\omega$ is a constant, the discipline that flow of the pump varies along with the cylinder's angle is in Figure 4.

In fact, besides pump's geometric flow pulsation, there are many other reasons lead to the pump's flow fluctuating such as the complex flow switch area change and the backward flow [11-12]. So the real flow pulsation is more severe and complex and all these influence factors of pump's pulsation will be taken into
consideration to reduce a working situation as accurate as possible. The accurate pump's pulsation simulation makes the modeling simulation more reliable.

According to pulsation analyzing, it is known that the flow is function of the angle $\phi$. But $\phi$ is just an intermediate variable in the simulation. It is equal to the integral of rotation rate of the pump. So the flow is in fact a function of rotation rate: $\omega$. Pump's flow pulsation has been simulated like in Figure 5 and the speed signal comes from the motor feedback. The simulation flow pulsation consists of geometric pulsation and others. The former contains time, rotation rate feedback and structure parameters. In order to simplify the complexity of the function, it is replaced by absolute value of a sine function. It has the same amplitude and frequency as the curve in Figure 4.

While the cylinder rotating, a certain piston chamber moves from oil inlet to outlet of the valve plate, the oil pressure is low at the beginning, so the high pressure oil will flow into the outlet. Then there is a backward flow phenomenon. And the backward flow is simulated by negative absolute value of a cosine function. The frequency is half of the former and the amplitude depends on the situation of the backward flow. The accurate flow pulsation simulation is complex and also hard to simulation, and the flow function has been simplified, so it is not necessary to get all the other accurate pulsation factors. On the other hand, the backward flow and geometric flow have a direct effect on the pulsation rate which we focus on most, so just these two most important influence factors are taken into consideration.

![Figure 5. Flow Pulsation Simulation](image)

2.4. Theoretical Analysis

A theoretical flow pulsation and EHA transfer function analysis is firstly researched (referring Figure. 1). Brushless DC motor:

$$
\begin{align*}
U &= E + L \frac{di}{dt} + Ri \\
E &= K_c \omega \\
T_e &= K_i i \\
T_e &= J \dot{\omega} + T_f
\end{align*}
$$


The transfer function of motor can be expressed as:

$$
\omega(s) = \frac{1}{K_c + (Ls + R)/K_i} T_e(s) + \frac{(Ls + R)/K_i}{K_c + (Ls + R)/K_i} T_f(s)
$$

(5)
Hydraulic pump model:

\[
Q_i = D\omega - K_{ip} (P_1 - P_2)
\]  

\[
Q_2 = D\omega - K_{ip} (P_1 - P_2)
\]  

\[
Q = \frac{1}{2} (Q_1 + Q_2) = D\omega - K_{ip} (P_1 - P_2)
\]  

Where, \(Q_i\) : flow rate at A1, \(Q_2\) : flow rate at B1, \(Q\) : flow rate at load, \(K_{ip}\) : internal leakage coefficient, \(P_i\) : pressure at A1 port, \(P_2\) : pressure at B1 port, \(D\) : displacement of pump, \(\omega\) : rotate speed of pump,

Hydraulic cylinder model:

\[
Q_1 = K_{ia} (P_1 - P_2) + \frac{V_A}{\beta_e} \dot{P}_1 + A\ddot{x} + K_{ib} P_1
\]  

\[
Q_2 = K_{ia} (P_1 - P_2) - \frac{V_B}{\beta_e} \dot{P}_2 + A\ddot{x} - K_{ib} P_2
\]  

\[
Q = \frac{1}{2} (Q_1 + Q_2) = K_{ic} (P_1 - P_2) + \frac{V_0}{4\beta_e} (\dot{P}_1 - \dot{P}_2) + A\ddot{x}
\]  

\[
A(P_1 - P_2) - F = M\ddot{x} + B\dot{x} + Kx
\]  

Where, \(Q_i\) : flow rate at A2 port, \(Q_2\) : flow rate at B2 port, \(K_{ia}\) : internal leakage coefficient, \(K_{ib}\) : external leakage coefficient, \(K_{ic} = K_{ib}/2 + K_{ia}\), \(V_A\) : volume of chamber A2, \(V_B\) : volume of chamber B2, \((V_A + V_B = V_0)\) \(A\) : area of piston, \(\beta_e\) : modulus of hydraulic fluid, \(\ddot{x}\) : displacement of piston, \(F\) : external force, \(M, B, K\) : mass, damping coefficient, stiffness of the equivalent load.

Then the block diagram can be got as Figure. 6.

**Figure 6. Block Diagram**

Defining \(P_L = P_1 - P_2\), \(\delta\) as flow pulsation, the transfer function about \(\delta\) and \(E_\delta\) can be get by Laplace transform ignoring the friction torque. The flow pulsation has effect on the control signal \(E_\delta\). It lowers the system performance, so it's necessary to investigate the relations.

\[
\frac{E_\delta(s)}{\delta} = \frac{-A}{\left(\frac{V_0}{4\beta_e} s + K_{ia} + K_{ip}\right)(Ms^2 + Bs + K) + A^2 s + \frac{AD}{K_c + (Ls + R) \frac{J}{K_i} s}}
\]  

(13)
3. FLOW PULSATION AND PERFORMANCE ANALYSIS

The commonly used EHA performance has been discussed in chapter 2, and the complete EHA model including flow pulsation is ready for simulation. So in this chapter we will simulate and analyze how the flow pulsation affects these performance and draw a conclusion.

3.1. EHA for Position Control

In the aircraft actuation system, actuators are used to control the rudders. So the position control is important. It is necessary to have a position control simulation and analysis.

3.1.1. Position Tracking Error

Firstly, the structure parameters of EHA model have been set up partly according Table 1. And we adjust the model's control parameters to make the system's step response well but without adding flow pulsation simulation segment. Then the pulsation simulation has been taken into the model. And we slowly change the flow pulsation control parameters to see how flow pulsation affects the position tracking error of the EHA.

Figure 7.

Figure 7 is the signal information after giving a step instruction to the EHA model. Figure 7(a) is the pump's flow, and three curves in the picture are respectively real flow, ideal flow and simulation flow pulsation from top to bottom. Figure 7(b) is the step response of the system with flow pulsation. After calculation, flow pulsation in the picture is about 10% which leads to a 0.15% position accuracy drop. Also many other values of flow pulsation have been set up into the model to get various relations between them. About 20% of flow pulsation value leads to a 0.185% position accuracy drop. At last, all the simulation data are drawn in a picture and fitted into a continuous and smooth curve via MATLAB as shown in Figure 8.

Figure 8.

Figure 8. Relation between Pulsation and Position Accuracy Drop
3.1.2. Dynamic Response

Just as position tracking error simulation, all the structure and control parameters are set up well to ensure that the system has a good dynamic response without flow pulsation simulation segment. And the sine curve is used as the test signal. Next the pulsation simulation segment has been added into the EHA model. Then the value of the flow pulsation has been slowly changed to watch the relation between the flow pulsation and the system’s dynamic response. On the other hand, for a certain value of flow pulsation, we can replace the sine curve with sweep frequency signal. Then the dynamic response of the system in different situations can be got.

Figure 9. Dynamic Response Simulation

Figure 9 is the result of EHA dynamic response simulation. Figure 9(a) is pump's flow, and three curves in the picture are respectively real flow, ideal flow and simulation flow pulsation from top to bottom. Figure 9(b) is the dynamic response of the system. The red one is dynamic response with flow pulsation. It is obvious that the magnitude attenuates with flow pulsation. After calculation, the magnitude of green curve is -3 dB when the frequency is 4.9 Hz without flow pulsation, greater than 2.8 Hz of red curve with about 13% flow pulsation. When flow pulsation is about 25%, system bandwidth drop to 1.3Hz. So the flow pulsation lowers the bandwidth of the system. When the dynamic response of the system is needed, we have to pay attention that the flow pulsation of the pump should be in a certain range.

3.1.3. Dynamic Stiffness

In dynamic stiffness simulation, all the parameters of the EHA model have been set up to ensure that the system has a good step response like accuracy simulation. The difference is that a variable frequency signal has been add into the load system one second after the step signal has been add into the control system. Like other simulations, the control parameters of the pulsation simulation segment have been regulated to see the flow pulsation's influence on the EHA's dynamic stiffness.

Figure 10. Dynamic Stiffness Simulation
Three curves in figure 10(a) are respectively real flow, ideal flow and simulation flow from top to bottom. Curve in figure 10(b) is the system’s step response when flow pulsation and variable frequency signal are taken into consideration at the same time. As bode diagram is not so clear that the time domain response is analyzed. Compared with the step response without flow pulsation simulation, the maximum amplitude of the step response with pulsation is larger when the simulation frequency relatively low. So the flow pulsation decreases the EHA’s dynamic stiffness to a certain extent.

3.2. EHA for Force Control

In aircraft actuation systems, actuators are used to control rudders by controlling position. But in many other industrial manufacture fields, force control in also widely used such as loading system. So static and dynamic force loop performance is researched. In force loop simulation, force signal of cylinder becomes feedback signal instead of position signal and the load system becomes fixed end. So control parameters are reset and then adjusted in order to ensure that system has a good force loop without flow pulsation. Next, pulsation simulation segment is added into the system. By adjusting flow pulsation parameters, we can get the flow pulsation's influence on the system force loop.

After simulation, it is found that in static force simulation, although flow pulsation has influence on the force response curve from step signal, it has nothing to do with the force tracking error. And its effect on the curve before stable is also faint.

![Figure 11. Dynamic Force Simulation](image)

Dynamic force simulation is shown in Figure 11. Fixing a certain value of the pulsation rate, we can give the system a sweep frequency signal input, and the frequency response can be got by FFT. Figure 11(a) describes different flow rate and figure 11(b) is the frequency response of the system. It can be seen from the picture that the red curve with pulsation is relatively lower than the green one. That means a performance drop. After calculation, when the magnitude is -3dB, the frequency of green curve without pulsation is about 11Hz and that of the red curve is about 9.3Hz. So the bandwidth of the force control system is reduced. About 15% flow pulsation rate leads to a 1.7Hz bandwidth drop, and the greater the pulsation rate is, the more the bandwidth drop. On the other hand, a certain frequency response curve is unsmooth and has a worse phase lag with pulsation in time domain. And the curve will be more inaccurate when the flow pulsation intensifies.

4. CONCLUSION

Through establishing the EHA's AMESim classic model and simulating pump's real flow pulsation, how flow pulsation affects EHA's performance is simulated.
The pump flow pulsation theoretical influence on EHA is analyzed through researching EHA transfer function and how the flow pulsation affect the EHA transfer function. Referring to the EHA model, the EHA transfer function with flow pulsation is researched.

In position tracking error simulation, about a 10% flow pulsation leads to a 0.15% position accuracy drop and a 20% flow pulsation leads to a 0.185% position accuracy drop. Continuous impact relation has been drawn in Figure 7.

In dynamic response simulation, the flow pulsation of pump has influence on the system's dynamic response. It lowers the magnitude of dynamic response bode diagram. So the system bandwidth is reduced. About 13% flow pulsation leads to a bandwidth of 1.3Hz drop and 25% pulsation leads to a 2.8Hz bandwidth drop. Although the relation of them may not be liner, we can still propose the pulsation demand by frequency response performance. And excessive flow pulsation directly leads to system's response distortion.

In dynamic stiffness simulation, the flow pulsation results in EHA's dynamic stiffness drop when the simulation signal is in low-frequency stage. And the extent of influence is slight.

In force loop simulation, the flow pulsation has little influence on the static force tracking error. But it reduces the bandwidth of the dynamic system. About 15% flow pulsation rate leads to a 1.7Hz bandwidth drop. And it makes the dynamic response curve delay and become unsmooth.

In different systems, the extent of influence that flow pulsation has on the performance of EHA varies. So all the performance should be taken into consideration. We have got their impact relations through modeling and simulation. Then the pulsation demand of pump used in the EHA can be obtained when the performance of the system is given.

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REFERENCES


REVIEW OF HYDRAULIC TECHNOLOGIES IN WIND TURBINES

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ABSTRACT
The reliability of wind turbines is receiving increasing attention as the scale of wind turbines gets larger. In modern wind turbines, hydraulic transmission plays an important role in keeping the turbines operating safely and steadily. The paper analyses the utilization of hydraulic technologies in brake system, pitch control system and power transmission system. In particular, the hydro-mechanical hybrid transmission is presented in detail, which combines the high efficiency of mechanical transmission with power output stability of hydraulic transmission, and becomes a new research trend in wind turbines.

KEYWORDS: hydraulic technology; wind turbine; hydro-mechanical transmission

1. INTRODUCTION
As one of the most predominant and technologically sophisticated power generation technology, with best prospects for commercial development in renewable energy, wind power generation has received considerable attention worldwide in recent years [1].

Hydraulic technologies are mainly used in the following parts of wind turbines: 1) Hydraulic brake system, which includes main shaft brake system and yaw brake system. It is the most common used hydraulic system and relates directly to the operational safety in wind turbines. 2) Pitch control system. In the early development stage of the modern wind turbines, fixed pitch technology was adopted and the blade tip spoiler is used to slow down the blade through hydraulic control. As wind turbines grow larger and larger, hydraulic pitch control technology is widely used, which can not only slow down the blade but also stabilise the output power. 3) Power transmission system. Hydraulic transmission is one of the transmission modes in wind turbines and its working fluid is hydraulic oil. Compared with mechanical transmission, hydraulic transmission has the merits of small structure, small inertia and large torque output. Operational experience shows that the gearboxes in the modern MW level wind turbines are their weakest-link-in-the-chain component [2]. Therefore, researchers are advising on the possibility of replacing the gear transmission with hydraulic power transmission. Based on the authors’ long-time researches in wind turbines and hydraulic technologies, the paper generalises the hydraulic technologies in wind turbines.
2. HYDRAULIC BREAK TECHNOLOGY IN WIND TURBINES

2.1. Hydraulic yaw system and its brake technology

Hydraulic yaw system and its brake system are the key components in horizontal axis wind turbines. The hydraulic yaw system drives the nacelle to rotate around the axis of the tower. The yaw system has two roles. One is to keep the swept area of rotor perpendicular to the wind direction when the wind velocity is below rated. When the wind speed is above the rated wind velocity, the hydraulic yaw system will drive the swept area of rotor parallel to the wind direction. When the nacelle is in the right direction, the hydraulic yaw system will offer necessary locking torque to guarantee the safety of wind turbines. The other role is to untwist the cable automatically when the cable is twisted after continuous wind direction tracking [3]. To provide a stable brake process, at least 6 brake discs are needed in the damping brake of the hydraulic yaw brake system and they should be distributed symmetrically.

![Figure 1. Schematic of hydraulic yaw brake system](image)

| 1, 13, 14 Oil filter | 2, 3, 8 Check valve | 4 Throttle valve | 5 Directional valve | 6, 9 Pressure relief valve | 7, 10 Stop valve | 11 Pressure sensor | 12 Accumulator | 15 Switching valve | 16 Hydraulic actuator | 17 Oil tank | 18 Hydraulic pump |

The schematic of hydraulic yaw brake system is shown in Figure 1. When the wind direction changes, the yaw motor will drive the nacelle to follow the wind. To make the yaw process smoother, the reversing valve 5 energises and the unloading valve 15 de-energises. The yaw brake clamp does not grip the brake discs under the back pressure provided by relief valve 6. After the nacelle turns to the right position, the reversing valve 5 and the unloading valve 15 de-energise, and the yaw brake clamp grip the brake discs tightly. During the cable untwisting or manual yawing procedure, the reversing valve 5 and the unloading valve 15 both energise and the clamp opens fully to eliminate the wear of the brake pad.

2.2. Main shaft brake technology

The main shaft brake system is used to halt the wind turbine. The working principle of the main shaft brake system is to grip the brake disc using hydraulic clamp. Different wind turbine manufacturers employ different hydraulic clamps and the number and mounting position are also different. The clamp can be located on high speed shaft or low speed shaft. Most wind turbines locate it on high speed shaft because locating on low speed shaft will demand larger brake torque. The fixed pitch wind turbines generally employ two clamps because the main shaft brake is the only braking measure which can be used, while the variable pitch wind turbines...
generally adopt one clamp as they can first depress the load through pitch control and then the main shaft brake system is applied [4].

![Figure 2. Schematic of main shaft brake system](image)

The schematic of main shaft brake system in fixed pitch wind turbines is shown in Figure 2. The brake is clamped by the spring force and released by the hydraulic force. When the wind turbine is running, the hydraulic pump is in operation and the solenoid directional valve energises. The high pressure oil goes into the hydraulic cylinder through the check valve and the clamps release. When the wind turbine performs a normal shutdown, the blade tip spoiler is cast off and the pump stops working. After the rotor speed is lower than the set value, the directional valve de-energises and the brake is clamped by the spring force. When the wind turbine performs an emergency shutdown, the hydraulic pump stops working and the blade tip spoiler is deployed. Meanwhile, the directional valve de-energises and the brakes are clamped by spring force.

2.3. Main shaft soft brake technology

During the braking process, the torque generated by the braking force acts on the main shaft, which results in load impact on the gearbox and thus shortens the working lifetime of it. For a fixed pitch wind turbine, though the unfolding the blade tip spoiler is used, it can only generate limited aerodynamic resistance and the main shaft brake is necessary. Some methods need to be taken to reduce the torque variation of the main shaft and the wear of the brake pad. In the early years of 21st century, the German manufacturing company Svendborg Brakes developed a wind turbine Soft Braking Control in which the control unit adjusts the braking force in real time according to the rotor speed [5].

The hydraulic schematic of the soft brake system developed for fixed pitch wind turbines is shown in Figure 3. Compared with the normal brake system in Figure 2, there is a high speed switching valve 9 and a directional valve 10 between the brake cylinders and the tank. During normal shutdown, the blade tip spoiler starts to work to decelerate the rotor speed rapidly. When the rotor speed is lower than the set value, the main controller gives a brake signal to start the hydraulic soft brake. PLC controller gives a Pulse Width Modulation (PWM) control signal according to rotor speed error, which can be obtained according to measured rotor speed and the predefined wind velocity-rotor speed curve, to regulate the on-off time of the high speed switching valve 9. In an emergency shutdown, the wind turbine unloads through the directional valve 10 and the brake is clamped by spring force in a short period. In addition, if the high speed switching valve 9 fails, the directional valve 10 can also be used for the brake.
3. HYDRAULIC PITCH CONTROL TECHNOLOGY IN WIND TURBINES

3.1. Blade tip spoiler in fixed pitch wind turbines

The blade tip spoiler is also known as a pneumatic brake. There is a hydraulic cylinder at the end of each blade, and the tip spoiler is rotated by the tip shaft which connects the piston rod of the hydraulic cylinder and the tip spoiler [3] (Figure 4). In the normal operation, the piston rod of the hydraulic cylinder moves to rotate the tip shaft and folds the tip spoiler to adjoin the blade. When the wind turbine wants to be shut down, we need to unload the hydraulic system and the tip spoiler will cast off to be perpendicular to the blade by the centrifugal force and the spring force. The resulted aerodynamic damping will reduce the rotor speed [6].

The hydraulic schematic of a typical blade tip spoiler is shown in Figure 5. When the wind turbine is started, the two-way solenoid valve is energised to block the fluid in the hydraulic cylinder from flowing back into the tank. The hydraulic pump works to output the high pressure oil to the hydraulic cylinder. The blade tip spoiler is folded and the pressure keeps increasing. When the oil pressure rises to the set value of the overvoltage relay, the motor is stopped by the controller after a time delay. During the delay time, the pressure continues...
to rise and reach the relief pressure of the relief valve. Then the relief valve opens and the system pressure will not rise. Because of the leakage, the pressure in the hydraulic cylinder will drop. When the pressure is lower than the set value, the overvoltage relay will emit a signal to restart the motor and pump to compensate the pressure. When the generator output power exceeds the maximum power, the two-way solenoid valve de-energies and the oil in the hydraulic cylinder flows back into the tank. And the blade tip spoiler will be deployed to decrease the speed of the rotor.

![Diagram of hydraulic system](image)

**Figure 5. Schematic of a typical blade tip spoiler**

In application of the hydraulic in the wind turbine, the hydraulic system is integrated. The hydraulic yaw brake system, the soft brake system and the tip spoiler system will share a hydraulic pump [7] which is shown in Figure 6.

![Photo of hydraulic system in wind turbine](image)

**Figure 6. The integrated hydraulic system in wind turbines**

### 3.2. Hydraulic collective pitch control technology

There exists a rated power for each wind turbine due to the design requirements. When the wind speed is higher than the rated value, the fixed pitch wind turbines will stall the blades to reduce the captured power, which, however, greatly lowers the operating efficiency of the system. Variable pitch control is a key technology in large-scale wind turbines. By changing the blade pitch angle, the output power of the generator remains near the rated [8]. There are two ways of pitch control, one is by servo motor and the other is by hydraulic.
Vestas Wind Systems A/S has been using the hydraulic pitch control while in China almost all wind turbine manufacturers employ the servo motor pitch control. However, with wind turbines growing larger and larger, manufacturers are considering hydraulic pitch control.

Figure 7. Structure of collective pitch control

Hydraulic pitch control can be sorted into the collective pitch control and the individual pitch control. The structure of collective pitch control is shown in Figure 7. The hydraulic cylinder rod is connected to a turn plate and the turn plate is linked with the three blades by link mechanism. When the rod moves, the three blades will rotate simultaneously and the displacement of the rod is linear with the blade pitch angle.

Figure 8. Schematic of collective pitch control system

The hydraulic schematic of collective pitch control is shown in Figure 8. In the pitch control process, the electro-hydraulic proportional directional valve regulates the displacement of the hydraulic cylinder so as to adjust the
blade pitch angle and power output. There is a differential circuit in the hydraulic system. In the blade feathering, the oil from the rod chamber and the pump both flow into the head chamber to increase the pitch rate. When the system failure occurs or an emergency stop is needed, the pump is shut down immediately. The oil for feathering is provided solely by the accumulator. If the hydraulic oil in the accumulator is not enough for a 90° feathering, the wind will drive the blade to the ideal position.

![Figure 9. Schematic of redundant digital electro-hydraulic proportional system with high speed switching valve in parallel](image)

1 Electro-hydraulic proportional directional valve 2 High speed switching valve 3 Two-way solenoid valve

In the hydraulic system, the jam in the slide valve is a common fault. If the spool of the electro-hydraulic proportional directional valve is jammed in the hydraulic pitch control system, then the pitch angle cannot be controlled, which not only makes the maximum power output unavailable but also damages the wind turbine when the wind speed is above the rated. Hence, the redundant digital electro-hydraulic proportional system with high speed switching valve in parallel is shown in Figure 9. When the electro-hydraulic proportional valve fails, the two-way solenoid valve 3.1 and 3.2 can be energised. The flow of the hydraulic oil is controlled by high speed switching valve 2.1 and 2.2. Assuming that port A is connected to the rod chamber and port B is connected to the head chamber, when the wind speed is above the rated wind speed, the switching valve 2.1 is de-energised to allow the oil to flow back to the tank, and the switching valve 2.2 is controlled by regulating the duty cycle of the PWM signal, so the displacement of the cylinder can be controlled by the output flow of port B [9].

![Figure 10. Hydraulic system of a wind turbine](image)
Figure 10 gives a photo which includes the hydraulic collective pitch control system, yaw brake system, main shaft brake system [10].

3.3. Hydraulic individual pitch control technology

Individual pitch control is another widely used method for blade pitch control. In individual pitch system, each blade has its own drive mechanism respectively. When one of the drive mechanisms fails, the others can still accomplish the pitch control to ensure the safety of the wind turbine [11]. In addition, for the large scale wind turbine, the unbalanced aerodynamic loads caused by wind shear effect and tower shadow effect are heavier. Reducing the blade vibration by individual pitch control is becoming a wind power research hotspot [12].

The hydraulic individual pitch control system is divided into two parts; one is in the hub and the other is in the nacelle (Figure 11 and Figure 12).

![Figure 11. Schematic of hydraulic individual pitch control system](image1)

![Figure 12. Structure of hydraulic individual pitch control system](image2)

In the hub, each blade is driven individually by a hydraulic cylinder, and each cylinder is controlled by an electro-hydraulic proportional valve (Figure 8). To improve the frequency response of the cylinders, the electro-hydraulic proportional valves are set up in the hub and close to the cylinders.
4. HYDRAULIC POWER TRAIN IN WIND TURBINES

Nowadays, there are mainly two types of transmission in wind turbines: gearbox transmission and direct-drive transmission. Though both of them are widely used, they have disadvantages. The gearbox transmission has a high failure rate, and for the direct-drive transmission, its generator is bulky, heavy and expensive. Thereby, some new transmission methods are rising these years, in which the hydraulic transmission attracts much attention for its feature of smoothing the power fluctuation and the access to “Variable Speed Constant Frequency (VSCF)” technology [13, 14].

4.1. Hydraulic transmission technology

The schematic and structure of hydraulic transmission in wind turbines is shown in Figure 13. The hydraulic pump rotates with the wind rotor, and pumps the oil to the hydraulic motor. Driven by the motor, the generator generates electricity. The whole process is controlled by the control cabinet in the nacelle.

![Figure 13. Schematic and structure of hydraulic transmission in wind turbines](image)

In 2007, ChapDrive Norway developed a 900kW hydraulic transmission wind turbine [15]. In 2009, Artemis UK developed a 1.5MW hydraulic transmission wind turbine [16]. Research teams from Gheorghe Asachi Technical University of Iași in Romania [17] and the University of Minnesota [18] are also engaged in related researches. In China, there are some institutions or corporations working on the hydraulic power train system in wind turbine, for example, Spark Energy Co., Ltd. Dalian imported the wind turbine hydraulic transmission technology from ChapDrive Norway in 2011.

![Figure 14. Axial piston pump in Artemis 1.5 MW wind turbine](image)

The design of the hydraulic pump is the key technology of the hydraulic transmission system in wind turbines, compared with the conventional pump-motor system. The hydraulic pump in wind turbine is directly connected to the wind rotor, and the rated speed is only about 15 rpm. In order to meet the power requirement, the pump needs to be equipped with the low-speed, large-displacement characteristics, which cannot be offered by...
conventional hydraulic pump. ChapDrive Norway took the Rexroth low speed high torque motor as the pump and Artemis UK developed a large axial piston pump (Figure 14). As the diameter of the pump and the number of piston cylinders increase, the displacement of the pump will increase while the output power fluctuation will decrease.

4.2. Hydro-mechanical hybrid transmission technology

Though Hydraulic transmission can reduce the torque fluctuation of the powertrain system to some extent, the overall efficiency is always not satisfying. Moreover, the low-speed large-displacement pumps need specific design and its efficiency in low speed cannot be guaranteed. Thus it is of great significance to combine the merits of mechanical transmission and hydraulic transmission.

Based on the researches of hydro-mechanical hybrid transmission in engineering machinery [19] and the characteristics of wind turbines, we designed a hydro-mechanical parallel transmission in wind turbines. In this system, the mechanical transmission will deliver the power and the hydraulic transmission is used to smooth the energy fluctuations. Figure 15 shows the schematic of hydro-mechanical hybrid transmission in wind turbines. The gears 2, 3, 4, 5 and ring gear 13 (fixed to the gear box) constitute a two stage NGW planetary gear. The wind rotor 1 is connected to the gearbox input shaft, while the output shaft is connected to gear 5’. The planet carrier 6 driven by gear 5’ will cause the rotation of gear 8 and 8’, from where the power is splitted into two branches. One branch of power is transmitted to the output shaft through gear 8, 7, 12, 11, and the other branch of power is transmitted to the output shaft as well through gear 8’, 9, hydraulic pump19, hydraulic motor 23, gear 10 and 11. They merge together and drive the permanent magnet synchronous motor 14 to rotate and generate electricity.

![Figure 15. Schematic of the hydro-mechanical hybrid transmission](image)

According to Figure 15, the control strategy is as follows: when the wind turbine starts, the displacement of the variable hydraulic pump is set to the maximum. Through regulating the displacement of the variable hydraulic motor, the rotational speed of the generator is controlled. When the frequency and phase of the output electricity meet the demands of the power grid, the wind turbine will connect the grid. When the wind speed is lower than the rated value, the displacement of the pump and the motor will both be zero and only the mechanical transmission is in operation under the circumstance of optimal tip speed ratio. When the wind speed increases, the rotor speed increase will lag behind the wind speed due to the inertia. Therefore the tip
speed ratio is not optimal any more. In this case we can regulate the hydraulic motor displacement, and the hydraulic oil in the accumulator will drive the hydraulic motor and the generator. These measures will reduce the counter torque of the wind rotor and the rotor speed will increase rapidly. When the rotor speed increases to the optimal tip speed ratio again, we can turn off the hydraulic motor. When the wind speed decreases, the rotor speed decrease will lag behind the wind speed due to the inertia. In this case we can regulate the hydraulic pump displacement, and the hydraulic oil will be pumped into the accumulator. These measures will increase the counter torque on the wind rotor and the rotor speed will decrease rapidly. When the rotor speed decreases to the optimal tip speed ratio again, the hydraulic pump is turned off. In this system, when the wind speed is higher than the rated value, the power output will be controlled by the pitch control, and the output power stability will be achieved in the power train by adjusting the pump displacement and the relief valve.

5. CONCLUSION

The hydraulic transmission system is less used in modern large wind machines, but it plays a crucial role in the brakes and pitch control system. With the wind technologies developed, some new technologies are being researched and tested in wind turbine, such as hydraulic soft brake and individual pitch control, which will greatly improve the performance of wind turbines. Especially in recent years, the hydraulic power transmission has been employed in wind turbine instead of the gear transmission, which is becoming a research hotspot worldwide. At the same time, the hydro-mechanical hybrid transmission that occupies the advantages of hydraulic transmission and gear transmission, will be a novel and prospective transmission method in wind power transmission.

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REFERENCES


RESEARCH ON THE STIFFNESS OF THE HYDRAULIC TRANSFORMER CONTROLLED SYSTEM

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ABSTRACT

In the hydraulic transmission control system, there are mainly two kinds of control systems: valve controlled system and pump controlled system. Valve controlled system has large throttling loss and low efficiency. Pump controlled system decreases the energy loss a lot and improves the system efficiency. Besides these two sorts of control systems, as the development of hydraulic common pressure rail, hydraulic transformer controlled system has been developed. Hydraulic transformer can realize system control without throttling loss theoretically.

The paper proposes the mechanical structure of the hydraulic controlled system, which consists of the hydraulic common pressure rail, hydraulic transformer and actuators. Both the hydraulic transformer controlled motor system and the hydraulic transformer controlled cylinder system were investigated simultaneously. Both system models were also built respectively. To realize better system control, the system stiffness equations were developed. The stiffness of pump controlled system was also discussed. An experimental apparatus for hydraulic controlled system was designed and constructed.

The results indicate that the hydraulic transformer cylinder speed is determined by both the oil pressure and the loads. The stiffness of the hydraulic transformer controlled system is smaller than the stiffness of pump controlled system. And the system stiffness changes with the variation of the hydraulic transformer controlled angle. It becomes large with the increase of the hydraulic transformer controlled angle.

KEYWORDS: hydraulic transformer controlled system, system stiffness, pump controlled system, hydraulic transformer controlled angle

1. INTRODUCTION

In the hydraulic control system, valve controlled system and pump controlled system have been developed very well [1, 2]. In the 1980s, German researchers Nikalaus, Kordak and Backe presented the common pressure rail technology which promoted the development of hydraulic transformer (HT) and hydraulic transformer controlled system [3, 4]. Compared with the conventional valve controlled system and pump controlled system, hydraulic transformer controlled system can realize the continuous adjustment of the system pressure without throttle loss. And the hydraulic transformer controlled system divided the energy source and the load which is beneficial to control the subsystems individually.

The structure of the hydraulic transformer developed from conventional structure which is connecting hydraulic pump with hydraulic motor coaxial to the structure with three ports on the valve plate [6, 7]. The principle of speed control of the hydraulic transformer controlled system is as same as the pump controlled
system. Both are volumetric speed control [8]. In the real working conditions, system load varies frequently.

In order to develop and apply the hydraulic transformer well, and realize better system control, it is essential to study the basic characteristics of the hydraulic transformer controlled system. The research of hydraulic transformer displacement characteristics indicates that the port displacement is just related with the port controlled angle and has no relationship with the load and the speed [9]. The torque characteristics show that the reason of vibration is the discontinuous variation of the port torque [10]. The efficiency characteristics research discussed the impact of the controlled angle, pressure and speed to the system efficiency [11]. The flow characteristics analysed the uniformity and continuity of the flow and explained the reasons of noise which provided theory instruction for the design of valve plate.

In order to improve the system robustness, this paper studied the system stiffness characteristics. At first, the system stiffness expression was built in terms of the basic principle of the hydraulic transformer controlled system. The system stiffness characteristics and the comparison between hydraulic transformer controlled system and pump controlled system were also discussed.

2. HYDRAULIC TRANSFORMER CONTROLLED SYSTEM

The hydraulic controlled system consists of the hydraulic common pressure rail, hydraulic transformer and actuators. The hydraulic transformer can drive both linear load and rotary load [13]. As is shown in the Fig. 1, the hydraulic transformer contains three ports: port A connects with the high pressure rail, port B connects with the load and port T connected with the low pressure rail. The adjustment of hydraulic transformer ratio can be realised by adjusting the angle of swash plate. The angle variation of swash plate can cause the variation of the port displacement [14]. Hydraulic transformer can adjust the system pressure in both directions which enable the hydraulic transformer to output energy to the load and recovery energy from the load [15]. This function enables the system to decrease the energy loss and improve the system efficiency. For getting the system stiffness and comparing with pump controlled system under the equal conditions, the open loop control is adopted.
3. SYSTEM STIFFNESS

3.1. Hydraulic Transformer Controlled System

For the hydraulic transformer controlled motor system, the continuous flow equation of high-pressure rail is:

$$\frac{V_0}{\beta} \frac{dp_b}{dt} = V_b n_{HT} - V_m n_m - C_i (p_b - p_t) - C_e p_b \tag{1}$$

Where $V_0$ is the controlled volume. $\beta$ is the oil bulk modulus. $p_b$ is the pressure of the port B. $p_t$ is the pressure of the low-pressure rail. $V_b$ is the hydraulic transformer port B displacement. $n_{HT}$ is the cylinder speed of the hydraulic transformer. $V_m$ is the hydraulic motor displacement. $n_m$ is the hydraulic motor speed. $C_i$ is the equivalent internal leakage coefficient. $C_e$ is the equivalent external leakage coefficient.

The torque balance equation of the hydraulic transformer is:

$$J_{HT} \frac{d}{dt} n_{HT} = p_a V_a - p_b V_b + p_t V_t - B_{HT} n_{HT} \tag{2}$$

Where $J_{HT}$ is the hydraulic transformer cylinder inertia. $B_{HT}$ is the viscous damping coefficient of the hydraulic transformer. $V_a$ is the hydraulic transformer port A displacement. $V_t$ is the hydraulic transformer port T displacement.

The torque balance equation of the hydraulic motor is:

$$J_m \frac{d}{dt} n_m = (p_b - p_t) V_m - B_m n_m - T_L \tag{3}$$

Where $J_m$ is the hydraulic motor inertia. $T_L$ is the load torque. $B_m$ is the viscous damping coefficient of the hydraulic motor.

From the Eqs. (1), (2) and (3), the transfer function of open loop system between $n_m(s)$ and $T_L(s)$ can be obtained:

$$n_m(s) = -\frac{(C_2 s^2 + C_0 s + C_0)}{A_3 s^3 + A_2 s^2 + A_1 s + A_0} T_L(s) \tag{4}$$

Where

$$\begin{align*}
C_2 &= J_{HT} V_0 \\
C_i &= J_{HT} \beta C_e + B_{HT} V_0 \\
C_0 &= \beta \left( K_1 + B_{HT} C_i \right) \\
A_3 &= J_{HT} J_m V_0 \\
A_2 &= B_{HT} J_m V_0 + J_{HT} \beta C_i J_m + V_0 B_m J_{HT} \\
A_1 &= J_{HT} \beta V_m^2 + J_m \beta K_1^2 + C_i \beta \left( B_m J_{HT} + B_{HT} J_m \right) + V_0 B_{HT} B_m \\
A_0 &= \beta \left( B_m K_1^2 + B_{HT} V_m^2 + C_i B_{HT} B_m \right) \\
K_1 &= V_{b0} = K_{b1} + K_{b0} \phi \_1 \\
C_i &= C_i + C_e
\end{align*}$$
\(K_{b1}\) and \(K_{b2}\) are displacement coefficient. \(C_t\) is the total leakage coefficient. \(V_{b0}\) is the initial displacement of hydraulic transformer port B, \(\phi_i\) is the initial controlled angle of the hydraulic transformer.

From the Eq. (4), the system stiffness of hydraulic transformer controlled motor system can be obtained:

\[
\left| \frac{T_k(s)}{n_m(s)} \right|_{HT} = \frac{B_m V_{b0}^2 + B_{HT} V_m^2}{V_{b0}^2 + B_{HT} C_t} \tag{5}
\]

For the pump controlled motor system [16], there are:

\[
\begin{aligned}
&\frac{V_m \, dp_s}{\beta \, dt} = Q_p - n_m V_m - C_p p_s \\
&Q_p = K_p r_p n_p - C_p p_s \\
p_s V_m = J_m \frac{d}{dt} n_m + B_m n_m + T_L
\end{aligned}
\]

Where \(Q_p\) is the pump flow, \(p_s\) is the load pressure, \(K_p\) is the gain of pump flow, \(r_p\) is the swash plate angle of the pump, \(n_p\) is the pump speed.

From the Eq. (6), the transfer function between \(n_m(s)\) and \(T_k(s)\) is obtained:

\[
n_m(s) = - \frac{V_p s + C_p \beta}{J_m V_0 s^2 + (B_m V_0 + C_m J_m \beta)s + V_m^2 \beta} T_k(s) \tag{7}
\]

Then the system stiffness of the pump controlled motor system can be obtained.

\[
\left| \frac{T_k(s)}{n_m(s)} \right|_p = \frac{V_m^2}{C_t} \tag{8}
\]

3.2. Hydraulic Transformer Controlled Hydraulic Cylinder System

For the hydraulic transformer controlled hydraulic cylinder system, there are:

\[
\begin{aligned}
&\frac{V_0 \, dp_b}{\beta \, dt} = V_s n_{HT} - v A_p - C_b p_b \\
&J_{HT} \frac{d}{dt} n_{HT} = p_a V_a - p_b V_b + p_i V_i - B_{HT} n_{HT} \\
m \frac{dv}{dt} = p_b A_p - F
\end{aligned}
\]

Where \(m\) is the total mass of hydraulic cylinder piston and the load, \(v\) is the speed of hydraulic cylinder piston, \(A_p\) is the area of no-rod end of hydraulic cylinder piston, \(F\) is the load resistance.

From the Eq. (9), the system stiffness of hydraulic transformer controlled hydraulic cylinder system can be obtained:

\[
\left| \frac{F(s)}{v(s)} \right|_{HT} = \frac{A_p^2 B_{HT}}{V_{b0}^2 + B_{HT} C_p} \tag{10}
\]
For the pump controlled hydraulic cylinder system [17], there are:

\[
\begin{align*}
\frac{V_p}{\beta} \frac{dp_p}{dt} &= Q_p - vA_p - C_p p_p \\
Q_p &= K_p p_p n_p - C_i p_s \\
m \frac{dv}{dt} &= p_s A_p - F
\end{align*}
\]

[11]

From the Eq. (11), the transfer function between \( v(s) \) and \( F(s) \) is obtained:

\[
v(s) = -\frac{V_0^s + C_i \beta}{V_0 m s^2 + C_i \beta m s + \beta A_p^2} F(s)
\]

[12]

Then the system stiffness of the pump controlled hydraulic cylinder system is obtained:

\[
\left| \frac{F(s)}{v(s)} \right|_p = \frac{A_p^2}{C_i}
\]

[13]

4. SYSTEM STIFFNESS CHARACTERISTIC

For the hydraulic controlled motor system, it can be observed from Eq. (5) that the impact factors of the system stiffness includes the displacement of hydraulic transformer port B, hydraulic motor displacement, viscous resistance coefficient of the hydraulic components and the leakage coefficient.

As is shown in the Eq. (5), the stiffness of the hydraulic transformer controlled system become large as the decrease of the port B displacement. The common range of hydraulic transformer controlled angle is 30° ~120°. As is shown in the Fig. 2, the port B displacement decreases as increasing the hydraulic transformer controlled angle in the common range. Therefore, the system stiffness can be improved by increasing the hydraulic transformer controlled angle. Increasing the hydraulic transformer controlled angle is increasing the transformer ratio of the hydraulic transformer [14]. The larger transformer ratio means that the hydraulic transformer controlled system has larger capacity to adjust the system pressure. It represents that larger pressure can be built between the port B and the load. And the hydraulic transformer controlled system can resist larger load disturbances. As is shown in the Fig. 3, the decline percentage of the motor speed decreases as increasing the hydraulic transformer controlled angle after applying the same load disturbance. This indicates that larger hydraulic transformer controlled angle can increase the system stiffness and resist the disturbances efficiently.

![Figure 2. The displacement variation of hydraulic transformer ports](image)
The Eq. (14) is the system stiffness compare result between the hydraulic transformer controlled motor system and the pump controlled motor system. As is shown in the result, the stiffness of hydraulic transformer controlled motor system is smaller than the stiffness of pump controlled motor system.

\[
\frac{T_L(s)}{n_m(s)}_{HT} - \frac{T_L(s)}{n_m(s)}_p = \frac{B_m V_b^2 + B_{HT} V_m^2}{V_b^2 + B_{HT} C_1} - \frac{V_m^2}{C_1} = \frac{V_b^2 (B_m C_p - V_m^2)}{C_1 (V_b^2 + B_{HT} C_1)} \leq 0
\]  

[14]

The Fig. 4 is the simulation results of the hydraulic transformer controlled motor system and the pump controlled motor system. Applying the same load of 100 N·m, the speed of hydraulic transformer controlled motor system decreased by 13%. But the speed of pump controlled motor system just decreased slightly. It indicates that the stiffness of the hydraulic transformer controlled motor system is small relatively.

The hydraulic transformer is different from the pump. Its speed is determined by the pressure and displacement of the three ports. The variation of pressure or displacement will cause the change of the torque balance of the interior of hydraulic transformer. The alteration of the torque balance will cause the speed alteration of the hydraulic transformer. The Fig. 5 is the test rig of the hydraulic transformer controlled motor system. The Fig. 6 is the test results of the hydraulic transformer controlled motor system. It is shown that the pressure of port B increased after applying the load of 50 N·m. From the hydraulic transformer torque balance Eq. (2), it can be seen that the increase of port B pressure causes the decrease of the hydraulic transformer speed. And then, the decrease of the hydraulic transformer speed causes the decrease of the flow of the hydraulic transformer port B. Because the hydraulic motor speed is determined by the port B flow and the motor displacement collectively. So when the hydraulic motor displacement keep constant, the decrease of the port B flow must cause the decrease of the hydraulic motor speed. From the test results, it can be seen that the hydraulic motor speed decreased by 34%.

Figure 3. The decline percentage curve of the hydraulic motor speed

Figure 4. The simulation results of the hydraulic transformer controlled motor system
For the pump controlled motor system, the pump is driven by the electric motor. So the pump speed doesn't decrease too much after the increase of the load. Because of the influence of the oil volume variation, the pump speed just decreases a little.

For the hydraulic transformer controlled hydraulic cylinder system, it can be seen from the Eq. (7) that the stiffness impact factors are hydraulic transformer B port displacement, the area of no-rod end of the hydraulic cylinder piston, the leakage coefficient and the viscous resistance coefficient. Improving the area of no-rod end of the hydraulic cylinder piston and hydraulic transformer viscous resistance coefficient, reducing the hydraulic transformer port B displacement and the total leakage factor, both are beneficial to improve the stiffness of the hydraulic transformer controlled hydraulic cylinder system.

The Eq. (15) is the stiffness comparison result of the hydraulic transformer controlled hydraulic cylinder system and the pump controlled hydraulic cylinder system. It can be seen that the stiffness of the hydraulic transformer controlled hydraulic cylinder system is smaller than the stiffness of the pump controlled hydraulic cylinder system. The comparison result is consistent with the qualitative analysis of reference [6].

\[
\left| \frac{F(s)}{v(s)} \right|_{HT} - \left| \frac{F(s)}{v(s)} \right|_{p} = \frac{A_p B_{HT}}{V_{b0} + B_{HT} C_t} - \frac{A_p^2}{C_t V_{b0}^2 + B_{HT} C_t} < 0 \tag{15}
\]

After the above analysis, it can be seen that the stiffness of hydraulic transformer controlled system is relatively small. In the real work conditions, the system speed is easy to be affected by the exterior disturbance which is a disadvantage for improving the system robustness. Therefore, the hydraulic transformer controlled system is not applicable for the occasions that require high precision of the speed control.
5. CONCLUSION

(1) The stiffness expressions of both the hydraulic transformer controlled motor system and the hydraulic transformer controlled hydraulic cylinder system were derived. The results showed that increase the motor displacement or the area of no-rod end of the piston, reduce the displacement of the hydraulic transformer port B appropriately and improve the system sealing performance are all beneficial to improve the system stiffness.

(2) Increasing the hydraulic transformer controlled angle can achieve these following points: improve the transformer ratio of hydraulic transformer, expand the adjustable range of pressure in the hydraulic transformer control system, resist the load changes and improve system stiffness apparently.

(3) The research proves that the stiffness of the hydraulic transformer control led system is smaller than the pump controlled system. And the hydraulic transformer controlled system is not applicable for the occasions that require high precision of the speed control.

REFERENCES


NOMENCLATURE

\[ A_p \] the area of no-rod end of hydraulic cylinder piston, m\(^2\).

\[ B_{HT} \] the viscous damping coefficient of the hydraulic transformer, N·m·s·rad\(^{-1}\).

\[ B_m \] the viscous damping coefficient of the hydraulic motor, N·m·s·rad\(^{-1}\).

\[ C_i \] the equivalent internal leakage coefficient, m\(^5\)·(N·s)\(^{-1}\).

\[ C_c \] the equivalent external leakage coefficient, m\(^5\)·(N·s)\(^{-1}\).

\[ C_t \] the total leakage coefficient, m\(^5\)·(N·s)\(^{-1}\).

\[ F \] the load resistance, N.

\[ J_{HT} \] the hydraulic transformer cylinder inertia, kg·m\(^2\).

\[ J_m \] the hydraulic motor inertia, kg·m\(^2\).

\[ K_{b1} \] the displacement coefficient

\[ K_{b2} \] the displacement coefficient

\[ K_p \] the gain of pump flow, m\(^3\)·s\(^{-1}\).

\[ m \] the total mass of hydraulic cylinder piston and the load, kg.

\[ n_{HT} \] the cylinder speed of the hydraulic transformer, rad·s\(^{-1}\).

\[ n_m \] the hydraulic motor speed, rad·s\(^{-1}\).

\[ p_b \] the pressure of the port B, MPa.
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p_i$</td>
<td>the pressure of the low-pressure rail, MPa.</td>
</tr>
<tr>
<td>$p_s$</td>
<td>the load pressure, MPa.</td>
</tr>
<tr>
<td>$Q_p$</td>
<td>the pump flow, m$^3$.s$^{-1}$.</td>
</tr>
<tr>
<td>$r_p$</td>
<td>the swash plate angle of the pump, rad.</td>
</tr>
<tr>
<td>$T_L$</td>
<td>the load torque, N·m.</td>
</tr>
<tr>
<td>$V_0$</td>
<td>the controlled volume, m$^3$.</td>
</tr>
<tr>
<td>$V_a$</td>
<td>the displacement of the port A, m$^3$.rad$^{-1}$.</td>
</tr>
<tr>
<td>$V_b$</td>
<td>the displacement of the port B, m$^3$.rad$^{-1}$.</td>
</tr>
<tr>
<td>$V_t$</td>
<td>the displacement of the port T, m$^3$.rad$^{-1}$.</td>
</tr>
<tr>
<td>$V_m$</td>
<td>the hydraulic motor displacement, m$^3$.rad$^{-1}$.</td>
</tr>
<tr>
<td>$V_{b0}$</td>
<td>the initial displacement of hydraulic transformer port B, m$^3$.rad$^{-1}$.</td>
</tr>
<tr>
<td>$\beta$</td>
<td>the oil bulk modulus, N·m$^{-2}$.</td>
</tr>
<tr>
<td>$\phi_i$</td>
<td>the initial controlled angle of the hydraulic transformer, rad.</td>
</tr>
<tr>
<td>$v$</td>
<td>the speed of hydraulic cylinder piston, m·s$^{-1}$.</td>
</tr>
</tbody>
</table>
DYNAMICS OF VOLUME CONTROLLED MECHANICAL VENTILATION SYSTEM

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ABSTRACT

Dynamics of mechanical ventilation system can be referred in pulmonary diagnostics and treatments. In this paper, to illustrate the influences of key parameters on the dynamics of the volume controlled ventilation system, firstly, a new mathematical model of the ventilation system is proposed. Secondly, the mathematical model is verified through comparison of simulation results and experimental results. At last, the influences of key parameters on the dynamics of the volume controlled ventilation system are carried out. This study can be helpful in respiratory treatment, diagnostics and design of various medical devices or diagnostic systems.

KEYWORDS: Mechanical ventilation, Volume controlled ventilation, Pneumatic system, Dynamics, Modeling simulation

1. INTRODUCTION

Mechanical ventilation is a vital treatment, which is usually adopted to ventilate patients who cannot breathe adequately on their own [1-3]. As an alternative mode of ventilation, Volume controlled ventilation (VCV) is widely used in respiratory failure [4-6]. Nowadays, it was accepted that VCV can ensure enough insufflation, and be beneficial to respiratory muscle rest. However, VCV ventilation has its shortcomings, for example it may cause hypertonia of respiratory system, and that may in turn injure the alveolar and respiratory tract [7].

In order to improve the safety of the VCV, the dynamics of the air in the patient’s respiratory system should be determined. But, the pressure in patient’s respiratory system cannot be measured directly and precisely. Therefore, simulation of the ventilation system (including a VCV ventilator and a patient’s respiratory system) is needed.

In present modelling and simulation studies of mechanical ventilation system, the system is commonly considered as an electrical system [8-11]. However, the versatility and applicability of these models are limited [12]. Furthermore, these models contain many parameters, and they are ill-suited to analyze the dynamic characteristics of a mechanical ventilation system.

To provide a novel method to study the dynamics of the VCV system, in this study, firstly, the VCV system is considered as equivalent to a pneumatic system, which provides greater versatility and applicability. Then a new mathematical model of the VCV system is derived.
Furthermore, through the simulation study on the VCV system, its dynamic characteristics are obtained. To verify the mathematical model, the simulation results are compared with experimental results in a reference. Finally, in order to improve the safety of VCV and provide guidance for VCV treatment, influences of key parameters on the dynamics of the VCV system are studied.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Variable</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Equivalent effective area</td>
<td>([m^2])</td>
</tr>
<tr>
<td>C</td>
<td>Respiratory compliance</td>
<td>([L/cmH_2O])</td>
</tr>
<tr>
<td>m</td>
<td>Mass</td>
<td>([kg])</td>
</tr>
<tr>
<td>p</td>
<td>Pressure</td>
<td>([pa])</td>
</tr>
<tr>
<td>q</td>
<td>Air mass flow</td>
<td>([kg/s])</td>
</tr>
<tr>
<td>Q</td>
<td>Air volume flow</td>
<td>([m^3/s])</td>
</tr>
<tr>
<td>R</td>
<td>Gas constant</td>
<td>(=287 ,[J/(kg \cdot K)])</td>
</tr>
<tr>
<td>t</td>
<td>Time</td>
<td>([s])</td>
</tr>
<tr>
<td>V</td>
<td>Volume</td>
<td>([m^3])</td>
</tr>
<tr>
<td>ρ</td>
<td>Density</td>
<td>([kg/m^3])</td>
</tr>
<tr>
<td>κ</td>
<td>Specific heat ratio</td>
<td>(=1.4)</td>
</tr>
<tr>
<td>θ</td>
<td>Temperature</td>
<td>([K])</td>
</tr>
</tbody>
</table>

Subscripts:

- a: The standard reference atmosphere state
- et: Endotracheal tube
- ev: Exhalation valve
- frc: Functional residual capacity
- i: Inspiration/input
- k: peak/maximum
- l: Lung
- o: Output
- p: Platform
- T: Tidal
- t: Tube
- v: Ventilator

2. METHODS OF STUDY ON THE VCV SYSTEM
2.1. Introduction of VCV System

A VCV system comprised of a human lung, a respiratory tract, a flexible tube, a endotracheal tube and a VCV ventilator. In inspiration process, positive pressure ventilation (produced by the ventilator) is utilized to force airflow into the human lung. In expiration process, due to the elasticity of the lung, air is exhausted to the atmosphere, through an exhalation valve, embedded in the ventilator. Therefore, according to their functions, the ventilator can be regarded as an air compressor, the exhalation valve and endotracheal tube can be considered as two throttles, and the human lung can be regarded as a variable volume container.

Hence, as shown in Figure.1, the VCV system can be considered as a pneumatic system. The compressor the throttle 1, 2, and the container represent the ventilator, the exhalation valve, the endotracheal tube and the human lung, respectively. The effective area of the endotracheal tube and exhalation valve are represented by \( A_{et}, A_{ev} \) respectively.

During the inspiration, the compressor and the flexible tube are connected with the solenoid valve. Then the outputted compressed air from the compressor is charged to the variable container through the flexible tube, the solenoid valve and throttle 2. During the expiration, the throttle 1 and the flexible tube are connected with the solenoid valve. Then the outputted compressed air from the variable volume container is expelled to the atmosphere through the solenoid valve, throttle 2, 1 and flexible tube.

![Figure 1. Structures of the equivalent pneumatic system](image)

2.2. Modeling of Mechanical Ventilation System

According to the working principle of the mechanical ventilation system, the following assumptions are made:

1. Air in the system follows all ideal gas laws;
2. The temperature, pressure and density field of air in the same capacity is uniform. At any time, the state parameter of air in the overall system is the same, the dynamic process is a quasi balanced process;
3. There is no air leakage during the working process;
4. The flow of air into and out of the lung simulator is stable one-dimensional flow, equivalent to the flow of air through the nozzle contraction.

2.2.1. Flow equations

1. Output flow of the VCV ventilator

According to the working principle of the VCV, the output flow of the VCV ventilator can be given as:
2. Air flow through the exhalation valve and the endotracheal tube

When air flows through the exhalation valve and the endotracheal tube, the mass flow can be calculated by the equation when air flows through the Laval nozzle (a tube that is pinched in the middle, making a carefully balanced, asymmetric hourglass-shape). As the pressure of the studied ventilation system is about 0 ~ 40cmH₂O, therefore, the ratio of the downstream pressure and the upstream pressure is always larger than 0.528. The mass flow equations of mechanical ventilation system can be obtained, according to ISO6358.

\[
Q_{vo} = \begin{cases} Q_T & V_{vo} < V_T \\ 0 & V_{vo} \geq V_T \end{cases}
\]

(1)

\[
Q_t = \frac{V_T}{T_t - T_p}
\]

(2)

So, the air flow through the exhalation valve and the endotracheal tube can be given by

\[
q_{ev} = \frac{A_{ev} p_T}{\sqrt{\theta}} \sqrt{\frac{2\kappa - 1}{\kappa - 1} R \left[ \left( \frac{p_a}{p_i} \right)^{\frac{2}{K}} - \left( \frac{p_a}{p_i} \right)^{\frac{K+1}{K}} \right]}
\]

(3)

\[
q_{et} = \begin{cases} \frac{A_{et} p_T}{\sqrt{\theta}} \sqrt{\frac{2\kappa - 1}{\kappa - 1} R \left[ \left( \frac{p_i}{p_t} \right)^{\frac{2}{K}} - \left( \frac{p_i}{p_t} \right)^{\frac{K+1}{K}} \right]} & p_t > p_i \\ \frac{A_{et} p_t}{\sqrt{\theta}} \sqrt{\frac{2\kappa - 1}{\kappa - 1} R \left[ \left( \frac{p_i}{p_t} \right)^{\frac{2}{K}} - \left( \frac{p_i}{p_t} \right)^{\frac{K+1}{K}} \right]} & p_i \leq p_t \end{cases}
\]

(4)

So, the air flow through the exhalation valve and the endotracheal tube can be given by

\[
Q_{ev} = \frac{A_{ev} p_t}{p_a \sqrt{\theta}} \sqrt{\frac{2\kappa - 1}{\kappa - 1} R \left[ \left( \frac{p_a}{p_i} \right)^{\frac{2}{K}} - \left( \frac{p_a}{p_i} \right)^{\frac{K+1}{K}} \right]}
\]

(5)

\[
Q_{et} = \begin{cases} \frac{A_{et} p_t}{p_a \sqrt{\theta}} \sqrt{\frac{2\kappa - 1}{\kappa - 1} R \left[ \left( \frac{p_i}{p_t} \right)^{\frac{2}{K}} - \left( \frac{p_i}{p_t} \right)^{\frac{K+1}{K}} \right]} & p_t > p_i \\ \frac{A_{et} p_t}{p_a \sqrt{\theta}} \sqrt{\frac{2\kappa - 1}{\kappa - 1} R \left[ \left( \frac{p_i}{p_t} \right)^{\frac{2}{K}} - \left( \frac{p_i}{p_t} \right)^{\frac{K+1}{K}} \right]} & p_i \leq p_t \end{cases}
\]

(6)

2.2.2. Volume Equation

According to the definition of compliance C, the compliance C of the lung can be described as \[ C_i = \frac{dV_i}{dp_i} \]
Then, the volume of the lung can be calculated by the following formula:

\[ dV_l = C_l dp_l \] (8)

2.2.3. Pressure Equation

The VCV system can be considered as an open thermodynamic system, and its work can be regarded as an isothermal process. The differential expression of Clapeyron equation \((PV=mR\theta)\) can be given as

\[ \frac{dp}{dt} = \frac{1}{V} R\theta q - \frac{mR\theta dV}{V^2} \frac{dt}{dt} \] (9)

Combine Eqs. (8) and (9), then the pressure in the flexible tube and the ventilated lung can be given by

\[ \frac{dp_t}{dt} = \frac{R\theta q V_t}{V_t^2 + CmR\theta} \] (10)

\[ \frac{dp_l}{dt} = \frac{R\theta q V_l}{V_l^2 + CmR\theta} \] (11)

3. STUDY ON THE DYNAMICS OF THE VCV SYSTEM

3.1. Experimental verification of the mathematical model

In the reference [7], an experimental study on a VCV system was carried out, Ti, Tp, Te, and VT are set to 1.6s, 0.4s, 2.7s and 500ml. The compliances of the flexible tube and the ventilated lung are 4ml/cmH2O and 25.7ml/cmH2O. The effective areas of the exhalation valve and endotracheal tube are 16mm2 and 9 mm2. The experimental VCV system described in the reference [7] was simulated with the mathematical model, which is coded in an S-function of Matlab/simulink. The experimental results and the simulation results are shown in Fig.2 and 3.

The curves of the pressure \((pt)\) in the flexible tube, the pressure \((pl)\) in the lung, the volume \((Vt)\) of the air which flows through the inlet of the flexible tube, the volume \((Vl)\) of the air which flows into and out from the lung, the air flow \((Qt)\) through the inlet of the flexible tube, and the air flow \((Ql)\) into and out from the lung are shown in Figs. 2 and 3.
As can be seen in Figures 2 and 3, it is obvious that

1. The simulation results are consistent with the experimental results, verifying the mathematical model. Therefore, the mathematical model can be used to study the VCV system.

2. The air pressure in the lung always lags behind the pressure in the flexible tube, and that is why the pressure in the lung cannot be maintained precisely. The main reason for this is that the respiratory resistance and compliance block the fluctuation of the pressure in the lung simulator.

3. The pressure (\(p_t\)) increases with an increase in the output pressure of the ventilator. But, after the peak pressure, the pressure (\(p_t\)) drops to its platform pressure, while the air pressure in the lung simulator continue to rise, until it is equal to the platform pressure of the ventilator. After that, the air pressure in the lung simulator decreases with a drop in the pressure (\(p_t\)).

4. In the inspiration process, with an increase in the pressure (\(p_t\)), the input air flow of lung simulator increases sharply. After the peak pressure, the pressure (\(p_t\)) drops to its platform pressure, the input air flow of lung simulator keeps a constant. And finally the lung simulator stops inspiration when the air pressure in lung simulator is as the same as the pressure (\(p_t\)).

5. In the expiration process, the output air flow of lung simulator rises sharply with a decrease in the pressure (\(p_t\)). When the pressure (\(p_t\)) declines to PEEP, the output air flow of lung simulator starts to decrease. And finally the output air flow towards zero when the pressure (\(p_t\)) forwards PEEP.

### 3.2. Influence of the key parameters on the dynamics

As the flow and pressure of human lung is critical to mechanical ventilation treatment, to guarantee a good treatment, it is necessary to study the influence of the key parameters, namely the effective area of the endotracheal (\(A_{et}\)) and the compliance (\(C_l\)) of the ventilated lung, on the flow and pressure dynamics of the VCV system.

Through simulation, influences of the effective area of the endotracheal (\(A_{et}\)) and the compliance (\(C_l\)) of the ventilated lung on the dynamic are shown in Figure. 4.

As shown in Fig. 4, it can be seen that:
1. The pressure dynamics of the air in the tube and the flow dynamics of the air in the lung are obviously influenced by the effective area (Aet) of the endotracheal. The peak pressure in the lung is slightly influenced by the effective area of the endotracheal.

2. With an increase in the effective area (Aet) of the endotracheal, the peak pressure (ptk) in the tube decreases, non-linearly. When the effective area of the endotracheal (Aet) is smaller than 12.56mm², the maximum flow (Qlk) of the air in the lung increases with an increase in the effective area (Aet) of the endotracheal. When the effective area of the endotracheal (Aet) is bigger than 12.56mm², the maximum flow (Qlk) almost is constant.

3. The pressure and flow dynamics of the air in the tube are slightly influenced by the compliance (Cl) of the lung. But, the pressure dynamics of the lung are significantly influenced by the compliance (Cl) of the lung.

4. When the compliance (Cl) of the lung decreases, the peak pressure (plk) in the lung increases non-linearly.

(a) Influences of the Aet on the dynamics of the tube
(b) Influences of the Aet on the dynamics of the lung
(c) Relationship between the plk, Qlk and the Aet
(d) Influences of the Cl on the dynamics the tube
4. CONCLUSIONS

In this paper, the VCV system was compared to a pneumatic system, and then a new mathematical model of the mechanical ventilation system was proposed. In order to verify the model, an experimental prototype VCV was simulated mathematically. Simulation studies on the dynamics of the VCV system were done, and the conclusions of this study are summarized as follows:

1. The simulation results are consistent with the experimental results, which verify the mathematical model, and the mathematical model can be used to study on the VCV system.
2. The peak pressure in the lung with VCV is slightly influenced by the effective area of the endotracheal.
3. With an increase in the effective area of the endotracheal, the peak pressure in the flexible tube decrease non-linearly.
4. When the effective area of the endotracheal is smaller than a fixed value, with an increase in the effective area of the endotracheal, the maximum flow of the air in the lung increases. When the effective area of the endotracheal is bigger than the fixed value, the maximum flow almost is constant.
5. When the compliance of the ventilated lung decreases, the peak pressure in the lung increases non-linearly.

This study can be of use in the VCV treatment, diagnostics and design of various medical devices or diagnostic systems.

In future studies, the proposed mathematical model will be verified clinically, and the influence of respiratory parameters on the precision of diagnostics will be determined.

ACKNOWLEDGEMENT

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REFERENCES


ABSTRACT

During the last years, hot forming of sheet metal parts has become an alternative production process of high strength components. New material concepts, e.g. boron-manganese steels, permitted the application of hot stamping, especially in the automobile industries. A request of knowledge about the thermo-mechanical material properties of these sheet metals at elevated temperatures and high strain rates demands the enhancement of existing test procedures. Since hot tensile tests can only be evaluated until comparable low strains, a hot gas bulge test at elevated temperatures and high strain rates is being developed.

Standardized bulge tests use hydraulic oil as forming medium. However, the application of hydraulic oil is not possible at hot stamping conditions i.e. with temperatures up to 950 °C. Furthermore, the forming speed has an increasing influence on the material behaviour at increasing temperatures. Therefore, a new design of a so called hot gas bulge test is being developed, which includes a closed loop control of the forming speed. An analysis of the expected material behaviour leads to possible valve configurations. A parallel valve concept is chosen to control the bulge test and examined for the use in a gas based bulge test.

KEYWORDS: Pneumatics, high pressure application, fast-switching valves, material characterisation, parallel valve connection, digital pneumatic

1. INTRODUCTION

In recent years, the implementation of lightweight design using sheet metals manufactured by hot stamping increased significantly. Hot stamping is a forming process with hot metal sheets in the temperature range of 800 - 950 °C. At the beginning of the hot stamping process, the metal sheets are heated up to 950 °C to form an austenitic crystal structure. Afterwards the sheets are formed using cold tools. The thermal shock in the forming die causes the microstructure of the metal sheet to transform from austenitic to martensitic crystal structure, resulting in high strength. With this process, components made of sheet metals with high strength can be realised. Especially the automobile industries apply hot stamping as forming method to save weight in the car’s safety structure and economise production costs [1].

To simulate and evaluate the process of hot stamping, material property data is necessary. The forming process during hot stamping is similar to that of deep drawing. Important material characteristics to describe the deep drawing process are flow curves. With help of the common tensile test, flow curves at controlled strain rates and high temperatures can be achieved by forming the sample until failure and measuring the
tensile force and the strain. With the tensile tests only one-dimensional stresses occur. However, the real forming process reaches a multi-axial state of stress. Caused by the multi-axial state of stress, higher deformations can be achieved than in the tensile test. Therefore, the material properties must be extrapolated from the tensile tests [2].

The error caused by the extrapolation can be reduced with help of the hydraulic bulge test which determines flow curves of sheet metals under biaxial stresses. The bulge test has already been developed for the determination of flow curves at room temperature [3]. Previous approaches to determine flow curves at elevated temperatures didn’t achieve the required temperatures of 950 °C with high strain rates similar to the real hot forming process at the same time. In addition, most of the bulge tests don’t realise any sufficient possibility to control strain or strain rate at the pole of the bulge.

In the following, a further development of the bulge test and its control design will be presented. This new approach will measure flow curves in ranges of the temperature (600 - 950 °C) and of the strain rate (0.1 - 1 s⁻¹), which represent the real forming process more closely.

2. THE BULGETEST

The bulge test is a hydrostatic cupping test. The hydraulic bulge test at room temperature enables the characterization of sheet metals up to high strains. Figure 1. shows the basic bulge test setup.

It consists of a pressure supply, a sheet, a die and a blank holder. Using hydraulic oil as pressure medium, the sheet is bulged through the die, while the die and the blank holder restrain the material flow from the outer perimeter. The forming process is realised contactless neglecting the comparatively small fluid friction.

![Figure 1. Scheme of a bulge test [3]](image)

The bulge test is an indirect measuring test. That means the flow curve is not measured directly but can be derived from the measurement. During the hydraulic bulge test the following measures are typically captured:

- Pressure inside the bulge (\(p\))
- Surface strains (\(\varepsilon\))
- Curvature radius of the bulge (\(\rho\))
- Height of the bulge (\(h\))

With these values and the assumption of an incompressible material behaviour, a post-analysis determines the stress and strain rates during the test to calculate a flow curve. The stress at the bulge pole can be calculated with the assumption of a homogenous and isotropic material behaviour by using the membrane...
theory in equation (1) [4]. This theory explains the relation of pressure \( p \), stress \( \sigma \), curvature radius \( \rho \) and material thickness \( t \) regarding the main stress axis (indices 1, 2). The thickness of the metal sheet can be derived from the original thickness \( t_0 \) and the local strain \( \varphi \) in direction of the main stress axis in equation (2), because of the conservation of volume for incompressible material. In the following, the strain is described as logarithmic ratio of the actual thickness \( t \) to the original thickness \( t_0 \) as in Equation (3). The strain rate is its derivative over time.

\[
\frac{p}{t} = \frac{\sigma_1}{\rho_1} + \frac{\sigma_2}{\rho_2} \\
(1)
\]

\[
t = t_0 \cdot e^{(-\varphi_1 - \varphi_2)} \\
(2)
\]

\[
\varphi = \ln \left( \frac{t}{t_0} \right) \\
(3)
\]

The standard bulge test using hydraulic oil as pressure medium is established for tests at room temperature. For higher temperatures, gas based bulge tests were developed in several projects but without a closed loop strain rate control and high forming temperatures. Furthermore the bulge test is already used at medium temperatures to examine the material behaviour of magnesium- and aluminium alloys during superplastic forming [5, 6].

3. PRELIMINARY CONSIDERATIONS

A new design of a hot gas bulge test that allows the determination of the material properties under the conditions of hot stamping must match the following requirements. During hot stamping, the temperature of the 22MnB5 sheet metal is 950 °C or higher. The strain rates that occur during the forming process are over 1.0 s\(^{-1}\). The resulting strains are very high and the state of stress should be similar in the test bench to the hot stamping.

In this project, the concept of the standard bulge test will be adapted to satisfy these needs. Similar to the operating conditions from hot stamping, the samples will be heated between 600 °C and 950 °C. The pressure medium is nitrogen because the use of oil is difficult at these temperatures. Furthermore, the properties of nitrogen are well known. The pressure supply level will be at 30 MPa. Aim of the project is a controlled strain rate up to 1.0 s\(^{-1}\). With this rate, a test cycle takes less than 1 s. According to Equation (3), a strain of \( \varphi = \varphi \cdot t = 1 \) corresponds to an engineering strain of about \( \varepsilon \approx 170 \% \).

The samples are made of 22MnB5 as sheet metal with a maximum thickness of 2 mm. Within the bulge test, the geometric form of the sample is defined by the clamping die. The inner diameter of the die is 100 mm. The form of the bulge, as seen in Figure 1, can be approximated as a part of a sphere. A strain of \( \varphi = 1 \) implies a bulge height of \( h = 40 \) mm and, therefore, a maximal bulge volume of about 200 ml.

Previously conducted FEM-simulations were used to estimate the needed pressure at the corresponding bulge size, strain and temperature. Yet, since there are no material properties available under the referred conditions, the simulation properties are interpolated from flow curves of compression tests [7, 8].

However, the simulation gives a basis to develop the pneumatic system of the bulge test. With the information of pressure and bulge volume, the pneumatic state variables can be analysed. The first thermodynamic law according to equation (4) can be used to determine the mass change in the system [9]. This law reflects the balance of energy at a change of state (indices 1, 2). Since the cycle is very fast, the assumption of nearly adiabatic changes is applicable. Equation (4) is simplified neglecting flow velocity in the pressure chamber and heat flow during the cycle time. Leaving only the mass flow and the volume change work as the bulge develops, the term results to equation (5)
\[ W + Q + \sum_i \Delta m_i (h_i + e_{ai}) = m_2 (u_2 + e_{a2}) - m_1 (u_1 + e_{a1}) \]  

\[- \left( \int p \, dV \right) + \Delta m \cdot h_{am} + m_1 \cdot u_1 - m_2 \cdot u_2 = 0\]  

The resulting mass flow for 650 °C and a strain rate of 1 s\(^{-1}\) is presented in Figure 2. The pressure rises up to 18 MPa and drops at the end to 15 MPa. The depicted volume change (\(\Delta V/\Delta t\)) is the volume (\(\Delta V\)) which is added to the dead volume \(V_{\text{dead}}\) of the pressure chamber by the bulge during the forming process. The pressure must be controlled precisely to assure a constant strain rate. The required mass flow depends on volume change of the pressure chamber and needed pressure. The simulation starts from an unsteady state caused by an oscillation of the controller in the beginning of the FEM-simulation. This phenomenon influences the first 50 ms. To the end of the cycle, the necessary mass flow becomes negative. At this point, the remaining strength of the sample declines faster than the pressure by the rendered volume work. The forming process becomes overcritical and uncontrollable.

![Figure 2. Mass flow, volume change and pressure over cycle time](image)

The maximum mass flow for the depicted cycle is about 0.3 kg/s at the beginning of the forming process. This mass flow defines the basic design of the pressure control system. As pressure supply, a gas bottle of nitrogen with 25 MPa and a choked flow through the valves is assumed. With its help, the configuration of the equivalent flow cross-section of the valves can be derived for the correlating strain rates. In Table 1., the configuration of the pneumatic system is indicated.

<table>
<thead>
<tr>
<th>strain rate [s(^{-1})]</th>
<th>mass flow [kg/s]</th>
<th>conductance ([10^5 \text{ Nl/s Pa}])</th>
<th>(C_v)-Value [-]</th>
<th>nominal diameter [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.3</td>
<td>0.93</td>
<td>0.26</td>
<td>3.3</td>
</tr>
<tr>
<td>0.1</td>
<td>0.03</td>
<td>0.093</td>
<td>0.026</td>
<td>1.04</td>
</tr>
</tbody>
</table>

4. TEST BENCH

The design of the test bench is shown in Figure 3. It enables the heating and the forming of the sample. The die and the blank holder are mounted between two plates. The pressure chamber with a pressure supply port is integrated in the blank holder. At the side, the electrodes and pneumatic actuators are fixed. The heating is achieved by resistive heating.
This test bench has the dimension to be embedded into a small hydraulic press with a maximum force of 650 kN. The press provides the clamping force to fix the sample. The forming process itself is gas based.

Figure 4. depicts a scheme of the bulge test cycle. The process can be divided into 3 steps. At the beginning, the test chamber is opened. A rectangular sample is placed in the middle of the die. To begin with the heating, the pneumatic cylinders lift the sample and press it with a defined contact area against the electrodes. Thus, the sample is heated within a few seconds by a current of up to 6000 A at 8 V. For the 22MnB5 alloy with a thickness of 1.6 mm, less than 3000 A at 4 V are required to heat the sample up to 1000 °C.

After the sample reaches the necessary temperature, the current is interrupted and the die presses against the blank holder. The force of the die is sufficient to seal the gap with the help of plastic deformation of the sample between the blank holder and the die. At this point, the pressure chamber can be pressurised and the forming process starts until the sample fails.

5. DEVELOPMENT OF THE PARALLEL VALVE CONTROL

To form the bulge, nitrogen is already chosen as pressure medium with its inertia gas properties. However, the principle of the pressure control can vary. It can be realised with a pneumatic, hydraulic, electric or even mechanical transmission system. In previous considerations a pneumatic system was chosen [10]. In the following, the design of the control, the preliminary tests and the resulting optimization are presented.

5.1. Design of the control system

To realise a constant strain rate, the pressure in the bulge volume during forming process (Figure 2.) needs to be controlled. A preferred solution would be a continuous 2/2 or 3/2-way valve which would allow a continuous adjustment of the flow cross section to adjust the mass flow into the bulge volume and therefore
to control the pressure. Because of the high gas pressure, these valves are not yet available on the market and the development of a valve appears for economic reasons not acceptable at the moment. To achieve similar behaviour with available components, a parallel setup of switching valves is considered.

The design of the parallel switching valves allows the adjustment of the effective flow cross-section in steps according to the valve combination. Figure 5. presents such a pneumatic circuit diagram. The valves are used to modulate the mass flow directed into the bulge volume. Both volumes, upstream and downstream of the valves, can be deflated in case of an emergency. Three pressure sensors and a laser sensor to determine the bulge geometry send information to the control unit, which reacts and changes the valve combination. The fast-switching valves have opening and closing times of about 30 ms at a supply pressure of about 25 MPa.

![Figure 5. Circuit diagram of parallel switching valve control](image)

The cross-section changes between the valves. With \( n \) different valves, where \( V_1 \) has the largest cross-section and \( V_n \) the smallest (\( V_1 > V_2 > \ldots > V_n \)), the effective overall cross-section can be realised. The sum of all valve’s diameter determines the highest possible mass flow while the smallest valve limits the accuracy of the control.

The examination of the control system, as depicted in Figure 5., leads to the simplified functional diagram shown in Figure 6. (a). Influences of temperature are neglected in the diagram. Also the switching behaviour of the valves is only roughly estimated. The activation of the valves results in a shortly delayed mass flow into the pressure chamber. The pressure change is also dependent on the volume work of the pressure. The pressure influences the forming of the sheet sample. The growing bulge correlates with a changing volume which is considered in the volume work. The actual bulge pressure is compared with a set pressure and controlled. The controlled pressure difference is used to set a new valve combination.
Figure 6. Functional diagram (a) and scheme of valve control (b)

Figure 6. (b) shows the principle of how the valve combination is activated. The volume in the chamber is calculated with help of the measured bulge height. A mass difference can be derived from the controlled pressure difference. With respect to the opening time of the valves with a down time \( T_{\text{dead}} \) and the duration to change the combination which is integrated in the valve map, a required flow is determined. With the supply pressure and the required flow, the optimal cross-section can be determined and a corresponding valve combination can be set.

5.2. Preliminary tests

To find an optimal valve combination, preliminary tests were performed to examine the properties of the valves.

Four different available valves of three manufacturers were measured to determine the applicability for the bulge test. First, their common properties were recorded with compressed air at a supply pressure of 0.7 MPa abs. The switching time was measured with the help of the electric current \[ 11 \]. The results of the measurements are listed in Table 2.

Table 2. Measurement results of valve properties

<table>
<thead>
<tr>
<th>manufacturer</th>
<th>valve</th>
<th>nominal diameter [mm]</th>
<th>Switching time open [ms]</th>
<th>Sonic conductance [10^5 \text{ Nl/s Pa}]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Müller Co-Ax</td>
<td>KB15</td>
<td>2.0</td>
<td>49</td>
<td>0.57</td>
</tr>
<tr>
<td>GSR</td>
<td>Baureihe 52</td>
<td>1.0</td>
<td>15</td>
<td>0.145</td>
</tr>
<tr>
<td>Era-Sib</td>
<td>EN 52050</td>
<td>0.7</td>
<td>9</td>
<td>0.07</td>
</tr>
<tr>
<td>Era-Sib</td>
<td>EN 52080</td>
<td>1.0</td>
<td>8</td>
<td>0.135</td>
</tr>
</tbody>
</table>
The switching time is a critical aspect of the valves. Considering the long switching time of the first two valves at the given pressure, they are too slow to be further considered. Therefore, the control is conducted with help of the Era-Sib valves which have the fastest switching time.

However, the sonic conductance of the Era-Sib valves does not seem to meet the requirements derived from the FEM-Simulations (Table 1.). To find the optimal configuration of the valve’s equivalent flow cross-section, first tests on the bench with 22MnB5 gave further insight. Figure 7. (a) depicts a bulge test with a sample temperature of about 900 °C until failure. This bulge test was conducted by opening one Era-Sib valve (EN52050) constantly without pressure control.

![Figure 7. Bulge test with heated sheet sample (a) and bulge test with one valve (b) (a)](image)

Another preliminary test to examine the pressure control with only one valve is plotted in Figure 7. (b). This test was also executed with one Era-Sib valve (EN 52050), the smallest of the four tested valves. Nevertheless, the pressure incline resulting by the valve is too large compared to the valve’s dynamic. In fact, the pressure increase inside the test chamber with an activated valve is up to 10 MPa/s.

Both results impose that the real equivalent flow cross-section in comparison with the valve’s dynamics must be smaller to control strain rates. Two reasons are responsible for these results. The real dead volume of the chamber is smaller than assumed at the beginning of the project and the material properties were extrapolated. So, the chosen Era-Sib valves are already large enough considering the flow-through to be applied in the bulge test.

5.3. Optimization of the control system

With the results of the preliminary tests, a valve combination needs to be configured which enables a pressure control as planned. Because the Era-Sib with a nominal diameter of 0.7 mm has already a high flow, smaller cross-sections are required.

Valves, smaller than the given Era-Sib valve, which satisfy the needed properties are not commonly available on the market. To realize a controlled flow at smaller cross-sections, the integration of nozzles with adequate bore diameter into the valves is considered. This allows a simple reduction of the flow with the available valves and furthermore a simple adjustment of the cross-section, if needed.
Nozzles at the needed nominal diameters of less than 0.7 mm are standardised in the sector of water jet cutting. Their size ranges from 0.08 to 3 mm diameter at maximum pressures up to 400 MPa. For this project, a combination according to Table 3. was selected. The nozzles (080 S of the manufacturer Teschke GmbH) are made of sapphire, with a high choice in the exact diameter and a pressure resistance up to 100 MPa.

<table>
<thead>
<tr>
<th>diameter [mm]</th>
<th>cross section [mm²]</th>
<th>conductance [NL/s Pa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>7.9 e⁻³</td>
<td>140</td>
</tr>
<tr>
<td>0.125</td>
<td>12.3 e⁻³</td>
<td>220</td>
</tr>
<tr>
<td>0.175</td>
<td>24.1 e⁻³</td>
<td>430</td>
</tr>
<tr>
<td>0.25</td>
<td>49.1 e⁻³</td>
<td>880</td>
</tr>
<tr>
<td>0.35</td>
<td>96.2 e⁻³</td>
<td>1720</td>
</tr>
<tr>
<td>0.5</td>
<td>196.3 e⁻³</td>
<td>3500</td>
</tr>
</tbody>
</table>

Table 3. Selection of nozzles

The nozzles are sorted by their cross-section. In the order given, the following nozzle is doubled in cross section compared to the previous. A measurement of the sonic conductance of these nozzles confirms the ratio of mass flow through the nozzles. The valve EN 52050 is exactly twice the size of the largest nozzle. In total 7 valves are used, 6 of them with an integrated nozzle.

Aside to the improved regulation of flow through, the nozzles damp the control systems. This has two opposing effects: the controllability can be raised and at the same time the dynamic is reduced. It is not yet examined, if the combination of the nozzles must be adjusted between slow and fast forming speeds.

Though, with the presented combination, a binary ratio of the mass flow is realized. The area of the cross-section increases exponentially with the base 2 (e.g. 1, 2, 4, 8, ... mm²). In this case, a pulse code modulation (PCM) can be adapted. The PCM switches a combination of the valves depending on the value of the actuating signal [9]. Figure 8. (a) demonstrates the possible combination of cross sections (A) in an ideal binary ratio.

Figure 8. Possible combination of binary areas [12] (a) and resulting combinations with nozzle selection (b)
The resulting combinations for the presented selection of nozzles are depicted in Figure 8. (b). With 7 different valves, 128 combinations are possible. The combined conductance of the combinations characterises an almost linear dependency. The parallel connection of valves can therefore substitute a proportional valve, assuming a sufficient valve dynamic.

Preliminary tests with the nozzles to examine the controllability are presented in Figure 9. With the reduced mass flow through the nozzles, the pressure adjustment is much better. With a nozzle of 0.25 mm diameter (b) the pressure incline of 1 MPa/s is controlled better than with a valve without using a nozzle (Figure 7.). The small nozzles (a) allow exactly very slow forming processes.

![Figure 9. Bulge test with two different nozzles (0.1 mm at 0.1 MPa/s (a) and 0.25 mm at 1 MPa/s (b))](image)

6. SUMMARY AND CONCLUSION

The Hot Gas Bulge Test is a promising method to determine flow curves of sheet metals at elevated temperatures and biaxial stresses. A previous examination showed that the requirements applied for the Hot Gas Bulge Test can't be provided with the standardised bulge test because of the temperature and the strain rates. With the presented development, these conditions are considered.

On the basis of previous FEM simulations, the test bench was developed. Further, the simulation results could be used to derive the thermodynamic requirements for the pressure control. The concept of the control was designed by using seven parallel connected valves. Preliminary tests demonstrated the applicability of the valves for the given requirements and indicated potential for optimisation of the configuration. For the optimisation, a combination of nozzles was presented which enables a pulse code modulation on base of a binary code regulation and the Hot Gas Bulge Test was adjusted to this control design.

Future work will focus on the optimization of the Pulse Code Modulation, analysis of the valve combination for different test conditions and the increase of the valve’s dynamic by a power boost with higher voltages. In addition, controlled strain rates will be tested in combination with the measurement systems for the forming process.

7. ACKNOWLEDGEMENT

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NOMENCLATURE

\[ \begin{align*}
Q & \quad \text{heat} \quad [\text{J}] \\
V & \quad \text{volume} \quad [\text{m}^3] \\
W & \quad \text{technical work} \quad [\text{J}] \\
e & \quad \text{external specific energy} \quad [\text{J/kg}] \\
h & \quad \text{bulge height} \quad [\text{mm}] \\
h_i & \quad \text{enthalpy} \quad [\text{J}] \\
m & \quad \text{mass} \quad [\text{kg}] \\
p & \quad \text{pressure} \quad [\text{MPa}] \\
t & \quad \text{material thickness} \quad [\text{mm}] \\
u & \quad \text{internal specific energy} \quad [\text{J/kg}] \\
\varepsilon & \quad \text{technical strain} \quad [%] \\
\rho & \quad \text{curvature radius} \quad [\text{mm}] \\
\sigma & \quad \text{stress} \quad [\text{MPa}] \\
\varphi & \quad \text{local strain} \quad [-]
\end{align*} \]

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DEVELOPMENT OF EXPERIMENTAL EQUIPMENT FOR THE ANALYSIS
OF FLOWMETER CHARACTERISTICS IN CONDITIONS
OF GAS PULSATING FLOW

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ABSTRACT

The work presents the results of test stand development for experimental analysis of flowmeter error caused by gas pressure pulsations. Error analysis is based on comparison of the gas pulsating flow measured by the test flowmeter and the same gas quantity but without pulsations by a master flowmeter. To eliminate pressure pulsations in a pipeline after the test flowmeter the displacement-type oscillation damper used. The work shows theoretical and experimental pressure oscillograms and pressure pulsation spectra in the test stand mainline as well as the pulsation error research results of the orifice flowmeter.

KEYWORDS: gas flow metering, orifice flowmeter, pressure pulsations, error, experimental equipment, development, error analysis.

1. INTRODUCTION

The important problem arising in the operation process of gas metering stations is to increase the accuracy of pulsating gas flow metering. This problem becomes acute in gas flow metering by orifice flowmeters which are 40% of the total number of flowmeters. Pressure pulsations generally caused by nonuniformity of charger’s supply, instability of units and vortexes lead to the additional error which can reach 3,5% and more [1-18] depending on pressure pulsation level. Creation of hardware and software for correcting orifice flowmeter readout is considered to be advantageous for the pulsating gas flow measurement accuracy increase. However, development and implementation of correcting unit cannot be done without testing its efficiency by special stand equipment.

Literary sources and patents propose a range of stand schemes for flowmeter error researches. Grimpson J. and Hay N. [17] suggest an installation for evaluation of pulsating flow rate metering error based on simultaneous gas flow metering with and without pressure pulsations. The flow has been created by an air blower and changed by a flow control valve. Local flow accelerations caused by inertial effects, acoustic phenomena appearing in the form of pressure waves moving in the pipeline at the velocity of sound, hydraulic resistance of flowmeter impulse lines are indicated as main factors effecting the accuracy of pulsating gas flow metering. Torigoe Ippegi [18] in his work describes the method for determination of
acceleration at straight pipe segments with liquid or gas flowing, based on measurement of pressure difference between two points located at a small distance from each other. The work shows the method of pulsating flow rate measurement where this method is connected with the throttle flowmeter method. The work [19] gives the analysis at the standard level of not only quadratic pulsation error but also of the influence of pulsating flow inertial constituent on the error.

Gaptrakhmanov R.R., Fafurin A.V. in their works [1, 10] have used an experimental unit for evaluating flow pulsation influence on the orifice flowmeter error and on the orifice flow rate coefficient. It is necessary to note that the use of a metering nozzle as a control one without pressure pulsation dampening is not fully examined and can result in the error of master flow rate measurement. The influence of acoustic boundary conditions connected to the test gas pipeline flowmeter is not taken into account as well.

The real stands used do not fully satisfy to the specified requirements concerning precision setup of static parameters (pressure drop), formulation of acoustic boundary conditions in general, realization of pulsating gas flow setup mode. For this reason, creation of experimental equipment for the flowmeter error analysis in conditions of pulsating gas flow, theoretical and experimental research, aimed at the increase of gas flow rate measurement accuracy in the presence of pressure pulsations is an important task.

2. DEVELOPMENT OF THE STAND FUNCTIONAL DIAGRAM

Figure 1 presents the stand with a siren pressure oscillation generator developed by the authors. Generator 3 is series connected in the stand gas circuit and there is a throttle 4 at its inlet for regulating amplitude of pulsating gas flow. The gas circuit with the throttle 5 is bypass connected to the generator 3 for regulating average pressure at the test flowmeter inlet. Correcting element 6 in a form of cylinder channel with a piston and the throttle 7 allows to change pressure pulsation level at the test flowmeter inlet at the stand steady operation mode, i.e. without any changes of regulating unit positions. Herewith, it is possible to define with more accuracy the test flowmeter error only caused by pressure pulsations.

Correcting throttle 10 belongs to the boundary condition part at the test flowmeter 8 outlet. For example, if it is necessary to realize a long nonreflecting segment of the pipeline, the throttle 10 should have acoustic resistance equal to acoustic wave resistance of the pipeline at the test flowmeter outlet, and the reservoir 11
should not have volume less than a specified value. At the same time the reservoir 11 is the pressure pulsation dampening element at the master flowmeter inlet 12. There is a reservoir 13 at the master flowmeter outlet to avoid penetration of pressure pulsations from the throttle 14 installed for average pressure adjustment in the gas mainline. The important element improving accuracy of experimental researches is a hydromechanical differential pressure controller 9 at the test flowmeter 8. There are temperature sensor 15, pressure sensor 16 and differential pulsation pressure sensor 17 installed at the inlet of the orifice type flowmeter 8. Same type sensors are installed on the master flowmeter 12.

Experimental research technique is that gas flow rate is calculated according to the standard technique by average pulsating pressure component, the test flowmeter inlet gas temperature and its differential pressure set by the controller 9. The standard technique is also used for calculation of gas flow rate in the master flowmeter 12 based on temperature sensor 18, pressure sensor 19 and differential pressure sensor 20 readings. By difference of readouts of the test and master flowmeters the test flowmeter relative error caused by the influence of differential pressure pulsations is defined. Not only orifice flowmeters but also other types of flowmeters, e.g. turbine, ultrasonic, fluidic, can be used as a master and test flowmeters.

To select stand parameters it is necessary to develop a mathematical model of its gas mainlines, to conduct researches on the influence of various parameters on pressure pulsation level, to evaluate efficiency of means on error reduction in orifice and other flow meters while measuring pulsating gas flow.

3. PARAMETER SELECTION FOR STAND GAS MAINLINE BASIC ELEMENTS

One of the main parameters for the stand mainline is the diameter of pressure head pipeline which is limited from above by the capability of the blower to create maximum flow rate and gas pressure at the test segment. The unit is provided with pressurized air by the compressor station with the total volume of receivers 80 m³. Minimum diameter of the stand mainline is connected with the possible influence of switched measurement circuits on oscillating process at the segment with the test flow meter. For this reason it has been suggested to use the minimum pipeline diameter widely applicable in gas industry – 50 mm.

Later, experimental researches have shown that diameter 50 mm is enough to avoid the influence of connected metering circuits with channel diameters up to 8 mm on the oscillation process in gas mainline. The stand inlet pressure is specified as 3…5 MPa, and the stand mainlines as 0,5…3 MPa. Parameter ranges for other stand units and elements are as follows: throttles in a form of ball valves with flow section diameter 50 mm; air generator disk window diameter from 10 to 40 mm. The test and master flowmeter orifices are the same with the relative diameter 0.3. Selected pipeline lengths at the orifice inlet and outlet are not less than recommended by GOST [23]. Maximum length of correcting pipeline 6 (figure 1) is 6 m. Rotation frequency of generator drive is selected on account of pressure oscillation spectrum realized in industrial gas mainlines in the range of 3...100 Hz.

It has been suggested to use a reservoir 11 as a oscillation dampener, which parameter selection comes down to reduced volume definition $V_{gi}$ on the basis of specified degree of pressure oscillation smoothing in the frequency band $[\omega_1, \omega_2]$ according to the technique presented in the research work by RAS academician V.P. Shorin [24]. To provide oscillation smoothing degree not lower than the specified degree at the system segment, located after the dampener, for the worst conditions it is necessary to observe the following condition:

$$V_{gi} \geq \sup \left\{ \frac{\rho \cdot a^2 \cdot (\eta + 1) \cdot |Y_2(\omega)|}{\omega} \right\}_{\omega_1, \omega_2}$$

(1)

where $|Y_2(\omega)|$ - argument module of oscillation source total conductance and connected circuit; $\omega$ - oscillation angular frequency, $\eta$ — pressure oscillation smoothing degree after the dampener. As the dampener is located at a distance from the oscillation source, the reduced volume selection only from the
condition (6) can lead to the increase of oscillation amplitude at the inlet segment. To avoid this it is necessary to keep to additional in equation:

\[ V_{gi} \geq \text{sup} \{ a \cdot [A_0 + \rho \cdot a \cdot Y_{\text{load}}(\omega)] / \rho \} \text{ for } [\omega, \omega_0, \omega_f] , \]

where \( Y_{\text{load}} \) - load input acoustic conductance. For the best simultaneous smoothing of oscillations at the circuit inlet and outlet sections we select the maximum value of the volumes calculated by the formulas (1) and (2).

From calculation of stand mainline pulsation characteristics and condition analysis (1) and (2) in the range of frequencies up to 30 Hz we obtain that the value of dampener capacity should be more than 0,068 \( \text{m}^3 \) and for future realization \( V_{gi} = 0,08 \text{ m}^3 \) accepted.

4. MATHEMATICAL MODEL OF THE STAND GAS MAINLINES

Mathematical model of stand gas mainline is based on the principle diagram (Figure1), acceptance of specific simplifying assumptions, equations of mechanics and gas dynamics. There have been accepted the following assumptions for the equations: gas is uniform, process of gas flowing in the pipes is isothermal, flow rate distribution by cross-section is equal, hydraulic friction losses are quasisteady, gas flow velocity in the pipelines is lower than the velocity of sound. As we analyze the solution of nonlinear problem, subject to the presence of nonlinear elements in the unit, we use the method of characteristics to solve the system of equations. The principle of calculation method is that the gas circuit is divided into sections so that pressure and flow waves at all pipeline segments pass the distances between boundary sections in one and the same time interval \( \Delta t \), i.e. the segment in pipelines with higher velocity of sound is selected longer for the value corresponding to the increase of wave velocity.

Equations for gas flow in a pipeline with the account of friction losses have been derived by Charny I.A. [20] on the basis of the momentum formula (momentum theorem), continuity and gas condition at the selected segment of the pipeline. With the gas dynamics equations presented in a difference form it is possible to define pressure and flow rate in the \( i \) unit of the pipeline at time moment \( t_0 + \Delta t \), if we know pressure and flow rate in \( i-1 \) and \( i+1 \) cross-sections at time moment \( t_0 \):

\[ \begin{align*}
G_{i+1,\Delta t} &= \frac{1}{Z_{w,i-1} + Z_{w,i+1}} \left[ p_{i+1,\Delta t} - p_{i-1,\Delta t} + (k_{\lambda,i-1} + k_{\lambda,i+1}) \cdot G_{i,\Delta t} \right] G_{i,\Delta t} / \rho_{i,\Delta t}, \\
\rho_{i,\Delta t} &= p_{i,\Delta t} - Z_{w,i-1} \cdot (G_{i,\Delta t}) + k_{\lambda,i-1} \cdot G_{i,\Delta t} / \rho_{i,\Delta t}, \\
\rho_{i+1,\Delta t} &= p_{i+1,\Delta t} / (R \cdot T_i)
\end{align*} \]  

where \( G \) - mass gas flow rate; \( p \) - pressure; \( \rho \) - gas density; \( R \) - gas constant; \( T_i \) - temperature; \( k_{\lambda,i-1} = \frac{\lambda_{i-1} \cdot \Delta x_{i-1}}{2 \cdot d_{\theta,i-1} \cdot A_{i-1}} \) and \( k_{\lambda,i+1} = \frac{\lambda_{i+1} \cdot \Delta x_{i+1}}{2 \cdot d_{\theta,i+1} \cdot A_{i+1}} \) - resistance coefficient for a segment, left and right from \( i \) cross-section; \( Z_{w,i-1} = a_{i-1} / A_{i-1} \) and \( Z_{w,i+1} = a_{i+1} / A_{i+1} \) - wave resistances of pipeline segments, left and right from \( i \) cross-section; \( A_0, d_\theta \) - flow area and diameter of gas main pipeline; \( a \) - velocity of sound emission in the pipeline; \( \lambda \) - hydraulic loss coefficient; \( \Delta x \) - length of pipeline segment; \( t \) - time.

Solving the system of difference equations (3) by successive approximation it is possible to define pressure, mass flow rate and density at \( i \) unit at the time moment \( t_0 + \Delta t \). The equations for pipeline connection unit are written and solved in a similar way.

To describe the throttle the equations of gas flow through the throttle are added to the equations (3):
\[
p'_{i,t_0 + \Delta t} - p_0 + Z_{w,i-1} \cdot (G_{i,t_0 + \Delta t} - G_{i-1,t_0}) + k_{i,j-1} \cdot G_{i,j,t_0 + \Delta t} \bigg| G_{i,t_0 + \Delta t} / \rho'_{i,r_0 + \Delta t} = 0, \\
- p''_{i,t_0 + \Delta t} + p_0 + Z_{w,i+1} \cdot (G_{i,t_0 + \Delta t} - G_{i+1,t_0}) + k_{i,j+1} \cdot G_{i,j+1,t_0 + \Delta t} \bigg| G_{i,t_0 + \Delta t} / \rho''_{i,r_0 + \Delta t} = 0, \\
p''_{i,t_0 + \Delta t} = p_{i,r_0 + \Delta t} / (R \cdot T), \\
p''_{i,t_0 + \Delta t} = \frac{\mu A_g}{2} \cdot \left[ 2 \cdot p'_{i,t_0 + \Delta t} \cdot \rho'_{i,t_0 + \Delta t} \cdot \frac{k}{k-1} \left( \frac{p''_{i,t_0 + \Delta t}}{p'_{i,t_0 + \Delta t}} \right)^{2/k} - \left( \frac{p''_{i,t_0 + \Delta t}}{p'_{i,t_0 + \Delta t}} \right)^{(k+1)/k} \right], \\
p''_{i,t_0 + \Delta t} = \frac{\mu A_g}{2} \cdot \left[ 2 \cdot p'_{i,t_0 + \Delta t} \cdot \rho'_{i,t_0 + \Delta t} \cdot \frac{k}{k+1} \left( \frac{2}{k+1} \right)^{2/(k-1)} \right], \\
\]

where \( \mu \) - flow rate coefficient, \( p'_{i,t_0 + \Delta t}, p''_{i,t_0 + \Delta t} \) - pressure at the throttle inlet and outlet, \( T'_i, T''_i \) - gas temperature at the throttle inlet and outlet; \( p''_{i,t_0 + \Delta t}, p'_{i,t_0 + \Delta t} \) - gas density at the throttle inlet and outlet.

Formula (5) is applied for the relation \( \frac{p''_{i,t_0 + \Delta t}}{p'_{i,t_0 + \Delta t}} > \beta_{cr} \), i.e. for the case of subcritical pressure drop, and formula (6) - for the relation \( \frac{p''_{i,t_0 + \Delta t}}{p'_{i,t_0 + \Delta t}} \leq \beta_{cr} \) or supercritical pressure drop on the throttle, \( \beta_{cr} = (2/(k+1))^{1/k} \); \( k \) - adiabatic coefficient; \( A_g \) - throttle flow area.

The system of equations for the throttle (4) is used for description of the generator, with the exception that in formulas (5) and (6) cross-section area \( A_g \) and flow rate coefficient \( \mu \) are functions of the chopping disk rotation angle and time. Flow area of the generator section channel is defined by the difference of sectional areas of the pipeline and the rotating generator disk with \( A_g = f(\omega t) \), where \( \omega \) - rotation frequency of the generator electric driving motor. Flow rate coefficient \( \mu \) is calculated by the formula [21] \( \mu = \frac{l}{\sqrt{\zeta + l}} \), where \( \zeta \) is calculated by the dependence presented in [22].

Modified difference equation for the mainline unit with a reservoir is as follows:

\[
G''_{i,t_0 + \Delta t} = G'_{i,t_0 + \Delta t} - C_a \cdot N_{bas} \cdot a_{bas} / l_{bas} \cdot (p'_{i,t_0 + \Delta t} - p_{i,t_0}) \cdot \rho'_{i,t_0 + \Delta t},
\]

where \( C_a = V_{gi} / (k \cdot P_0) \) - acoustic capacity of the cavity volume; \( V_{gi} \) - reduced cavity volume, \( P_0 \) - average pressure in the reservoir; \( l_{bas} \) - basic length of main pipelines; \( N_{bas} \) - partition number of basic lengths; \( a_{bas} \) - velocity of sound in the basic pipeline.

Pressure drop controller on the orifice is a pneumohydromechanic unit with a long-time constant and, by this reason, it is not considered in the mathematical model of stand oscillating processes.

Calculation of stand mainline characteristics is made in the following sequence: first we calculate the static parameters including flow rate in a gas circuit. Then, taking this flow rate for the first approximation, we calculate pressure losses on other elements of the circuit. We calculate flow of the second approximation with residual pressure drop on the outlet throttle. The system of equations is solved by the method of successive approximation. Algorithm and calculation program for static and pulsation characteristics of stand mainlines have been developed on the basis of presented mathematical models of stand assemblies and methods. The computer calculation program has been realized by C++ programming.
5. ANALYSIS OF THE STAND PULSATION CHARACTERISTICS

Pressure and flow oscillations in gas pipe special sections were calculated based on developed mathematical model. During researches of the stand pulsation characteristics the parameter ranges have been set preliminary to define limits for realization of the set pulsation mode at the test segment of gas mainline. Figure 2 shows calculated dependency of pressure drop and its spectrum for test and master orifice flow meters at 30 Hz pulsation frequency and fully closed generator 3 bypass.

![Diagram of pressure drop and its spectrum](image1)

**Figure 2.** Theoretical oscillogram and pressure oscillation amplitude spectrum at the test (a) and master (b) flowmeters with oscillation frequency 30 Hz.

Figure 3 shows similar charts with dependency on gas flow rate oscillation through test and master flow meters.

![Diagram of gas flow rate oscillation and its spectrum](image2)
Figure 3. Theoretical oscillogram and flow oscillation amplitude spectrum through the orifice of the test (a) and the master (b) orifice flow meters with oscillation frequency 30 Hz.

Application of generator bypass throttle 5 for regulating effective value level of the pipe gas pressure oscillations is feasible to a greater extent on frequencies up to 30 Hz. For frequency higher than 50 Hz and with bypass closing more than 10% the influence of resonance phenomena becomes more noticeable, and it is impossible to achieve smoothness of regulating the effective oscillation value by the generator bypass valve 5.

Connecting a correcting device 6—a pipe with changeable length with open valve 7 at its end gives a drop in pressure oscillation amplitude at the test flow meter inlet on its resonant frequency. The use of correcting device 6 significantly changes the harmonic amplitude at its tuning frequency with saving gas pressure and flow average values. Thus for 40 Hz pressure oscillation frequency connecting 4 m long correcting pipe 6 decreases base harmonic amplitude 10 times (figure 4). By this it is possible to change pressure drop at the test orifice flow meter without altering the flow average values, which allows improving the accuracy of experimental readings. Efficiency of correcting unit application is shown in Figure 4.

Figure 4. Pressure oscillation amplitude spectrum at the test orifice flow meter inlet with frequency 40 Hz and a shutoff generator bypass:

The use of the correcting pipeline causes a major change in harmonic amplitude on its adjustment frequency without changing average values of pressure and gas flow rate. With length of the correcting pipeline 4 m and oscillation frequency 40 Hz, it will lead to approximately 10-time (tenfold) decrease of main harmonic amplitude.
Figure 5. Experimental oscillogram (a) and spectrum (b) of the test orifice flowmeter differential pressure oscillations with frequency 50 Hz.

Figure 6. Experimental dependence of quadratic oscillatory error of the test orifice flowmeter (1) and empiric dependence on the ISO (2) formula of relative effective oscillation value in the frequency range 5…60 Hz.

It is seen from figures 5 and 6 that developed on described methodology stand equipment can be used for estimation of flowmeter readings additional error due to gas pressure pulsations and also to measure the effectiveness of the proposed improvements.

6. CONCLUSION

- Experimental research showed that the stand equipment for the study of pressure pulsation influence on error of gas flow meters should consist of a gas pressure source, oscillation generator, test flowmeter pressure drop controller, master flowmeter, boundary conditions unit, correcting element in form of a resonator pipe connected through the throttle to the test flowmeter inlet.
- Selection procedure for main stand parameters is described herein. The work presents mathematical model, calculation methods and algorithm for gas flow parameters in test stand mainlines, realized with the software package.
- The work contains theoretical and experimental data on pressure oscillations and their frequency spectrum, showing the test stand options for realization of pulsation patterns at the position of the test and master flow meters.
- This test stand can be recommended for efficiency evaluation of means on error reduction in orifice and other flow meters while measuring pulsating gas flow.
REFERENCES


ENERGY EFFICIENCY COMPARISONS OF PNEUMATIC SYSTEM: EFFECTS IN AFTER TREATMENT, THEORY AND VERIFICATION WITH TIME SERIES MEASUREMENTS

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ABSTRACT

The idea of air circulation for reuse purposes in pneumatic systems has become an important alternative. The CPS (Closed Pneumatic System, patented innovation) has been implemented in some industrial plants in Finland. Author has dealt the subject in detail in his publication [1]. However the effect of the circulation of compressed air on the total energy efficiency of a specific system strongly depends on different factors, like compressor type and type of the after treatment.

The effect of the after treatment system to the total energy efficiency of the pneumatic system varies considerably in different applications. Especially when a pneumatic system with an heatless adsorption dryer is in use, the proportion of compressed air of the total production needed for degeneration purpose is considerable large (according to various references 15-20%).

In this paper, theoretical calculations for the consumption of energy based on generic models of pressure behavior in different cases (pressure-dependent/independent, 2-point-controlled and inverted control) are presented. According to actual measurements in industrial plants, the theory is in line with actual pneumatic systems. These measurements include values for compressed air flow, pressure (bar) and power/energy. In these measurements the measurement data is first collected from the open pneumatic system and then the measurements have been repeated with the closed system.

KEYWORDS: After treatment, adsorption dryer, energy efficiency, time series measurements
1. INTRODUCTION

According to many investigations [3] there is plenty of room for improvements in the energy efficiency of industrial pneumatic systems. Since 10% of the electrical energy in EU is used for the production of compressed air, the energy efficient use is important in the development of any industrial process. A comprehensive investigation [4] lists various means to improve the energy efficiency including the electrical energy usage of production machines and further different life-cycle oriented optimization measures that have been derived [5]. The idea of finding measures of quantifying energy efficiency based on analytical calculations and measures of air consumption [6],[7] as well as exergy-based analysis [8] have also been a subject of discussion.

The latest innovations [1] have made it possible to avoid the energy consuming effects of the after treatment system. The effect of the after treatment to the energy efficiency is dealt in detail with analytical approach from first principles. In this consideration, there are two separate points of view.

One possible approach is to consider the longer term effects working period of the after treatment system. The maintenance of the system typically takes place only once or twice a year making the working period thousands of hours long.

Another approach is to consider the compressor to be two-point controlled i.e. it is either switched on and produces at a maximum capacity or is switched off and is idle running. With this approach, plenty of measurement data from industrial application is available for time-series based analysis.

2. PRESSURE LOSSES IN AFTER TREATMENT OF PNEUMATIC SYSTEM

The pressure losses between production and consumption points typically take place in after treatment system. Every filter in the after treatment has an effect of its own. The number and type of filters typically depends on the application and the quality class of CA [8,9]. In addition, the shape, material and topology of piping network causes pressure losses [8,9]. In this study, the energy losses in after treatment system of the compressor are analysed.

Typically, when the quality class is high, adsorption dryer is used. However, the adsorption dryer is also used in many industrial applications that have no quality class specifications. When a system of one compressor is used, the compressor type is often a screw compressor and regulation method is inverted.

The pressure losses in an after treatment system characteristically grow when the contamination level of filters deteriorates. Typically, the need of CA varies on daily basis. By denoting the pressure losses in after treatment at the moment when the need (consumption of CA) is at largest by $\Delta \pi_{\text{max}}$ and assuming that the time period starts at the point of maintenance ($t=0$) when the initial pressure losses = $\Delta \pi_0$. The loss of pressure at the maximum consumption grows as a function of time. It can be further assumed, that the consumption grows rather linearly after the maintenance so the pressure loss at a time moment $t$ is

$$\Delta \pi_{\text{max}}(t) = \Delta \pi_0 + kt$$  \hspace{1cm} [1]

where $k$ is constant

$t$ is time in hours, typically the maintenance takes place after 3000-4000 working hours.

A special case $k = 0$ corresponds to a constant pressure loss. This can take place when the after treatment and its effects are eliminated and only the piping network causes the pressure losses.

The consumption of CA is a time function. In that case, the pressure loss $\Delta \pi(t)$ can be expressed as a function of time

$$\Delta \pi(t) = \left\{q'(t)/q'_{\text{max}}\right\} (\Delta \pi_0 + kt)$$  \hspace{1cm} [2]
\( q'(t) = \) production of compressor (normal m\(^3\))

\( q'_{\text{max}} \) is the maximum production of compressor

In order to generate pressure level \( \pi_2(t) \) to the consumption point, let the needed outlet pressure of the compressor be \( \pi_1(t) \). Concluding from the previous, the pressure (bars) in the consumption point is

\[
\pi_2(t) = \pi_1(t) - \Delta \pi(t) = \pi_1(t) - \left\{ \frac{q'(t)}{q'_{\text{max}}} \right\} (\Delta \pi_0 + \kappa t)
\]  

[3]

In practice, an inverter controlled compressor's output pressure is constant = \( \pi_1(T) \), in order to make to inlet pressure reasonable high during the operating period \([0,T]\). The unit of time \( T \) is in hours.

From thermodynamics [8], the total input power of the compressor during the operating period \([0,T]\):

\[
P(t) = A \cdot q'(t) \ln(\pi_T(T))
\]  

[4]

, in which \( A \) is a constant that depends on the isothermal coefficient of efficiency

The average inlet power \( P_{av} \) during the operating period can be obtained from:

\[
P_{av} = \frac{A \ln(\pi_1(T))}{T} \int_0^T q'(t) dt = A \ln(\pi_1(T)) Q_{av}
\]  

[5]

, in which

\[
Q_{av} = \left( \frac{1}{T} \right) \int_0^T q'(t) dt
\]  

[6]

If the pressure losses in the after treatment system can be avoided, the outlet power of the compressor can be set at a lower level to the value of the relative pressure \( \pi_0 \) and keep it at that level permanently during the operating period \( T \). The energy savings \( \Delta P \) during the period thus become

\[
\Delta P = P_{sv} - 1/T \int_0^T A q'(t) \ln(\pi_0) dt
\]  

[7]

\[
= P_{sv} - A \ln(\pi_0) \left\{ \frac{1}{T} \int_0^T q'(t) dt \right\}
\]  

[8]

\[
= A \ln(\pi_0(T)) Q_{av} - A \ln(\pi_0) Q_{av} = A Q_{av} \ln\left( \frac{\pi_0(T)}{\pi_0(0)} \right)
\]  

[9]

In fig.1 typical profiles for actual and theoretical pressure losses as a function of time are presented. The magnitude of the time is thousands of hours. The linear part of the curve \( (\Delta \pi(t,q_{\text{max}})) \) presents the theoretical maximum losses while the broken line below it \( (\Delta \pi(t,q(t))) \) corresponds the actual pressure losses.
During a working period (typically $T$ is of magnitude of thousands of hours) the pressure consumption is likely to grow quite linearly due to contamination in after treatment. In practice, at the starting moment $t = 0$, immediately after the maintenance of the after treatment, the pressure losses are at lowest. The pressure losses at the maximum (in the end of the maintenance period) grows up to $\Delta \pi_{\text{max}}$.

In modern industrial pneumatic systems, the pressure levels of compressors are often set manually too high. The pressure levels typically are not changed during the operation period. Pneumatic actuators naturally work with over pressure, however it is obvious, that this is very energy consuming.

3. CASE STUDY

Typically, when a compressor is two-point controlled, its production may be either pressure-dependent or pressure-independent. The shape of the pressure curve depends on the type of consumption. Usually the consumption is pressure-dependent [2,10] making the pressure curve exponential (Fig.2). For computational simplicity, on the following the process is assumed to be pressure-independent.

The input power of the compressor can be obtained from the equation:

$$P = \frac{1}{\eta}RT_0 \ln(p_f/p_0)$$

[10]
in which $R =$ gas constant, $T_0 =$ suction air temperature, $p_1 =$ final pressure after compression, $p_0 =$ atmospheric pressure (= 1 bar) and $\eta =$ compressor's isothermal coefficient of efficiency, i.e. relation of isothermal compression ($P_{iso}$) compressor's input power (shaft power $P_{axial}$)

$$\eta = \frac{P_{iso}}{P_{axial}}$$  \[11\]

The isothermal coefficient of efficiency is typically between 0.5 ... 0.7 \[8\]. When the suction volume flow $q =$ 1$m^3$/s the isothermal power for the final pressure 8 bar is 208 kW,

$$P = qRT_0\ln(p_1/p_0)$$ \[12\]

, from which

$$RT_0 = P/(q\ln(p_1/p_2))$$ \[13\]

3.1. Examples of savings potential in an industrial application

The average power $P_{AVE}$ during the duty cycle can be obtained from the equation:

$$P_{AVE} = \left\{ \frac{P_{ave} t_1 + P_0(t_2 - t_1)}{T} \right\}$$ \[14\]

in which $P_{ave}$ is the average value of the compressor power when it is switched on $t_1$, $P_0$ is the idle running power during period $(t_2 - t_1)$ and $T$ is the length of the duty cycle. During the production, the outlet pressure of the compressor rises from value $\pi_1$ ($\pi_{min}$) to value $\pi_2$.

$$\pi(t) = \pi_{min} + \left( \frac{\Delta \pi}{t_1} \right) * t$$ \[15\]

in which $t \in [0,t_1]$.

By integrating the previous equation

$$\left( \frac{1}{t_1} \right) \int_0^{t_1} \pi(t) dt = \left( \pi_2 \ln(\pi_2) - \pi_1 \ln(\pi_1) - \Delta \pi \right) / \Delta t$$ \[16\]

The average power needed per cubic meter is:

$$(P_{ave}/1Nm^3/min) = 1.67 \left( 1/\eta \right) \left( \pi_2 \ln(\pi_2) - \pi_1 \ln(\pi_1) - \Delta \pi \right) / \Delta \pi [kW]$$ \[17\]

When $q =$ 1$m^3$/min,

$$RT = 208/60*\ln(8) = 1.67 \text{ kW}.$$ \[18\]

The inlet power of compressor for the production of 1Nm$^3$/min can be obtained for different absolute final pressures:

**Example 1:** The average production of the compressor is 15 Nm$^3$/min. The input power of the compressor for coefficients of efficiencies a) $\eta_1 = 0.5$ ; b) $\eta_1 = 0.7$. for the final pressure $p_1 = 8$ bar. a) $P = 105 \text{ kW}$, b) $P = 75 \text{ kW}$.

During the duty cycle, the compressor is either switched on, when its outlet pressure is a function of time $P_1(t)$ or is switched off, when it is idle running and its input power is a constant $P_0$. 
Example 2: If π₂ = 7 and π₁ = 6 making Δπ =1. η = 0,5, and consumption 10 normal m³/min. The average power when the compressor is switched on is \( P_{\text{ave}} = 63 \text{ kW} \).

The average power can also be expressed with equation:

\[
P_{\text{AVE}} = \left( \frac{Q}{q} \right) P_{\text{ave}} + \left( \frac{q' - Q}{q} \right) P_0,
\]

in which \( Q \) is the consumption during the duty cycle, \( q' \) is the production of the compressor and \( P_0 \) is idle running power.

Example 3: In the previous example, the production \( q' = 15 \text{Nm}^3/\text{min} \) and idle running power \( p_0 = 30 \text{ kW} \). During the duty cycle, the consumption is constant = 10 Nm³/min and the compressor is switched on during the idle running period. During the duty cycle the power is \( P_{\text{AVE}} = 25,2 \text{ kW} + 15,0 \text{ kW} = 40,2 \text{ kW} \), in which the proportion of idle running is 15.0 kW (37%).

3.2. Comparison between the power consumption in open system and the closed system

When the CPS is taken into use in an industrial pneumatic system, the after treatment system can sometimes be totally bypassed. Actually when the CPS is in use, there is a need for CA to compensate the losses in consumption. It is naturally impossible to gather all the CA for recirculation, but in practice, the need of compensation air is minor and can be supplied for example by another, booster-like compressor. Also in industrial applications, the CA systems are only partially implemented and the need of the compensation air can be easily supplied. The proportion of the regeneration of the outlet air of the compressor is quite large, in some applications even larger than the air needed in consumption points.

Typically, in a two-point controlled compressor system, the values of pressure (bars) in a consumption point act quite monotonically. The values increase and decrease depending on the pressure difference values of the compressor. An illustrative example of pressure time series is presented in figure 3. In this example (60 measurement samples) the pressure typically increases and decreases between 6.5-8 bar. i.e. the pressure difference of a two-point controlled compressor is about 1.5 bars.

![Figure 3. An example of a pressure curve of a two-point controlled compressor](image)

Characteristically, the consumption in an open system is considerably larger than in a closed system, i.e. \( Q^{\text{OPS}} >> Q^{\text{CPS}} \). Also, verified in measurements [1], figures 4-5, and the following equation for the cycle time holds:

\[
t_1^{\text{OPS}} >> t_1^{\text{CPS}}
\]

[20]
and

\[(t_2 - t_1)^{OPS} \ll (t_2 - t_1)^{CPS}\]  \[21\]

The graphs in figures 4 and 5 present the actual measurement data performed at a woodworking plant (a two-point controlled compressor, adsorption dryer in after treatment). The measurements represent the relative pressure (bars) as a function of time. In this case, the sampling period is 5 minutes and sampling rate one measurement in second making the length of the time series 300. From the figures, it can be observed that the profiles are somewhat different. Figure 4 has some similarities with the theoretical model (fig.2) while in the profile in figure 5, the rise time is shorter. According to the measurement data [1], the proportion of the rise time of the total cycle time was 68% when OPS in use and 43% when CPS in use.

It is noteworthy that, compared with the figure 3, the measurements in the figures 4 and 5 are sampled at a higher frequency making the time series more broken line kind. However, the general trend can be easily identified from the figures.

![Figure 4. Time series measurement data of OPS (sampling frequency 1/s).](image)

![Figure 5. Time series measurement data of CPS (sampling frequency 1/s).](image)

**Example 4:** \(Q^{CPS} = 0.65 \text{Nm}^3/\text{min}\) and \(Q^{OPS} = 1.95 \text{Nm}^3/\text{min}, V = 0.3 \text{m}^3,\) and \(q^* = 2.2 \text{Nm}^3/\text{min}\). The need of regeneration air is \(1.3 \text{Nm}^3/\text{min}\).

The average power consumption then depends on the idle running power of the system.

The power consumption with open and closed systems are \(P^{OPS} = 10.9 + 0.7 = 11.6 \text{ kW}, P^{CPS} = 1.2 + 4.2 = 5.4 \text{ kW},\) making savings 6.2 kW i.e. 53%.
If the CPS is implemented with an inverted oil injected screw compressor, the inlet pressure of the consumption is a constant \( \pi^{CPS} = \pi^{OPS}_{\text{min}} - \Delta \pi^{OPS} \), in which \( \Delta \pi^{OPS} \) is the pressure loss in after treatment and \( \pi^{OPS}_{\text{min}} \) is the starting pressure of the compressor. The compressor then produces the air volume:

\[
Q^{CPS} = Q^{OPS} - \Delta Q^{OPS},
\]

in which

\( \Delta Q^{OPS} \) = average air volume needed in regeneration. Since there is no idle running period, the inlet power of the compressor is then optimal and can be obtained from the equation

\[
P^{CPS}_{\text{OPT}} = 3.34 \times (Q^{OPS} - \Delta Q^{OPS}) \ln(\pi^{OPS}_{\text{min}} - \Delta \pi^{OPS}).
\]

**Example 5:** The pneumatic system of previous example is implemented with an inverter controlled compressor in CPS. In that case, the outlet pressure is lower than with the open system. The difference \( \Delta \pi^{OPS} = 0.6 \) bar. There is no idle running period so the optimal power consumption is \( P^{CPS}_{\text{opt}} = 3.7 \) kW. The power need is reduced by 30% compared to the open system.

4. CONCLUSIONS

In this consideration, the overall effect of the after treatment on the energy efficiency has been analyzed with different configurations starting from first principles. The heatless adsorption dryer, which has especially energy consuming properties has dealt separately as a case study. From the analysis, it becomes evident that the after treatment system has a major effect on the energy consumption on industrial pneumatic systems.

On the other hand, in many applications, the innovations [1] can be applied to the existing CA systems with an energy efficiency improving way.

A considerable improvement to the open system is achieved for two reasons. First of the effects are related to the heatless adsorption dryer, which uses a considerable large proportion of the compressed air for regeneration. However, in some applications, the adsorption dryer can be even left out from the pneumatic system and its losses can be avoided. Secondly, all the components (filters) of the after treatment system typically cause pressure losses of magnitude of 0.5-0.7 bar, when open system is in use.

**LIST OF SYMBOLS**

- \( \Delta \pi^{0} \) initial pressure loss caused by after treatment
- \( \Delta \pi^{t, \text{end}}(t) \) pressure at the moment of time (hours)
- \( \Delta \pi^{t}(t) \) pressure loss
- \( q(t) \) production of compressor (normal cubic meters)
- \( q^{\text{max}} \) maximum production of compressor (normal cubic meters)
- \( \pi^{1}(t) \) outlet power of the compressor
- \( \pi^{2}(t) \) pressure level at a consumption point
- \( \pi^{0} \) constant
- \( T \) time period
- \( P(t) \) total input power of the compressor during the operating period
- \( P_{\text{av}} \) average inlet power during the operating period
- \( Q_{\text{av}} \) average consumption during the operating period
- \( \pi_{\text{opt}} \) the optimal pressure
- \( \Delta P \) energy savings
- \( R \) a gas constant
- \( T_{s} \) suction air temperature
- \( p_{0} \) atmospheric pressure
- \( p_{f,2} \) final pressure after compression
- \( P_{\text{axial}} \) isothermal compression power
- \( P_{\text{shaft}} \) compressor’s input power (shaft power)
• $\eta$ compressor's isothermal coefficient of efficiency
• $P_{ave}$ average power during the duty cycle
• $P_{on}$ average value of the compressor power when switched on
• $P_0$ idle running power
• $t_{1,2}$ moment of switching (on/off) in a two-point controlled compressor
• $Q$ consumption during the duty cycle
• $Q'$ production of the compressor
• $Q^{OPS}$ consumption in an open pneumatic system
• $Q^{OPS}$ consumption in a closed pneumatic system
• $t_{ops(cps)}$ cycle time in an open (or closed) pneumatic system
• $\Delta Q^{OPS}$ average air volume needed in regeneration
• $P_{opt}$ optimal inlet power of the compressor

ABBREVIATIONS

• CA Compressed air
• CPS Closed Pneumatic System
• OPS Open Pneumatic System

REFERENCES

DESIGN AND FABRICATION OF AN ELECTROMAGNETIC MICROVALVE FOR PNEUMATIC CONTROL OF MICROFLUIDIC SYSTEMS

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ABSTRACT

This work presents the design, fabrication and experiment of an electromagnetic microvalve for pneumatic control of microfluidic chips to miniaturize the whole pneumatic control devices. The microvalve is composed of two parts, a miniature electromagnetic actuator and a valve body. The two parts are fabricated separately and assembled. The valve body are fabricated by a soft lithography process of elastomeric materials (Polydimethylsiloxane, PDMS). Tests showed that the designed microvalve worked at pressures up to 280kPa with the applied voltage of 24V. The fast open response (~6.4 ms) and close response (~4.0 ms) of the valve was achieved with a leak rate as low as 1.56×10^{-5} m^3/h at 200kPa (N_2) pressure. We tested the pertinent dynamic parameters such as flow rate in on/off mode, flow rate of duty cycles and actuated frequencies in pulse width modulation mode (PWM). Our method provides a simple, cheap and small microvalve which avoids the bulky and expensive external pressure control solenoid manifold. This allows it to be easily integrated into portable and disposable devices.

Keywords: microvalve, PDMS thin diaphragm, encapsulation process, fluidic behavior, pneumatic control.

1. INTRODUCTION

Microfluidic technique has attracted increasing wide attention and become one of the hot topics in recent years [1]. There are various kinds of actuation methods developed for micro devices on microfluidic chip to control the small volume output of chemical or biological working fluids. The bulky volumes of actuators have definitely affected the degree of Microfluidic Large-Scale Integration (mLSI) [2]. One exception is pneumatic actuation micro devices, which usually use actuators are located outside the fluidic chips, other actuators must be embedded in the microfluidic chips. However, the microfluidic systems themselves have a small footprint, the necessary support equipment is often bulky and non-portable [3]. The outside pneumatic control system usually includes: a supply source of compressed air, a solenoid valve array, a sensor and a controller unit/PC [4]. The necessary valve array and pneumatic control system stand-alone to support the pump occupy large spaces. For example, many microfluidic systems rely on external valves to drive valving internal to the microfluidic chip. As the number of internal valves can run into the hundreds, or even thousands, the number of external valves and the size and cost associated with them is a significant problem [5].

To avoid the usage of an external pneumatic control system for pneumatic microfluidic chips (PMC), some general approaches are presented. The Whitesides research group [6] experimented with a torque-actuated
valve that directly used small machine screws to press on PDMS micro channels for controlling fluid flows. Also, Y Zheng et al. [7] studied a screw-actuated pneumatic valve, which, by turning the machine screws up and down for pressure control in the control channel filled with water, increased pressure leading to collapse the PDMS membrane into the microfluidic channels and control the flow of fluids therein. One of the drawbacks of these last two microvalves is that the valves cannot be operated automatically because of the required slow manual manipulation. Sia and co-workers [8] used a modified solenoid with a microcontroller and peripheral circuitry instead of a manual screw-actuator mentioned in Zheng et al., to solve the problem of slow manual manipulation. However, the microvalve via liquid-filled control channel cannot be reused because of the evaporation of water through surrounding PDMS wall in very short span of time (less than 24h), and refilling liquid into the control channel would be need at least 48h to cure the sealing membrane. Another way of solving the problem of a bulky external air supply and its control solenoid valves is a thermo-pneumatic PDMS microvalve, used with a heating electrode, which can be coupled to a flexible membrane for generating an on-off valve that requires a simpler control circuit and less peripheral equipment [9]. However, the thermo devices can generate high temperatures and destroy the bioactivity of biological reagent. Phase change valves based on the paraffin solid-liquid phase transition are reported in references [10, 11]. The devices with an integrated two-level cooling/heating system are simple to operate, and valve operation is relatively slow. The Takayama research group [12] proposed a microvalve integrated with multilayer soft lithography technique and controlled by Braille display equipment, instead of pneumatic actuators. The disadvantage of the Braille actuator is its inflexibility in one sense because of the position-fix movable pins. A polyvinylidene fluoride (PVDF) microvalve for gas flow control was designed by Wiederkehr et al [13] to control gas flow rate with piezoelectric actuators through a glass micronozzle. The drawbacks are a complicated microeletromechanical systems (MEMS) processing method for rigid material used in valve body and high voltage DC300V. Anjewierden et al. [5] developed an electrostatic microvalve for pneumatic control of microfluidic systems, which was able to diminish the external actuation solenoid valves for gas serving. However, the disadvantage is a huge footprint for the final assembled device, with its two 75mm long valves, PMMA (Polymethylmethacrylate) valve plate and high power that has an actuated voltage even higher than 680V. Such a system would not allow for a portable, low power system with integrated controls.

The purpose of the research presented here is to design a microvalve that is inexpensive, easy to manufacture, has a small footprint, can function as an external valve controlling a pneumatic internal valve and can be easily integrated with a complex polymer microfluidic system. What is unique about the valve presented in this paper is not only that it takes advantage of PMC, in which the control system and pressure generation is usually off chip, but also that it avoids the huge off-board solenoid valve array—usually part of the outside of the pneumatic control systems.

We presented the design, fabrication, and experimental results of the electromagnetic microvalve, and its fluidic characteristics using N₂ flow under differential pressures, duty cycles, and frequencies were tested in this paper. The measured results suggest its application to PMC because its differential pressure is fully able to meet the PMC’s integrated valve array and can minimize the outside pneumatic control system.

2. WORKING PRINCIPLE AND DESIGN

We designed its novel structure which includes a miniature electromagnetic actuator and a valve body with a microchannel. An electromagnetic actuator is suitable for the microvalve because it has a larger displacement, is suitable for linear control, and has faster actuation and lower power usage than other popular actuators used in microfluidic systems. The schematic of the electromagnetic actuator is shown in Figure 1 (a). It shows that the actuator consists of several main components: 1=plastic housing; 2=ferromagnetic shell; 3=coil; 4=static iron; 5=moving iron; 6=return spring; 7=positioning pins; 8=fixing device; 9=glass slide; 10= valve core. The valve core moves up when voltage is applied because of electromagnetic suction between the moving iron and the static iron. The valve core is parallel with the lower surface of the
glass slide, and the return spring is in a compressed state. Then, without voltage, the valve core moves down under the action of the return spring to the initial state again.

The basic structure of the microvalve is shown in Figure 1 (b): the top is the electromagnetic actuator with a valve core, the bottom is the PDMS block with a microchannel, and the middle layer is a soft PDMS thin film, which also acts as the valve diaphragm. The valve core and the diaphragm consist of the valve head, which can close or open the microvalve. As shown in Figure 1 (c), the microvalve is normally closed because of the return spring, which makes the middle diaphragm bend down, and the microvalve close. The valve core moves up when a voltage is applied, the deflectable diaphragm recovers because of its elasticity and the air flows through the micro channel. With the voltage off, the deflectable membrane deflects downwards to seal off the gas channel, fully closing the microvalve.

![Diagram of the microvalve structure](image)

3. MATHEMATICAL MODELS

The PDMS elastomer is modelled as a near-incompressible Neo-Hookean material, a constitutive model which describes the behaviour of rubber-like materials undergoing large deformations [5]. The large deformation membrane behaviour theory is also called plane-strain theory, which must be satisfied with three assumptions of rectangle shape: a) the length of the membrane \( l_m \) is at least two times larger than the width \( w_m \); b) the width \( w_m \) of the membrane is ten times the membrane thickness \( t_m \) [21]. If all the assumptions are satisfied, then a) the variation in deflection along the length can be ignored, and b) the centreline stretch of the membrane dominates bending. We resort to the finite element method (FEM) to represent the geometry of the rectangle elastic thin membrane and to solve the equations governing its deformation.

Maximum displacement of deflection is a function of the pressure over the membrane, which is can be calculated by [21]

\[
\delta_{\text{max}} = \frac{w_m}{4} \left( \frac{3AP w_m}{E_{\text{lm}}} \right)^{\frac{1}{3}}
\]

(4)
Where, \( \delta_{\text{max}} \) is the maximum deformation of the membrane; \( \Delta P = P_a - P_l \) is the applied pressure to the membrane, \( P_a \) is the pneumatic pressure in the gas microchannel, \( P_l \) is the other forces to the membrane, which direction contrary to gas pressure, such as the hydraulic pressure in the liquid microchannel of three layers pneumatic microfluidics; \( \bar{E} = E/(1-\nu)^2 \) is the plane-strain modulus, \( E \) is the Young’s modulus, \( \nu \) is the Poisson ratio for the material.

4. MATERIALS AND FABRICATION

In this research work, the microvalve body is applied in elastic PDMS material, due to the key merits of PDMS materials, such as transparency, remarkable mechanical property, simple structure bonding process and low production cost [14]. Furthermore, the PDMS material is employed in the body of the microvalve, which permits easy integration into microfluidic system, because the PDMS material is also engaged in various microfluidic applications.

Figure 2. Protocol for manufacturing the electromagnetic microvalve. (a) print a photomask; (b) specific master fabrication process using dry film photoresist on a stainless steel wafer; (c) craft steps of the PDMS microvalve body; (d) encapsulated the microvalve with the electromagnetic actuator.

The packaging process using PDMS materials of the proposed microvalve is different from tradition methods. The valve body portion and the electromagnetic actuator were fabricated separately, it is shown in Figure 2. Design the structure by CAD software and print it onto a film using a high resolution inkjet printer (Stylus Photo R2000, Seiko Epson Corporation, Suwa, Japan) as a photomask, it is shown in Figure 2 (a). Figure 2 (b) illustrates the detailed fabrication process employed for the master. The master fabrication process employs the permanent dry-film (FF-9050S, Changchun Group Company, Taiwan, China) with 50\( \mu \)m thickness instead of SU-8 liquid photoresist for cheaper, simpler and save-time. The fabrication starts with the silicon master production using dry film photoresist, which is the dry-film made by negative thick photoresist is pressed onto a stainless steel wafer to form a layer with a thickness of 50\( \mu \)m using a sealing machine (LR-230, Yatai office facilities Company, Wenzhou, China) at room temperature. Then, the pressed
master is cured for 10 minutes at 80°C in an oven. Take it out and cool it down, when the surface temperature of the pressed wafer drops to room temperature, pressed another layer with same thickness of 50µm dry-film photoresist on the master. Next, let the steel wafer with 100µm dry film photoresist pressed five times using the sealing machine with 100°C in order to make the dry film photoresist fixed on the stainless steel wafer. Let photomask cover the wafer (ink face down) and put them to exposure three minutes away from an UV lamp (UVC-25W, Shenzhen new modern technology Company, Shenzhen, China) 10cm. The exposure time under UV light depends on the power of UV and thickness of dry-film. Less exposure will lead to a fuzzy development and difficult to rinse out unexposure part in the next step, and overexposure will bring the exposure part harden and poor adhesion. Exposure 110 seconds is suitable for two layers dry-film with 100µm thickness using the same UV lamp. To enough image development, soak the wafer into Na₂CO₃ solution (1:60~70) for 5 minutes. The wafer is rinsed for several times using isopropanol and DI water, followed by a blow-dry process using compressed N₂. A post-exposure bake at 80°C for 5 min is performed. Figure 2 (c) starts with the PDMS valve membrane, which is spin-coated on the planar side of a silicon wafer, spin-coating 100µm thickness PDMS from a prepolymer solution (Sylgard 184, base/curing agent 15:1; Dow Corning Corporation, Midland, MI). To protect the mold, the surface of the silicon/photoresist master is treated with chlorotrimthylsilane (A13651, 25ml, (CH₃)₃ClSi, Alfa Aesar Corporation, Ward Hill, MA). The purpose of silanized treatment is also to prevent excessive air bubbles when spin-coating. Then, the filled mold is cured for 3h at 65°C in an oven. The air channel and its PDMS slab were made by replica casting of prepolymer solution (base/curing agent 5:1) on a silicon master, which is then cured for 30min at 80°C in an oven. The PDMS slab and membrane is treated by a 50W plasma for 2 min and then sealed irreversibly to form a valve body. A voltage of DC24V is applied initially to the electromagnetic actuator during bonding onto the valve body, as shown in Figure 2 (d). Once bonded, this assembly procedure ensures that the valve head is pressed downward for a fully closed microvalve.

The total size of the encapsulated actuator is 20.5×12×9.5mm, as shown in Figure 1 (d). It is much smaller than other actuators used in microfluidic fields, such as piezoelectric actuator (deformation of 90µm, need a length size is 90mm) and linear steering engine (25×22×20mm, miniature GS-1502, size after encapsulated into the adaptive actuator). In order to facilitate the measurement, the designed microchannel length is 30mm, which easily connects with external gas systems. The thickness of the valve diaphragm is 100µm, typical cross-sectional dimensions are approximately 100µm high and 500µm wide for the microchannel in the bottom PDMS slab, which makes the valve intersections approximately 1000×500×100µm. The picture of the encapsulated the microvalve with the electromagnetic actuator is shown in Figure 1 (e). FESTO solenoid valve array including eight 3-way valves for off-board pneumatic control of pneumatic microfluidic chips; the total size is about 200×100×80mm, and the price is about 490$. (II) Designed valve array including eight three-way microvalves; the total size is 110×45×30 mm, and the cost is less 100$.

![Diagram of the microvalve test set-up.](image-url)
5. EXPERIMENTAL RESULTS

Experimental study of the whole system mainly includes two parts, the electromagnetic microvalve character test experiment; the second part is the designed microvalve will be tested for its use in a microfluidic system, it is shown in Figure 3.

5.1. Response time of the microvalve

To measure the response time of the microvalve, a micro displacement sensor (LK-G5000, KEYENCE Company, Osaka, Japan) was used to observe the displacement so that the response time of opening or closing the actuator can be seen, as shown in Figure 4. When there is no flow, the opening microvalve response time is about 6.4ms, while the closing microvalve response time is about 4.0ms.

5.2. Leaking rate of the closed microvalve

The encapsulated microvalve can sustain N\textsubscript{2} flow of an applied pressure of 280kPa without bursting open. This fully satisfies the air pressure needs of a PMC. Leak testing of a microvalve using a small flow sensor showed an extremely low leak rate of 1.56×10\textsuperscript{-5}m\textsuperscript{3}/h at an inlet pressure of 200KPa. While the microvalve is closed, the leaking rate is determined from the inlet pressure. Figure 5 (a) shows the leak rates of the microvalve for various inlet pressures. The return spring and the thin membrane, which work like a gasket in a macroscopic valve and use rubber washers as valve seats, are responsible for the high seating pressure and the extremely low leak rates.

5.3. Flow test of the microvalve

If the designed microvalve is fully open, the valve hole is the air channel. From Figure 5 (b), we can see that flow rate and differential pressure are proportional. When the differential pressure is 120KPa, the flow rate is 1.4×10\textsuperscript{-2}m\textsuperscript{3}/h. For varying the flow rate delivered by the microvalve, the microvalve can be actuated in PWM mode. The PWM mode resulted in a linear relation between duty cycle and flow rate. The dependence of flow rate on duty cycle is plotted in Figure 5 (c). A duty cycle of 0% means that the valve is fully closed and a duty cycle of 100% means that it is fully open. Figure 5 (d) presents the measured flow rates of N\textsubscript{2} obtained for valve-operating frequency from 0 to 50Hz using a square wave actuation voltage with the same voltage amplitude. In the previous period, measurements made at a constant duty cycle and constant inlet pressure
show that the flow rate drops with higher frequency operation because of the slow response time of the flow meter at high frequency.

Figure 5 Leak rate of the microvalve under the on/off mode (a); N₂ flow rate of a single open microvalve (b); flow rates versus duty cycle of a microvalve at differential pressures (c); measured N₂ flow rate as a function of actuation frequency at differential pressures (d).

5.4. Integration with a microfluidic chip

Figure 6 Diagram of the electromagnetic microvalve connected to the microfluidic valve (a); experiment results of the microfluidic membrane valve displacement versus actuated time under different pressures with dimension of 200μm×500μm×63μm (b).
The electromagnetic microvalve is designed and tested for use in a microfluidic system. The system is simplified for testing purposes. The electromagnetic microvalve is connected with the microfluidic system by connecting the electromagnetic microvalve outlet to the control microchannel of the microfluidic system. When the electromagnetic microvalve is on, the air flows through the electromagnetic microvalve, and the pressure drives the membrane of the microfluidic membrane valve upwards. When the electromagnetic valve was closed, the air pressure stopped, and the fluidic pressure opened the microfluidic membrane valve.

The value of air pressure decides the valve opening of the microfluidic membrane valve. The response times of the membrane displacement of the microfluidic membrane valve using N\textsubscript{2} flow are measured under different air pressures in control microchannel. And the results confirm it is sufficient for the intended application. Our method provides a simple, cheap and small microvalve avoids the bulky and expensive external pressure control solenoid manifold and it allows it to be easily integrated into portable and disposable devices.

6. CONCLUSIONS

In conclusion, an electromagnetic microvalve with a simple structure and the appropriate fabrication process and characterization results is presented. This microvalve is fast-response and leak-tight for precise gas flow rate control of the PMC. The simplicity permits the design of a valve body with small dimensions and the integration of a valve array which can act as a module outside of the PMC. The novel idea of eliminating the external solenoid valves, if executed, can help us realize the real meaning of mLSI for the whole PMC system.

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SIMPLIFIED FLUID TRANSMISSION LINE MODEL
FOR PNEUMATIC CONTROL APPLICATIONS

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ABSTRACT

The neglect of transmission lines in pneumatic controller synthesis limits the possible control performance or the length of the connecting tubes. In literature, complex and accurate models have been derived for the description of flow phenomena in tubes, but low-order models for controller design and stability analysis are rare. This paper proposes a second-order modal approximation of the one-dimensional linear resistance compressible model and includes non-linear tube friction. The model was validated in time domain as well as in frequency domain. An application for the model in pressure control was shown and compared to the state of the art.

KEYWORDS: Modeling, fluid transmission line, pneumatics, model reduction, frequency response validation

1. INTRODUCTION

In pneumatics, the influence of tubes on pressure and mass flow is a well known research field. Numerous models with various levels of detail have been derived and experimentally validated. But when it comes down to controller synthesis and stability analysis of closed-loop control (e.g. pressure control or servopneumatic positioning) the influence of transmission lines is mainly neglected or oversimplified.

An overview of different fluid transmission line models is given in [1]. The presented models vary in the assumptions made and therefore in complexity of their analytical solutions. These models give good insight into different flow phenomena but are mainly not applicable in simulation of fluidic systems. For this reason, many different approximation methods exist for the fluid transmission line models in literature. [2] reviewed four different methods. Two of them are the finite difference approximation and the lumping-by-length method. Both are afflicted with frequency errors that increase with the number of nodes respectively lumps. Further methods, i.e. the method of characteristics and a modal approximation, yielded considerably better results. The modal approximation is superior in terms of stability and ease of use.

Similar approaches were investigated by different authors. [3] chose a decoupled representation (stiffness matrix) of the line model for a modal approximation. Compared to Hullender’s approximation [4] of the transfer matrix representation, they found their method to yield several simplifications while maintaining the same accuracy. The presented model is of order 17. An approximation of significantly lower order is given in [5]. The
combination of a time delay with transfer function elements for damping achieved good results that are comparable with the ones in [3]. The stiffness matrix representation was the basis in [6], too. The approximation is carried out with a variational Ritz method, reproduces the modes accurately yet leads to steady-state errors. The minimum system order is two but increases with the correction of the steady-state error. An expansion to non-linear friction was shown.

These research topics cover mainly the general behavior of pneumatic or hydraulic transmission lines for time domain simulations. [7, 8] derived an approximation of the line model with special focus on use in controller design. It was modeled as a time delay including an attenuation of the mass flow's amplitude. Laminar as well as turbulent flow was considered. It was shown that the consideration of the tube model in controller design improved the performance at the expense of high computational effort.

The goal of this paper is to provide a low-order pneumatic transmission line model that reproduces the dynamics of pressure and mass flow in time as well as in frequency domain. The model’s structure should enable stability analysis and time domain controller design for pneumatic applications, i.e. servopneumatic positioning or pressure regulation. The considered tube lengths vary from one to five meters and are typical for this application field. The basis for the simplified transmission line model is the one-dimensional linear resistance compressible model [1]. Although this model is of lesser overall accuracy than models with frequency dependent friction, it fits the purpose of a low-order approximation. Subsequently, the tube dynamics are extended by a non-linear friction model.

2. FLUID TRANSMISSION LINE MODEL

The one-dimensional linear resistance compressible model is given by the first-order partial differential equation (PDE) system

\[ \frac{\partial p}{\partial x} = -\frac{1}{A} \frac{\partial \dot{m}}{\partial t} - R \dot{m}, \]

\[ \frac{\partial \dot{m}}{\partial x} = -\frac{A}{c^2} \frac{\partial p}{\partial t}, \]

where \( p = p(x, t) \) and \( \dot{m} = \dot{m}(x, t) \) are pressure and mass flow, dependent on time \( t \) and the paraxial coordinate \( x \), \( A \) the tube’s inner cross-sectional area and \( c \) the sonic speed. The coordinate \( x \in [0, l] \) is bounded by the length \( l \) of the tube. These variables are designated in the schematic drawing of the tube in Figure 1. The line resistance \( R \) is constant hence flow is assumed to be laminar. This assumption is necessary for the existence of the algebraic solution that, evaluated at the border conditions

\[ p_A = p(0, t), \]

\[ p_E = p(l, t), \]

\[ \dot{m}_A = \dot{m}(0, t), \]

\[ \dot{m}_E = \dot{m}(l, t), \]
is stated as follows

\[ p_E = \frac{1}{\cosh \left( \Gamma(s) l \right)} \left( p_h - \dot{m}_E Z(s) \sinh \left( \Gamma(s) l \right) \right), \quad (3a) \]

\[ \dot{m}_h = \frac{1}{\cosh \left( \Gamma(s) l \right)} \left( \dot{m}_E + p_A \frac{1}{Z(s)} \sinh \left( \Gamma(s) l \right) \right). \quad (3b) \]

The mixed boundary conditions are needed for simulation in order to obtain a causal system behavior. The dimensionless propagation function \( \Gamma(s) \) and the characteristic impedance function \( Z(s) \) characterize the connection between pressure and mass flow in the tube and are given by

\[ \Gamma(s) = \frac{1}{c} \sqrt{(AR + s)s}, \quad (4a) \]

\[ Z(s) = \frac{c}{A} \sqrt{AR + s}. \quad (4b) \]

The algebraic solution is a transcendent two-by-two transfer matrix with infinitely many poles and eigenfrequencies. From a control theory point of view the location of the poles is of great significance. By solving the equation

\[ \cosh \left( \Gamma(s) l \right) = 0, \quad (5) \]

one can calculate the transfer function's poles respectively the natural frequency and damping of the resonance \( i \) by

\[ \omega_{0,i} = \frac{\pi (2i + 1)}{2l}, \quad (6a) \]

\[ D_i = \frac{Rl}{c \pi (2i + 1)}. \quad (6b) \]

In the following, the exact solution (3) is taken as a reference for all approximations.

2.1. Modal approximation

While the transmission line model (3) gives a good insight into the fluid dynamics in the tube, it is not suitable for time domain stability analysis and controller synthesis. Such tasks require a low-order system that reproduces the general behavior and matches the dominant eigenvalues of the exact solution.

The modal approximation was detailed in [9] for the inviscid line model. It is based on a truncated Taylor series (TTS) of the hyperbolic functions. The result did not produce a good match for the first resonance frequency. In the following, the three different approximations in Table 1 of the hyperbolic functions are applied and analyzed. Approximation 1 is the truncated Taylor series. It is added for completion and used for comparison with the other methods. The second approximation uses the truncated product series (TPS) for the hyperbolic cosine

<table>
<thead>
<tr>
<th>Function</th>
<th>Product series</th>
<th>Taylor series</th>
<th>Appr. 1 (TTS)</th>
<th>Appr. 2 (TPS)</th>
<th>Appr. 3 (LTTS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \cosh(x) )</td>
<td>( \prod_{k=1}^{\infty} \left( 1 + \frac{4x^2}{(2k - 1)^2 \pi^2} \right) )</td>
<td>( \sum_{k=0}^{\infty} \frac{x^{2k}}{(2k)!} )</td>
<td>( 1 + \frac{x^2}{2} )</td>
<td>( 1 + \frac{4}{\pi^2}x^2 )</td>
<td>1</td>
</tr>
<tr>
<td>( \sinh(x) )</td>
<td>( x \prod_{k=1}^{\infty} \left( 1 + \frac{x^2}{(k \pi)^2} \right) )</td>
<td>( \sum_{k=0}^{\infty} \frac{x^{2k+1}}{(2k + 1)!} )</td>
<td>( x )</td>
<td>( x )</td>
<td>( x )</td>
</tr>
</tbody>
</table>
Figure 2. Magnitude plot of the exact solution, the TTS model and the TPS model. Tube of length 1 m and inner diameter 4 mm.

and the Taylor series for the hyperbolic sine. It results in a transfer matrix

\[
\begin{pmatrix}
\dot{m}_E \\
p_A
\end{pmatrix}
= \begin{pmatrix}
\frac{\pi^2 c^2}{4 l^2 2 s^2 + 4 AR l^2 s + \pi^2 c^2} & -\frac{\pi^2 A l s}{4 l^2 2 s^2 + 4 AR l^2 s + \pi^2 c^2} \\
\frac{\pi^2 c^2 l (AR + s)}{A (4 l^2 2 s^2 + 4 AR l^2 s + \pi^2 c^2)} & \frac{\pi^2 c^2}{4 l^2 2 s^2 + 4 AR l^2 s + \pi^2 c^2}
\end{pmatrix}
\begin{pmatrix}
\dot{m}_A \\
p_E
\end{pmatrix},
\] (7)

consisting of four second-order systems. The natural frequency and damping of the poles

\[
\omega_0 = \frac{\pi c}{2l},
\]

(8a)

\[
D = \frac{RI}{c}\pi
\]

(8b)

match those of the exact solution’s first resonance.

Figure 2 shows the frequency response of the exact solution, the TPS model (7) and the TTS model using the truncated Taylor series for both hyperbolic functions. The accuracy of the model was increased with the product series expansion. Only small deviations concerning the amplitude of the resonance frequency are left. In contrast to the TTS approximation in [9], the TPS method results in a better fit to the first resonance frequency. The algebraic controllability and observability analysis shows that the minimal state space realization of (7) is of order four.

The third approximation (LTTS) substitutes the hyperbolic cosine by the constant 1 and is therefore the lowest-order approximation. The application to (3) and transformation into time domain yields the second-order system

\[
\frac{d\dot{m}_A}{dt} = \frac{A}{T} (p_A - p_E - RI \dot{m}_A),
\]

(9a)

\[
\frac{dp_E}{dt} = \frac{c^2}{A l^3} (\dot{m}_A - \dot{m}_E).
\]

(9b)

The result of this approximation is closely related to the finite volume method [10] which divides the transmission line alongside the x coordinate into N parts. Each part describes a finite volume with pressure and mass flow dynamics. With increasing N, the accuracy of the solution increases as well as the order of the resulting system. For N < 5 the error of the first resonance frequency is greater than 10%. Setting the number of finite volumes to N = 1, the finite volume method produces the same model as the introduced LTTS method.

In contrast to the previous transfer function, the boundary conditions change from \(\dot{m}_A\) and \(p_E\) to \(\dot{m}_E\) and \(p_A\). To
get the same boundary conditions as in (7), model (3) needs to be solved for $\dot{m}_A$ and $p_E$ and approximated, achieving the corrected LTTS (cLTTS) model

$$\frac{d\dot{m}_E}{dt} = \frac{A}{T} \left( p_A - p_E - R \dot{m}_E \right),$$

(10a)

$$\frac{dp_A}{dt} = \frac{k c^2}{Al} \left( \dot{m}_A - \dot{m}_E \right).$$

(10b)

Since the low-order approximation leads to an error concerning the resonance frequency, the factor $k$ was inserted supplementary to correct that error. The natural frequency and damping of the eigenvalues are given in dependence of $k$ as

$$\omega_0 = \sqrt{k} \frac{c}{l},$$

(11a)

$$D = \frac{RL}{2c\sqrt{k}}.$$  

(11b)

By comparing Equations (6) and (11), one can calculate

$$k = \frac{\pi^2}{4},$$

(12)

such that the eigenvalues of the approximation fit the exact solution. This factor already showed up in the truncated product series expansion and leads to the improved matching of the cLTTS model. The transformation of (10) into frequency domain makes the difference to the TPS model clear. The transfer matrix

$$\begin{pmatrix}
\dot{m}_E \\
p_A
\end{pmatrix} = \begin{pmatrix}
\frac{\pi^2 c^2}{4l^2 s^2 + 4AR/R^2 s + \pi^2 c^2} & \frac{4A}{4l^2 s^2 + 4AR/R^2 s + \pi^2 c^2} \\
\frac{\pi^2 c^2 / (AR + s)}{A (4l^2 s^2 + 4AR/R^2 s + \pi^2 c^2)} & \frac{\pi^2 c^2}{4l^2 s^2 + 4AR/R^2 s + \pi^2 c^2}
\end{pmatrix} \begin{pmatrix}
\dot{m}_A \\
p_E
\end{pmatrix}$$

(13)

equals (7) except for the gain of the upper right element.

Figure 3 shows the magnitude plot of the cLTTS model (13) and LTTS model against the exact solution’s magnitude plot (3). The uncorrected solution differs in terms of resonance frequency. The correction leads to a far better approximation in three transfer functions and only results in a magnitude error of the transfer function from pressure to the mass flow at the end of the tube. This transfer function will later be eliminated by the coupling of the tube with a terminal volume.
In Figure 4, the phase plot of the exact solution, the LTTS and the cLTTS model is shown. The corrected approximation fits the phase response of the exact solution precisely up to the first resonance frequency. For higher frequencies, the phase response of the approximations stays on the same level while the exact solutions phase response oscillates or decreases respectively. In contrast to the TPS model, the cLTTS model can be realized with only two first-order differential equations. It is remarkable that the second-order model has the same transfer function structure as the fourth-order model used in literature.

2.2. Non-linear friction modeling

Until now, both models only consider friction in form of linear damping that depends on mass flow. This is valid for laminar flow. In pneumatic applications like pressure control or servopneumatics, high flow rates over a short period of time are more common. For this reason, the model must be valid for turbulent flow, too. There are two common methods that describe the pressure drop in tubes, considering laminar as well as turbulent flow — the Hagen-Poiseuille respectively the Darcy-Weisbach equation (\( \lambda \) method) and the pneumatic related \( \{C, b\} \) method. The \( \lambda \) method was applied in [10] for a high-order model of a pneumatic transmission line. Dependent on the Reynolds number, different linear and non-linear equations must be evaluated for the calculation of the pressure drop.

A more compact method for obtaining the pressure drop is the modeling of the tube as a pneumatic nozzle. The mass flow

\[
\dot{m} = C \rho \psi (q, b) p_{\text{high}}
\]

through the nozzle depends on the sonic conductance \( C \), the critical pressure ratio \( b \), the pressures \( p_{\text{low}} \) and \( p_{\text{high}} \) respectively their ratio \( q = p_{\text{low}}/p_{\text{high}} \). An empirical formula for sonic conductance and critical pressure ratio dependent on tube geometry is given in [11] as

\[
C = \frac{0.029 d^2}{\sqrt{\frac{l}{d^{5/4}} + 510}},
\]

\[
b = \frac{474 C}{d^2}.
\]

To calculate the pressure drop via the \( \{C, b\} \) method, Equation (14) needs to be solved for the pressure. Since
the flow function

\[ \psi(q, b) = \begin{cases} 1 & \text{if } q < b \\ \sqrt{1 - \left(\frac{q - b}{1 - b}\right)^2} & \text{if } q \geq b \end{cases} \]  

(16)

is not invertible for pressure ratios smaller than the critical pressure ratio, a polynomial approximation

\[ \tilde{\psi}(p_{\text{high}}, p_{\text{low}}) = a_{\psi} (p_{\text{high}} - p_{\text{low}}) \]

(17)

is used for the inversion. The factors \( a_{\psi} \) and \( c_{\psi} \) must be chosen appropriately. The resulting pressure drop is expressed by

\[ \Delta p = \Phi(\dot{m}_E, p_E). \]  

(18)

In contrast to the \( \lambda \) method which depends solely on the mass flow, the \( \{C, b\} \) method depends on mass flow and pressure. Combining the cLTTS model (10) and the pressure drop (18) adds up to the non-linear second-order model

\[ \frac{d\dot{m}_E}{dt} = A \left( p_A - p_E - \Phi(\dot{m}_E, p_E) \right), \]

(19a)

\[ \frac{dp_A}{dt} = \frac{\pi^2 c^2}{4At} (\dot{m}_A - \dot{m}_E). \]

(19b)

2.3. Line model with terminal volume

For experimental validation or pneumatic applications, the tube model cannot be considered without the influence of a terminal volume. Coupling mass flow and pressure at the end of the tube with a simple model of a pneumatic volume

\[ \frac{dp_E}{dt} = \frac{n R_a T_0}{V_V} \dot{m}_E, \]

(20)

leads to the three transfer functions

\[ G_{p_A}(s) = P_A(s)/\dot{M}_A(s), \]

(21a)

\[ G_{p_E}(s) = P_E(s)/\dot{M}_A(s), \]

(21b)

\[ G_{\dot{m}_A}(s) = \dot{M}_E(s)/\dot{M}_A(s). \]

(21c)

Figure 5 shows the frequency response of \( G_{p_A}(s) \) with different terminal volume sizes. The resonance frequencies increase with decreasing volume size. In comparison to Figure 3, an additional antiresonance can be seen. Since the same characteristic can be observed in measurements as well as in the model, this effect is not a model artifact but real system behavior. The effect gets significant as soon as the terminal volume ap-

![Figure 5](image-url)  

*Figure 5. Magnitude plot of \( G_{p_A}(s) \) of a tube (inner volume \( V_S \approx 0.07 \text{ L} \)) with a big (\( V_V \approx 0.75 \text{ L} \)) and a small (\( V_V \approx 0.1 \text{ L} \)) terminal volume. Simulation on the left and measurement on the right side.*
proaches the tube’s inner volume. With an infinitely big volume, the resonance frequencies converge towards the resonance frequencies of the tube model without volume.

3. EXPERIMENTAL VALIDATION OF THE TUBE MODEL

The experimental validation is based on time and frequency domain responses. For the frequency response identification the valve slide is excited by band limited white noise with a maximum frequency of 100 Hz. The amplitude of the excitation is chosen low enough for laminar flow conditions to apply and non-linear friction effects to be neglected. Time domain is validated with a sinusoidal excitation of the valve slide. Those measurements are used to identify the non-linear friction effects resulting in a pressure drop. The validated transfer functions in the following section are $G_{pA}(s)$ from valve mass flow $\dot{m}_A$ to valve pressure $p_A$ and $G_{pE}(s)$ from valve mass flow $\dot{m}_A$ to terminal volume pressure $p_E$. The transfer function of the mass flows $\dot{m}_A$ and $\dot{m}_E$ is neglected since the reactionless determination of the mass flow $\dot{m}_E$ is not possible in this test setup. In frequency domain, pressure and mass flow are normalized to $1 \times 10^{-5}$ Pa and $1 \times 10^3$ kg/s.

The test rig, shown in Figure 6, consists of a proportional valve, the pneumatic tube and a terminal volume. The mass flow $\dot{m}_A$ through the valve is calculated with Equations (14) and (16) with the measurement of the valve slide $h$, the supply pressure $p_1$, the exhaust pressure $p_0$ and the pressure $p_A$. In contrast to direct measurement of the mass flow with a sensor, the calculated quantity has the higher dynamics that are needed for frequency response determination.

Figure 7 shows the model’s and the measured frequency responses of $G_{pA}(s)$ and $G_{pE}(s)$. The tube has a length of 3 m and an inner diameter of 5.7 mm. The terminal volume’s size is 0.75 L. The model matches the resonance frequency of about 25 Hz and the frequency characteristic up to 50 Hz sufficiently. The position

![Figure 6. Schematic representation of the test rig.](image)

![Figure 7. Frequency responses of $G_{pA}(s)$ on the left and $G_{pE}(s)$ on the right. Tube of length 3 m and inner diameter 5.7 mm with a terminal volume of 0.75 L.](image)
Figure 8. Frequency responses of $G_P(s)$ of a tube of length 1 m on the left and 5 m on the right. The tube’s inner diameter is 5.7 mm and the terminal volume size 0.75 L.

Figure 9. Frequency response of $G_P(s)$ of a tube of 3 m length and 5.7 mm inner diameter. The terminal volume size is 0.1 L on the left and 0.01 L on the right.

of the first antiresonance of $G_P(s)$ as well as the steady state amplification of $G_P(s)$ depend mainly on the terminal volume. The erroneous frequency of the antiresonance cannot be corrected without changing the low-frequency amplification of $G_P(s)$. Aside from small differences, the second-order cLTTS model and the fourth-order TPS model show the same behavior. The same is valid for the frequency response in Figure 8. Here, the measurement of two tubes with the lengths 1 m and 5 m is shown. Again, both approximations provide comparable results.

Figure 9 shows the frequency response of the same tube with two different terminal volume sizes, respectively 0.1 L and 0.01 L. The size of the first terminal volume is similar to the tube’s inner volume of approximately 0.07 L. The second terminal volume is much smaller. Such ratios are an exception in pneumatic applications but they illustrate the difference between the TPS and the cLTTS model. The amplitude response of the higher-order TPS model forms two resonances: one that depends only on pipe geometry and one that also includes the terminal volume’s parameters. With decreasing volume sizes, both resonance frequencies diverge from each other (Figure 9 right side). The amplitude for low frequencies matches the measurement.

While the cLTTS model replicates the frequency response on the left reasonably well, the antiresonance frequency deviates significantly on the right. Compared to the fourth-order TPS model, the resonance frequency provides a better fit. The error for low frequencies comes from the resonance correction (12). The correction resulted in an amplitude error of the transfer function from $p_E$ to $\dot{m}_E$ that was insignificant for large terminal volume sizes. The exact solution of the transmission line model however indicates that amplitude and resonance
errors originate in the approximation. Higher-order approximations with the finite volume method will result in a better replication of the frequency characteristic for very small terminal volumes.

The time domain validation of the non-linear cLTTS model is displayed in Figure 10. The valve slide is excited with a 2 Hz sinusoid of decreasing amplitude. The measurement clearly shows the necessity of considering tube dynamics and friction. With a tube of 3 m length and 5.7 mm inner diameter, the difference in pressure at valve and terminal volume (0.75 L) is up to 400 kPa. The detail view on the right side shows that the tube model reproduces the large-signal behavior as well as small oscillations while the mass flow direction changes. The measured and simulated mass flows are depicted in the middle part. The mass flow $\dot{m}_{E,s}$ at the end of the tube is lagged and lower in amplitude than the mass flow at the beginning. Additional oscillations, similar to those of $p_A$, can be seen. The lower part of Figure 10 shows a comparison between the linear and the non-linear friction model. This comparison stresses the necessity of the non-linear extension, which attenuates the pressure oscillations to a large extent and results in a pressure drop of up to $2 \times 10^5$ Pa. In this example, laminar flow conditions only apply at mass flows lower than approximately 2.2 kg/s. In time domain, the neglect of turbulent flow conditions has a much larger impact than the choice of model (e.g. linear resistance, viscous compressibility, thermal effects) or the approximation method.

The validation in frequency domain showed a good agreement of the TPS and the cLTTS model with measurements. Tube lengths of 1 m, 3 m and 5 m, as well as different terminal volume sizes were considered. Only very small terminal volume sizes revealed the difference between the two models. The observed model deviations are not a big limitation, since such tube volume to terminal volume ratios are rarely found in pneumatic applications. The extension with the non-linear friction model made the tube model also valid for turbulent flow conditions and resulted in an excellent match in small- and large-signal behavior.

\begin{center}
\includegraphics[width=\textwidth]{figure10.png}
\end{center}

\textit{Figure 10. Validation of the pressures and mass flows at beginning and end of a tube of 3 m length and 5.7 mm inner diameter. Index m for measured and s for simulated.}
4. APPLICATION IN PRESSURE CONTROL

The standard model for pressure control applications does not include tube dynamics and hence the pressures $p_A$ and $p_E$ as well as the mass flows $\dot{m}_A$ and $\dot{m}_E$ are considered equal. The model is given by

$$\frac{P_E(s)}{M_v(s)} = \frac{1}{s} \frac{n R_a T_0}{V_v} \frac{1}{T_V s + 1} = \frac{k_V}{s(T_V s + 1)}.$$  \hspace{1cm} (22)

It consists of a pneumatic volume (20) and valve dynamics, modeled as a first-order lag with time constant $T_V$. The input is the mass flow $\dot{m}_V$ in front of the valve and the output the pressure $p_E$ in the volume. In reality the mass flow depends on the valve pressures and the valve slide position and is not the actual control input. This dependence can be inverted and the mass flow becomes a virtual input of the plant. With a proportional controller, that regulates the pressure in the volume, the closed-loop’s transfer function is

$$G_{cl} = \frac{k_p k_V T_V^{-1}}{s^2 + T_V^{-1} s + k_p k_V T_V^{-1}}.$$  \hspace{1cm} (23)

and the controller gain can be expressed in dependence of a desired damping factor $D$ by

$$k_p = \frac{1}{4 D^2 k_V T_V}.$$  \hspace{1cm} (24)

The derived control law is now identically applied to the nominal model (22) and to a model that is extended by the linear tube dynamics (10). The tube has a length of 2 m and an inner diameter of 5.7 mm. Figure 11 shows on the left the desired pressure trajectory $p_{E,d}$ and the time responses of the control loop with and without tube dynamics. The pressure transient of the nominal model shows the expected behavior of a second-order lag with a damping factor of $D = 0.85$. According to the Nyquist diagram on the right, the closed loop has a large phase margin and is robust towards amplitude errors. The influence of the tube dynamics in the control loop can directly be seen in the initial lag at 0.1 s and the decaying oscillations around the desired pressure level. The Nyquist diagram reveals that the tube dynamics drive the control loop close to the stability limit.

Aside from considering the tube model in closed-loop analysis, it will be used for design, too. To increase robustness, the controller gain is lowered to meet certain criteria for phase and amplitude margin. The result is slower closed-loop dynamics, as can be seen in Figure 12 on the left. To improve the tracking behavior, the feedforward control \[ \text{[12]} \]

$$P_{E,ff}(s) = \left(1 + \frac{k_1 s}{T_d s + 1}\right) P_{E,d}(s)$$  \hspace{1cm} (25)

is added. The control law includes the desired pressure in the volume as well as its time derivative, that is attained by filtering and differentiation. The factor $k_1$ contains parameters of the plant including tube dynamics and the controller. The time response in Figure 12 now shows similar dynamics as in the nominal case. The oscillations are only slightly suppressed, but the robustness of the control loop is significantly higher (see Nyquist diagram in Figure 12 on the right). To avoid the pressure oscillations completely, the control law can
Figure 12. Time response and Nyquist diagram of the pressure control in a constant volume with consideration of the linear tube model in controller design.

be extended by a derivative term or replaced by a state space controller, since the simple tube model is also available for controller design and dimensioning. The presented model could then be applied as an observer to reconstruct the system’s state variables.

5. CONCLUSION

The proposed pneumatic tube model is an approximation of the one-dimensional linear resistance compressible transmission line model of dynamic order two with special focus on the dominant eigenvalues. The augmentation with a pressure drop, based on the \( \{C,b\} \) method, accounts for essential non-linear friction effects in the tube. The model shows a good agreement with measurements in time as well as in frequency domain and has a lower order than comparable line models in literature.

The advantage of the model was shown for a pressure control application. By considering the tube dynamics in controller analysis and design, often neglected robustness issues were revealed and the control performance was increased.

REFERENCES


OPTIMAL PAIRED IN-LINE BLADDER-STYLE SUPPRESSORS FOR BROADBAND NOISE CONTROL

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ABSTRACT

The dimensions and charge pressures of in-line bladder-style suppressor pairs were optimized to find the configuration with maximum broadband noise control for a range of operating pressures. The dimensions under consideration for both suppressors within a pair are the inner radius of the bladder, the inner radius of the outer shell and the length of the bladder, which are the dimensions that determine the pressurized gas volume within the device. The separation distance between the suppressors was also considered. Prior work optimized the charge pressure of commercially available devices in several configurations: a single suppressor, two similarly-sized suppressors in series, and two dissimilarly-sized suppressors in series. Results of example optimizations show the optimal suppressor pairing does have a large pressurized gas volume; however, the volume is not the largest volume attainable within the bounds on the design variables. The respective dimensions for each suppressor within an optimal pair are dissimilar between the pair. Dissimilar dimensions for each suppressor within the pair yield different frequency-dependent performance for each suppressor; therefore, there is not a shared deficiency that would lead to poor performance in a given frequency range as could be the case with identical devices. This result underscores the importance of optimizing the dimensions of a suppressor pair instead of simply employing two of the largest devices available. While the dimensions of the suppressors have a substantial influence on the noise control effectiveness, the separation distance between the suppressors is shown to have little consequence on the effectiveness of the pair.

Keywords: Noise control, bladder-style suppressor, optimization

1. INTRODUCTION

Fluid-borne noise, the varying pressure component superimposed onto the mean static pressure of a hydraulic system, creates problems within a hydraulic circuit: both within the working fluid and in the surrounding environment. The pressure pulses increase stress on sealing surfaces, and in some instances, exceed the rated limit of the seal causing a leak. The cyclical nature of the fluid-borne noise induces fatigue cycles on components shortening their lifetimes. The varying pressure induces vibrations that couple with structural components of a hydraulic circuit to create structure-borne vibrations; the vibrations make precise end-effector control difficult as well as the cab environment uncomfortable for a machine operator. In addition, some of the fluid-borne noise breaks out into the surrounding airborne environment becoming perceptible noise. Noise levels may disrupt communication on a worksite. When the noise reaches
hazardous levels, hearing protection for workers becomes a necessity. In a recent customer survey, New Holland Agriculture determined a quiet cab was the third most important feature for prospective buyers behind adequate horsepower and good fuel economy [1]. Effective reduction of the fluid-borne noise will serve to reduce the severity or eliminate the associated problems, including a reduction in cab noise. A current, commercially-available noise control device, schematically depicted in Figure 1, is a bladder-style suppressor [2-4]. Bladder-style suppressors treat noise by creating an acoustic impedance mismatch at both device ports. The mismatch is created by the change in cross-sectional area of the device as compared to the flow diameter at the end caps, as well as the added compliance of constrained gas within the bladder. The performance of a device of a given outer shell and annulus geometry is dependent on the pressure of the constrained gas, particularly the ratio of the charge pressure and the system pressure [5, 6]. Prior work by the authors investigated the optimal charge pressure for bladder-style suppressors operating in both single and dual configurations. For dual configurations, both identical and dissimilar pairings were investigated. However, all prior work focused on established suppressor geometry [7, 8]. In Gruber and Cunefare [8], using dissimilarly sized suppressors within a hydraulic circuit was beneficial in some system usage cases, particularly for systems with especially broad or unpredictable system pressure ranges, prompting an investigation to find the optimal geometry of a suppressor pair; however, no work has been done to find the most effective suppressor geometry to treat noise.

Correct determination of the optimal dimensions of a suppressor will allow for more effective noise control, which in turn will allow for quieter machines and more effective solutions to the problems listed previously. There is no current literature related to the optimal sizing of bladder-style suppressors. As shown in Gruber [9], a current drawback of the current geometry is a transmission loss null in the frequencies of interest; an optimal design will target high transmission loss in those frequencies by changing the geometries of the suppressor pair.

The following sections of this paper will discuss how to find optimal suppressor geometry and analyse the results. First, a brief discussion of suppressor modelling and optimization will be presented, coupled with modifications to prior work in order to find optimal geometry. The following section will analyse the results of the objective function for a single system pressure. Next, example optimizations of a system with varying pressure will be optimized. A discussion on the effect of the annulus radius will then be presented. Finally, conclusions and suggestions for future work will be discussed.

2. MODELLING AND OPTIMIZATION

The noise control behaviour of a bladder style suppressor is dependent on its geometry, charge pressure and the system pressure, particularly the ratio of charge pressure to system pressure. For a single operating pressure, the manufacturer suggests a ratio of 50% between charge pressure and system pressure for optimal noise treatment [10]. However, Gruber et al. [7] demonstrated a ratio of 90% between charge
pressure and system pressure will yield the highest transmission loss; this charge pressure is known as the single pressure optimum. Transmission loss is the ratio of incident acoustic energy to transmitted acoustic energy across a device; it is a property of a given device and is not a function of flow rate through the device. Transmission loss varies by frequency; a higher magnitude of transmission loss indicates more effective noise treatment. The behaviour of the suppressors is predicted using a linear model to predict the wavefield upstream and downstream of the device by using acoustic pressure and particle velocity relations in regions both upstream and downstream of the device as well as within the device. A series of boundary conditions are enforced and the eigenvalue problem is solved to determine acoustic wave amplitudes. In order to calculate transmission loss, the model assumes an anechoic termination to the downstream pipe. The model only considers linear acoustics, makes an assumption of plane-wave behaviour upstream and downstream of the suppressor and considers the bladder as a limp mass; a more detailed derivation and explanation is present in Marek et al. [6].

An objective function was previously formulated to find the optimal charge pressure of a suppressor pair for a system operating over a range of system pressure [7]. The objective function is

$$
\bar{\mathcal{F}}(p_{c,1}, p_{c,2}) = \frac{1}{U \Omega} \sum_{i=1}^{N} \left| D_i \right| \sum_{j=1}^{M} \left| W(f) \right| \left| TL(f, p_{d,i}, p_{c,i}, p_{s,i}) \right|,
$$

where $D_i$ is a time weighting factor, $W_i$ is a frequency weighting factor, $f$ is frequency, $TL$ is transmission loss, $p_s$ is the system pressure and $p_c$ is the charge pressure of the suppressors. The objective function weights the transmission loss by the spectral content of noise within the system through $W_i$ as well as the usage history of the system through $D_i$. The optimal charge pressure condition is defined by the argument which maximizes the objective function,

$$
(p_{c,1}, p_{c,2})^* = \arg \max_{p_{c,1}, p_{c,2}} \bar{\mathcal{F}}(p_{c,1}, p_{c,2}).
$$

The objective function is weighted towards the frequencies with the highest acoustic energy by the frequency weighting factor (FWF)

$$
W_i(f) = \frac{\left| P_{d,i}(f) \right|}{\max_{m,j} \left| P_{d,j}(f) \right|};
$$

an example FWF is shown in Figure 2. The FWF is calculated by taking the magnitude of the fluid-borne noise, $P_d$, and normalizing it to the maximum fluid-borne pressure variation across all frequencies and all system pressures; the maximum for the example FWF occurs at a frequency of 240 Hz at a system pressure of 13.8 MPa. The total energy in each system pressure increases with system pressure from 3.45 MPa to 13.8 MPa; the acoustic energy stays relatively constant for the system pressures of 13.8 and 20.7 MPa. The higher noise level in the higher system pressures necessitates more effective noise control at those pressures. Both the suppressor performance and the spectral content of the fluid-borne noise are dependent on system pressure; therefore, the objective function, equation (1), is weighed to the most used system pressures by the time weighting factor (TWF),

$$
D_i = \frac{t_i}{t_{total}},
$$

where $t_i$ is the amount of time the system spends at the $i^{th}$ system pressure. Time weighting factors (TWF) can have any number of system pressure divisions. Two representative TWFs with four system pressure divisions are shown in Figure 3: Figure 3 (A) shows the duty cycle of a boom cylinder on an excavator performing a back filling task; Figure 3 (B) shows the time history of the same component performing a trenching task. The TWFs shown in Figure 3 will be used for optimizations in the following sections.
The objective function represented by Equation (1) may be extended to include suppressor geometry as design variables, yielding

$$\bar{\mathcal{F}}(p_{c,1}, p_{c,2}, G_1, G_2, S) = \frac{1}{|\mathcal{U}|} \sum_{c,j,k} |D_c| \left( \frac{1}{|\Omega|} \sum_{f \in \Omega} W(f) |TL(f, p_{c,j}, p_{c,k}, G_1, G_2, S)| \right)$$

where $G$ is the geometry of the suppressor and $S$ is the separation distance between suppressors, as shown in Figure 4. The geometry of a suppressor is defined by the design variables

$$G_n = \{r_{\text{annulus}, n}, r_{\text{shell}, n}, L_n\}$$

for each suppressor. Also shown in Figure 4 is the port radius, $r_{\text{port}}$, which is a system parameter and not included in the optimization as changing the port radius would require large and invasive changes to the system, i.e. the example optimizations are being conducted on an existing system to which the suppressors will be added. For the purpose of this work, the port radius is fixed at 0.95 cm for all cases. The optimal condition is also adjusted to account for the geometry of the suppressors,

$$(p_{c,1}, p_{c,2}, G_1, G_2, S)^* = \arg \max_{p_{c,1}, p_{c,2}} \bar{\mathcal{F}}(p_{c,1}, p_{c,2}, G_1, G_2, S).$$

Each suppressor has four volumes relevant to its noise control effectiveness. The first is the shell volume defined by

$$V_{\text{shell}} = \pi r_{\text{shell}}^2 L,$$

shown in Figure 5 A. Next, the initial bladder volume is defined by

$$V_{\text{bladder, initial}} = \pi \left( r_{\text{shell}}^2 - r_{\text{annulus}}^2 \right) L,$$
shown in Figure 5 B. For suppressors with charge pressures less than system pressure, i.e. the undercharged cases, the bladder will compress to the compliant volume of

$$V_{\text{compliant}} = \pi \left( r_{\text{shell}}^2 - r_{\text{compress}}^2 \right) L,$$

(10)

shown in Figure 5 C; $r_{\text{compress}}$ which is the radius of the bladder when exposed to system pressure, can be calculated from the initial charge pressure, the ideal gas law and a force balance on the bladder. When the suppressor is overcharged, i.e. the charge pressure is higher than system pressure, the suppressor behaves as an expansion chamber (the hydraulic analogue of a muffler in an airborne system) of volume

$$V_{\text{expansion}} = \pi r_{\text{annulus}}^2 L,$$

(11)

shown in Figure 5 D; in addition for the overcharged case, $V_{\text{compliant}}$ is defined to be zero [6].

Three thousand suppressor pairings were simulated in order to predict their transmission loss and find the optimal configuration in the following sections. For each simulation, the design variables of the suppressor pair were randomly selected between the bounds shown in Table 1 for each design variable; the geometry was constrained so the suppressor was feasible, i.e. the annulus is always smaller than the shell. The lower and upper bounds of radii and suppressor length match the corresponding dimensions of a WM-5081 and WM-5138 suppressors, respectively. The transmission loss for each suppressor pair was predicted at the
four system pressures in the TWFs. Prior optimizations calculated the transmission loss for a wide range of charge pressures; however, previous work showed all optimal cases were made up of single pressure optima; therefore, only single pressure optima were simulated for computational efficiency.

<table>
<thead>
<tr>
<th>Table 1. Optimization bounds</th>
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<tr>
<td>Dimension</td>
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<tr>
<td>$r_{annulus}$</td>
</tr>
<tr>
<td>$r_{shell}$</td>
</tr>
<tr>
<td>L</td>
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<tr>
<td>S</td>
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</table>

3. SINGLE PRESSURE OPTIMIZATION

An optimization for the simpler case of a system operating at a single pressure was analysed for trends that may provide insight into optimizations of systems operating with varied system pressures. For a given system pressure, a suppressor can be in two charges states: either undercharged, when the charge pressure is less than system pressure, or overcharged, when the charge pressure is greater than or equal to system pressure. For a given suppressor geometry, an undercharged state exhibits several orders of magnitude higher transmission loss than an overcharged state, which is more desirable from a noise control perspective. The geometry effect of both an undercharged and overcharged suppressor pair will be analysed to determine how a given dimension affects noise control.

The first case examined is a system pressure of 13.8 MPa and both suppressors charged to 13.1 MPa; this charge pressure is the single pressure optimum for the system pressure. In Figure 6, each point represents a simulated suppressor pair geometry when both suppressors are charged to 13.1 MPa. Furthermore, in Figure 6 the objective function values have been normalized to the maximum objective function value; this normalization is applied for all following optimizations as well. The objective function values are plotted as a function of shell volume in Figure 6 (A), the shell volume is normalized to the maximum possible combined shell volume within the design variable constraints, and there is a general trend that a suppressor pair with more combined volume will generally out-perform a smaller suppressor pair with less combined volume; however, the suppressor pair with the maximum objective function has a volume 93% of the maximum possible combined volume. In addition, the shell volume with the highest magnitude of suppressor volume is outperformed by four suppressor pairs with less combined volume. The maximum objective function values are plotted as a function of combined compliant volume in Figure 6 (B), the compliant volume is normalized to the maximum possible combined bladder volume. Similarly to the previous case, the largest combined compliant volume is not the highest performing suppressor geometry in this optimization, since suppressor geometry influences the frequency characteristics of transmission loss. The optimal configuration for noise control does have a high volume in comparison to maximum possible volume, both in terms of combined shell volume and combined compliant volume, and has a geometry which exhibits high transmission loss at frequencies in the FWF with high magnitudes, i.e. the transmission loss exhibited by the geometry targets the noisiest frequencies.
Figure 6. Objective function value: (A) as a function of combined shell volume, (B) as a function of combined compliant volume

The second single pressure case examined is for a system pressure of 13.8 MPa; however, the charge pressure of both suppressors is 20 MPa. The suppressors are purposely overcharged to determine which dimensions influence noise control in the overcharged state. Figure 7 (A), normalized in a similar manner as Figure 6 (A), shows the relation between combined shell volume and objective function value; there is no discernible relationship between combined shell volume and objective function. An overcharged suppressor behaves as a rigid walled expansion chamber with a cross sectional area equivalent to the size of the annulus. Figure 7 (B) was normalized to the maximum possible expansion chamber volume; it shows a strong linear relationship between combined expansion chamber volume and objective function value. The relationship is not important for a single pressure system as undercharged suppressors are orders of magnitude more effective and would be used for noise control in this instance, but will become important for a system operating over a range of pressures, which may induce an overcharged state in one or both of the suppressors. In a varied pressure case it is important for the suppressor to have a sufficiently large annulus radius to treat noise at lower system pressures.

Figure 7. Objective function value: (A) as a function of combined shell volume, (B) as a function of combined expansion chamber volume
4. VARIED SYSTEM PRESSURE OPTIMIZATION

Many common hydraulic systems operate over a broad range of systems pressures; an effective noise control solution must effectively treat noise for the entire pressure range. The objective function is weighted for systems operating with varied system pressures through use of the TWF. Two example optimizations will be presented, using the TWFs shown in Figure 3. The two selected TWFs are beneficial for analysis because of their different duty cycles; the back-filling TWF is weighted towards low pressures, while the trenching TWF is weighted towards higher pressures. The back-filling TWF may have many optimal suppressor configurations with a suppressor in an overcharged state. Because the trenching TWF spends more time at higher pressures, it is significantly less likely for a suppressor to be in an overcharged state.

The objective function values calculated for a system in a back-filling operation are analysed to find the optimal geometry and charged pressure. Figure 8 (A) shows the normalized objective function as a function of normalized combined shell volume. A key difference between the data presented in Figure 8 and the data presented in the figures of the previous section is the suppressor configurations shown here are not constrained to have the same charge pressure; each configuration has the charge pressure that makes it the most effective at noise control. Similar to above, the largest possible combined shell volume is not the most effective at noise treatment. Figure 8 (B) shows the objective function plotted against normalized time-weighted compliant volume; the normalization constant is the maximum possible combine initial bladder volume. The compliant volume is dependent on the initial charge pressure and system pressure; therefore, it will change during the duty cycle; it is time-weighted similar to the objective function above,

\[ V_{TW,\text{compliant}} = \sum_{i=1}^{N} DV_{\text{compliant},i}. \]  

(12)

Again, the most effective suppressor geometry is not the maximum possible combined time-weighted compliant volume. Since an overcharged suppressor is defined to have no compliant volume, a suppressor with a high charge pressure will have a significant reduction in time-weighted compliant volume without an associated drop in objective function value, as a high charge pressure treats noise in higher system pressures more effectively; the shift in compliant volume is the cause of the stratification of the objective function values in Figure 8 (B).

![Figure 8. Back filling objective function values: (A) as a function of compared shell volume, (B) as a function of combined time-weighted compliant volume. Red square represents a pair of WM-5138 suppressors.](image)

The individual geometry of the suppressor pair affects the frequency content of the exhibited transmission loss; an optimal noise control solution exhibits high transmission loss in the frequencies which carry the most acoustic energy. The design variables of the optimal suppressor pair are shown in Table 2; the separation distance for the optimal pair is 38 cm. Bladder-style suppressors behave linearly: if flow direction was reversed in direction suppressor noise control behaviour would not be affected. Suppressor 1 is in an
overcharged condition for approximately 40% of the duty cycle during which it behaves as an expansion chamber. It follows from the discussion in the previous section regarding overcharged suppressors that the annulus radius of suppressor 1 would be at the upper end of the allowable range. In actuality, the annulus radius of suppressor 1 is near the minimum allowed by the dimensions in Table 1. The explanation is the 3.45 MPa system pressure carries comparatively little acoustic energy, as demonstrated in Figure 2, and suppressor 2 is correctly charged to successfully treat noise at this pressure; therefore, suppressor 1 is superfluous for low pressure noise treatment. Its large initial bladder volume, which is a function of a small annulus radius, allows it to treat noise well at higher pressures which have more energy. The combined effect is effective noise treatment for the range of systems pressures. Furthermore, the suppressor pair does not have a transmission loss null at a frequency with high spectral content. In Figure 8 (A) and (B) the red square represents a pair identical commercially-available Wilkes and Mclean of WM-5138 suppressors, the largest variant made which is rated for the system pressures examined; the pairing of WM-5138 suppressors is out-performed by suppressor pairings with similar combined volumes because the identical suppressors have the same transmission loss null which is detrimental to its noise control effectiveness. This underscores the need for suppressor geometry to be considered when seeking an effective noise control solution.

<table>
<thead>
<tr>
<th>Table 2. Optimal Suppressor Pair Dimensions</th>
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<tr>
<td>Dimension</td>
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<tr>
<td>Annulus radius [m]</td>
</tr>
<tr>
<td>Shell radius [m]</td>
</tr>
<tr>
<td>Length [m]</td>
</tr>
<tr>
<td>Charge Pressure [MPa]</td>
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The transmission loss for the optimal suppressor pair and a pair of WM-5138 suppressors, both charged to their respective optimal charge pressures, for all system pressures is shown in Figure 9; the FWF for each system pressure is also shown. Neither suppressor pair exhibits a transmission loss null at a system pressure of 3.45 MPa; however, at all other system pressures the both suppressor pairings exhibit a transmission loss null. The null for the optimal suppressor is at lower frequencies and generally less severe decrease in transmission loss. By moving the transmission loss null to lower frequencies, the optimal suppressor pair treats noise more effectively at the frequencies which carry the most acoustic energy. The difference in transmission loss between the optimal suppressor pair and a pair of WM-5138 suppressors is almost 20 dB through the frequency ranges which carry the most acoustic energy, which is a significant difference in performance. In the higher frequency ranges, above 500 Hz which is not shown on Figure 9, the transmission loss of the pair of WM-5138 suppressors exceeds that of the optimal suppressor pair. However, there is little acoustic energy at those frequencies so this difference does not materially affect suppressor effectiveness.

Figure 9: Transmission Loss of Optimal Suppressor pair and WM-5138 suppressor pair, both at optimal charge pressure for all system pressures. FWFs for all system pressures also presented for comparison.
5. ANALYSIS OF ANNULUS RADIUS

The size of the annulus radius has been demonstrated to be important to effective noise control in the previous sections; this section will further investigate its impact. In order to fully examine the influence of the annulus radius on device performance, the shell radius, device length and separation distance were fixed to their maximum bounds in Table 1. The annulus radius was varied for both suppressors in a pair for 200 transmission loss simulations. The results were first analysed with a similar methodology as the single pressure optimization section. An example case where both suppressors were charged to 13.1 MPa and the system operated at 13.8 MPa was analysed by calculating the objective function value for all 200 cases simulated. The result of the analysis when both devices were undercharged is shown in Figure 10; small annulus radii outperform large annulus radii, as expected because a small annulus radius allows a suppressor to have a larger initial bladder volume and compliant volume. A second optimization was conducted with the same system pressure but both suppressors charged to 20.0 MPa, the objective function values are shown in Figure 11. For this setup, a large annulus radius is better for effective noise control as the larger radius allows the over charged suppressor to behave as an expansion chamber with a larger cross-sectional area change.

![Figure 10](image10.png)

*Figure 10. Objective function values as a function of annulus radii, suppressors undercharged.*

![Figure 11](image11.png)

*Figure 11. Objective function values as a function of annulus radii, suppressors overcharged.*

The objective function when varying the annulus radii of the suppressor pair can also be calculated for systems with varied pressures. The objective function for a back filling usage is shown as a function of both annulus radii in Figure 12. The optimal case is the smallest annulus radii pair, with suppressor 1 charged to
6.21 MPa and suppressor 2 charged to 2.76 MPa. Comparable to the undercharged single pressure simulation, the noise control effectiveness of the suppressor pair decreases as the total of the annulus radii increases. The objective function for this case is also plotted as a function of normalized compliant volume in Figure 13. The data becomes stratified because the compliant volume of an overcharged suppressor is defined to be zero. As with other cases, the right most stratum has the most effective noise control solution; however, for the set of compliant volumes where the strata overlap, the left stratum performs better. The data becomes stratified when plotted as a function of time-weighted compliant volume because an overcharged suppressor has no compliant volume, effectively shifting the compliant volume to lower values; however, the relatively high charge pressure treats higher system pressures effectively which keeps the objective function value high.

![Figure 12](image1.png)

*Figure 12. Objective function values as a function of annulus radii for a back filling usage*

![Figure 13](image2.png)

*Figure 13. Objective function value as a function of compliant volume for a back filling usage*

6. CONCLUSIONS & FUTURE WORK

Three thousand geometries of suppressor pairs were analysed to find the optimal geometry and charge pressure to treat noise in a hydraulic system operating with at two example duty cycles with a given spectral content. Furthermore, two hundred suppressors with varying annulus radii were analysed to determine the importance of this dimension to noise control. Analysis showed that the largest suppressor pair, in terms of combined shell volume, initial bladder volume or time-weighted compliant volume, is not the most effective
The most effective suppressor did have a relatively large volume, but the distinguishing feature is its geometry generated high transmission loss in the frequencies that carry the most acoustic energy. The simulated geometry which accomplished this was the most effective at treating noise for both of the varying-pressure usage cases analysed. It was also shown that it is desirable to have a small annulus radius when the suppressor is undercharged to facilitate a large bladder volume. When the suppressor was overcharged, it was desirable to have a large annulus radius, during which the suppressor performs as an expansion chamber. Generally, an undercharged suppressor treats noise significantly better than an overcharged suppressor, regardless of geometry; therefore, it is important to consider charge pressure as an optimization variable, as well. Future work in this area would focus on determining a procedure for finding optimal suppressor geometry. In addition, future work would generate FWFs for more system pressures; this would allow for the duty cycle data to also be generated at a finer resolution. Other work could include analysing systems with different frequency weighting factors generated from different pumps or system architectures to analyse the effect of spectral content on finding the optimal noise control geometry.

7. ACKNOWLEDGEMENTS

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REFERENCES

AN APPROACH TO OPTIMIZE THE DESIGN OF HYDRAULIC RESERVOIRS

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ABSTRACT

Increasing demands regarding performance, safety and environmental compatibility of hydraulic mobile machines in combination with rising cost pressures create a growing need for specialized optimization of hydraulic systems; particularly with regard to hydraulic reservoirs. In addition to the secondary function of cooling down the oil, two main functions of the hydraulic reservoir are oil storage and de-aeration of the hydraulic oil. While designing hydraulic reservoirs regarding oil storage is quite simple, the design regarding de-aeration can be quite difficult. The author presents an approach to a system optimization of hydraulic reservoirs which combines experimental and numerical techniques to resolve some challenges facing hydraulic tank design. Specialized numerical tools are used in order to characterize the de-aeration performance of hydraulic tanks. Further the simulation of heat transfer is used to study the cooling function of hydraulic tank systems with particular attention to plastic tank solutions. To accompany the numerical tools, experimental test rigs have been built up to validate the simulation results and to provide additional insight into the design and optimization of hydraulic tanks which will be presented as well.

KEYWORDS: Hydraulic reservoirs, filter-tank system, numerical optimization, experiment, simulation, de-aeration performance, degassing function, heat transfer

1. INTRODUCTION

A basic function of hydraulic reservoirs in mobile applications is the storage of oil. Most of the mobile machines are equipped with differential cylinders which have a variable fluid capacity. In consequence of the cylinder operation the oil volume in the tank varies. The oil volume in the tank also changes as a conclusion of the thermal expansion of the oil during the warm-up of the machine. In addition to this variable amount of oil stored in the tank there is always a fixed amount of oil. This is necessary to ensure the degassing of the free air in the oil. The source of the air will be explained in the following.

Henry’s law describes the amount of air \( V_G \) that can be solved in a defined volume of hydraulic oil \( V_{Fl} \) at a specific pressure \([1]\).

\[
V_G = V_{Fl} \cdot \alpha_v \cdot \frac{p}{p_0}
\]

For mineral oils in balanced condition at atmospheric pressure \( p_0 \) approximately seven percent by volume is in solution. Due to local pressure drops in the suction lines of pumps as well as due to jet streams especially
in hydraulic valves the amount of air that can be solved according Henry’s law is reduced proportionally to the pressure. As a consequence free air in form of air bubbles is present. A further source of air ingressation to the oil is the sloshing of the oil in the tank. There is a big spectrum of bubble diameters in the range of approximately one µm and five mm. In some cases the bubble diameter can be even bigger when air can be deposited in the hydraulic system.

The free air in hydraulic systems causes degradation of the oil as a conclusion of oxidation. Furthermore due to the compression of the air bubbles high temperatures can occur which also damages the oil. A further consequence of this effect is the reduction of the overall efficiency of the hydraulic pumps and therefore an increase in fuel consumption. Due to the increased compressibility the handling and control of the hydraulic systems also can decline.

Degassing bubbles of free air in the hydraulic tank is a complex function and is depending on many influencing factors. Increasing the amount of fixed oil in the tank generally improves the degassing function. An optimization of the design of hydraulic tanks in regard of the de-aeration performance allows improving the degassing function without increasing the amount of oil or even in combination with a reduction of the tank size. As manufacturer of mobile machines are strongly driven by a big cost pressure due to global competition as well as packaging issues mainly due to changes in legislation the interest in optimizing hydraulic reservoirs is very high. Concrete guidelines how to optimize tank systems are still missing and have to be worked out.

As the return filter which is normally mounted into the tank is essentially influencing the degassing function in the following the term filter-tank system will be used.

A further function of the tank is the reduction of the oil temperature due to heat transfer to the environment. This property is especially for smaller machines very interesting. In some applications an improvement of the tank design even allows the elimination of the main cooler.

2. EXPERIMENTAL CHARACTERIZATION OF HYDRAULIC RESERVOIRS

In order to be able to optimize hydraulic tank systems a characterization of the degassing function is necessary. An experimental approach will be described. Furthermore a test setup to determine the heat transfer of tank systems will be shown.

2.1. De-aeration performance

In order to characterize the de-aeration performance of hydraulic reservoirs a test setup for hydraulic filter-tank systems was build up. Furthermore to be able to measure the amount of free air in hydraulic systems a sensor was developed which is able to measure the amount of free air in the oil.

2.1.1. Air Content Sensor

In order to characterize and optimize hydraulic reservoirs regarding de-aeration performance the content of free air in hydraulic fluids has to be determined. For this purpose a special sensor was developed. The sensor is based on the electric capacitive principle. The effective dielectric constant of the fluid in the capacitor defines the capacity of the capacitor. While oil has a dielectric constant of approximately 2.7 air has a value of approximately 1. An increased amount on free air in the oil will reduce the capacity of the capacitor and can thereby be used for quantification. Because of the dependency of the dielectric constant with the temperature a compensation of the measurement result has to be performed. The sensor as well as the test setup will be introduced in the following.
2.1.2. Test setup

Figure 1 shows the hydraulic schematics of the test setup.

![Test setup diagram]

The volume flow through the tank is regulated using a variable pump. Using a special nozzle in combination with a mass flow controller air is injected to the suction line in a defined way. The Air Content Sensor (ACS) is used to measure the air content in the suction line with a sampling rate of one Hz. Figure 2 shows an exemplary measurement result comparing two different tank concepts.

![Measurement results graph]

One percent of air is injected at the point two minutes for the duration of one minute. Looking at the transient air content it is obvious that the de-aeration performance of the optimized filter-tank system is on a better level.

The test rig with the special Air Content Sensor allows a characterization of tank systems regarding de-aeration performance. It allows a comparison of different tank designs and can be used to validate optimizations evaluated using simulation tools.

2.2. Heat transfer

In order to quantify the cooling function of tank systems a wind tunnel test was performed. The hydraulic reservoir is placed inside the wind tunnel and is equipped with several thermocouples measuring the temperature of tank, air and oil. The hydraulic system is mounted on the outside of the tunnel in order to avoid disturbances. Figure 3 shows the tank placed inside the tunnel.
At the inlet of the wind tunnel on the opposite of the fan drive the velocity field was measured. The result of the flow field during the test is shown in Figure 4.

This measurement allows the calculation of the total volume flow through the tunnel as an input value for the simulation model. The temperature of the oil in the tank was used in order to determine the heat transfer. Before starting the fan drive the oil was heated up to 60°C. Afterwards the oil flow through the tank was stopped. Figure 5 shows the profile of the oil temperature measured during the test time.

This measurement profile will be used in chapter 3.2. in order to validate the results from simulation.
3. SIMULATION OF HYDRAULIC RESERVOIRS

In the following an approach to determine the de-aeration performance of hydraulic filter-tank systems is shown. Furthermore the results of a thermal heat transfer simulation will be shown and will be compared to the experimental results shown in chapter 2.2.

3.1. De-aeration performance

A multi-phase bubble flow simulation can be used in order to calculate the de-aeration performance of filter-tank systems. Bubbles with different sizes in diameter are injected to the return line. The term simulated de-aeration performance which is used in the following is defined as the number of bubbles de-aerated inside the tank in relation to the number of bubbles injected to the tank. It has to be noted that this value is dependent on the bubble size. Figure 6 shows two simulation results done for the two different tank systems which were used in chapter 2.1.2.

![Simulated de-aeration performance](image)

Figure 6. Simulated deaeration performance

This approach can be used to optimize hydraulic tank systems. Nevertheless an experimental validation as shown in chapter 2.1.2 is strongly recommended.

3.2. Heat transfer

In the following a comparison of the results from the heat transfer simulation with the experimental investigations shown in chapter 2.2. is shown. Figure 7 shows the simulated velocity profile during the test.
The volume flow at the fan drive as a boundary condition was adjusted to a value that fits the simulated volume flow at the tunnel inlet to the measured flow. Figure 8 shows the simulated heat transfer density at the surface of the tank. It can be observed, that the heat transfer at the upper level of the tank where air is present is on a much lower level as at the part where the tank is filled with oil.

An integration of the heat transfer density over the tank surface gives the overall heat transfer. Figure 9 shows a comparison of measurement and simulation.
It can be seen that the simulated results are qualitatively as well as quantitatively in a very good agreement with the measurement. It can be concluded that the usage of thermal simulations to determine heat transfer at tank systems is quite accurate.

4. SUMMARY

The main function of hydraulic reservoirs is the degassing of the free air in the oil. The amount of oil inside the tank is only secondarily accountable for the degassing performance. Primarily it is the design of the inner tank geometry as well as the type and position of the filter. Because of the lack of knowledge in the design of tanks many hydraulic reservoirs are still sized in a very conservative way. Optimizing the design of hydraulic filter-reservoir systems allows a reduction of the tank size. This allows a significant cost reduction as well as improving the package of components on mobile machines.

By usage of an experimental setup which contains a sensor to measure the amount of free air the degassing performance of filter-reservoir systems can be determined. The usage of multi-phase bubble flow simulation tools allows a computer aided optimization. Nevertheless an experimental validation is always strongly recommended.

A secondary function of hydraulic reservoirs is the cooling of the oil due to heat transfer to the environment. A quantitative analysis helps to dimension the cooling circuit of the system. A comparison with experimental results has shown that the results from a thermal simulation are quite accurate and therefore very useful to be used for the dimensioning of the cooling circuit.

REFERENCES

SECOND ORDER DYNAMIC ACCUMULATORS, THE FEATURES, THE APPLICATIONS AND THE FEASIBILITY

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ABSTRACT

In this paper, new type of compact hydraulic accumulators is introduced with the capability to store high amount of hydraulic energy and to change its mean working pressure without exchanging effective volume of oil with the system. This new accumulator, called “Dynamic Accumulator”, is a sort of a second order dynamic system where the swept volume of the accumulator would depend on the dynamic periodic variation of the working pressure in addition to its dependency on the static variation. Various features of this second order dynamic accumulator which comprises spring-mass system are presented. The proposed accumulator has the advantage of being capable of tuning its natural frequency, change its damping coefficient and control its mass movement online. Thus, the accumulator is suitable for wide range of working pressures and operating conditions. Applications in suppressing the pulsations and suggested locations to attach the accumulators are presented. The usage of this accumulator to drive and control pumping heads of a new under development pump of geometric volume control by small amount of control oil is explained. Feasibility regarding the developed stresses in the accumulator components is proven by finite elements simulation and capability to dynamically suppress the pulsation when integrated with the system is proven analytically.

KEYWORDS: second-order, dynamic accumulator, spring-mass, pulsations, digital hydraulics

1. INTRODUCTION

One of the functions of hydraulic accumulators is to suppress pulsations in hydraulic lines. Accumulators could store hydraulic energy in the form of volume of pressurized oil. Thus, they store amount of pressurized oil during feed surplus in order to supply the system during feed shortage under the operating pressure. Accumulators are also used in applications such as energy storage, emergency operation, leakage fluid make-up, volume compensation and shock absorption [1].

P. Drexler et al [1] summarized the types of accumulators as follows: 1) gas-charged accumulator, this type includes the accumulators with separators between the charged gas and the hydraulic oil and the types with no separators. Common gas-charged accumulators are the bladder accumulators, piston type accumulators, and diaphragm accumulators; 2) spring-loaded Accumulator, such as the spring-loaded piston type; 3) the weight loaded accumulator, where the storage media is the potential energy transferred to left the weight.

To cope with pulsation suppression requirements, gas-charged accumulators with elastic elements are commonly used due to the low inertia of the moving parts. Absence of inertia leads to perform as a first order
system, and hence the system response to static pressure variations and variations of frequencies below the accumulator corner frequency are almost the same. Moreover, the operating pressure should be considered in the accumulator pre-charging pressure, which is difficult to be changed online. Changing the operating pressure could be achieved by storing or releasing considerable amount of stored pressurized oil due to the high flexibility of the compressible charging gas. This may not be desirable in some applications and leads to using large accumulators due to the low energy density stored in the compressed gas.

In order to cover the drawback of operating pressure range change, Swift [2] suggested some sort of a piston type accumulator with controllable charging gas volume and pressure through a highly sensitive compensator valve system. Stauch et al [3] introduced a new type of gas charged multi-area accumulator for the same purpose of covering various ranges of operating pressures. The accumulator considered utilizes digital fluid power principles by allowing for a stepwise variation of the actual pressurized area in the section facing the load connection. This is realized by means of several binary coded fluid chambers in parallel, connected to a switching valve manifold.

Elgamil [4] proposed to use the concept of the second-order spring mass system shown in Figure 1 with damper of low damping coefficient is utilized in building the dynamic accumulator. The idea is to implement the dynamic system frequency response in suppressing the pulsations. Second-order systems magnify the amplitude response to frequency input variations with respect to response to static input variations of the same magnitude according to the excitation frequency, the system natural frequency and the damping ratio. Therefore, the accumulator storage capability to store certain pressurized oil volume will be more sensitive to variations due to pulsations than the variation in the mean value of the operating pressure. Moreover, if the natural frequency is tuned to the pulsation frequency, accumulator can suppress the pulsation to very small values, same behavior as the dynamic absorber effect on a vibrating system, this effect is shown in [5].

Pulsations are not pure sine waves, but as a periodic wave, it could be considered as a series of harmonically related sines and cosines by using Fourier series [5].

---

![Second-order spring mass system](image)

*Figure 1. Second-order spring mass system*

Dynamic accumulator, same as dynamic absorber, will replace an original second-order system of one frequency response peak with a system of two frequency response peaks after integration. Therefore, the hydraulic accumulator natural frequency should be carefully tuned to avoid resonance of any of the harmonics of the Fourier series with any of the new peaks of the integrated system.

2. THE SECOND-ORDER DYNAMIC ACCUMULATOR

Figure 2 shows one of the configurations that can implement the second order system concept as an accumulator. The accumulator shown in the figure is attached to a hydraulic line. The spring is a heavy spring of continuous closed walls to isolate the system pressurized oil from its outsiders. The density of the strain energy stored in the stressed spring material is high with respect to the density of the energy stored in
compressed and pressurized charging gas. High density of strain energy and large dynamic swept volume lead to build compact accumulators. The optional gas pre-charging can increase the spring stiffness and reduce the developed stresses inside the spring material in addition to increase the media that store energy.

3. SECOND-ORDER DYNAMIC ACCUMULATOR FEATURES AND APPLICATIONS

The simplicity and compactness of the configuration shown in Figure 2 provide design flexibilities to adopt features that cope with different applications and working conditions. In Figure 3, the motion of the mass is guided by controlled guides. The roller provides lowest viscous friction so that the moving materials internal damping would be the only damping in the system. Stoppers can restrict the mass motion when no exchange of pressurised oil with the system is required. Mass movement damping could be controlled through the guides’ motion control, where the mass movement is coupled with the guides’ movement by the stoppers.

Figure 4 shows how to control the spring stiffness and hence the accumulator natural frequency by the control of the pre-charge gas pressure and amount. Pre-charging with oil will make the spring stiffer. The control could be more effective by adding another small accumulator inside the accumulator as shown in Figure 5. This small accumulator is pre-charged with oil while keeping the space between the two bellows filled with pressurized gas.
Figure 4. Spring stiffness control by the control of pre-charge gas pressure and amount

Figure 5. Increasing the spring stiffness control effectiveness by using pre-charge oil

Figure 6 shows the feed of the pre-charge oil from the system own pressurized oil instead of external source.

Figure 6. Feeding the pre-charge oil from the own system oil
Elgamil [6] and Elgamil et al [7] introduced a new pump of high speed of response which its geometric volume is controlled by small amount of control oil volume. By utilizing the dynamic accumulator as shown in Figure 7, only one pumping head of this pump type could be used, where the dynamic accumulator attached to its control oil chamber can provide a compact component to store energy during pump suction stroke and drive the pump followers during its discharge stroke. Another accumulator attached to its discharge line could be used for suppressing the high pulsations due to using only one pumping head as shown in the figure.

![Pump Diagram](image)

*Figure 7. Using the dynamic accumulator to drive a new pump type of only one head*

Digital control, on/off control and piston to piston pump control are growing technologies that become more in use in many industries and applications. Second-order dynamic accumulator provides a compact and effective component to suppress the pulsations associated with the applications of these technologies.

4. FEASABILITY

In this part the feasibility is investigated regarding the capability of storing pressurized oil and the associated developed stresses at certain swept volume and displacement as well as the accumulator capability to dynamically suppresses the pulsations.

4.1. Swept Volume and Associated Developed Stresses

To illustrate the feasibility of the accumulator regarding its capacity to store oil volume and the associated stresses developed due to its spring deflection that reveals the swept volume in which the oil is stored, some numerical values are presented in this section. The spring is built of fourteen disc springs of 6 mm thickness, 28.5 mm inner radius, 56 mm outer radius and 8.2 mm total height. Each disc spring is connected to its next one through a proper ring with dimensions selected to reduce the developed stresses. The system has 3.5 kg attached mass and 170 mm mass spring arrangement total height. The analysis is carried out by using ANSYS structural analysis and the natural frequency \( w_n \) is found to be 1900 rad/s.
Applying 20 MPa static pressure to the outside surface of the accumulator results in 5.2 mm mass deflection, this deflection is equivalent to 25 cc swept volume. The maximum developed stress is about 1000 MPa. Stress distribution is shown in Figure 8 while displacement distribution is shown in Figure 9.

When the accumulator is charged with 16 MPa pressurized gas, the maximum stress decreases to only 200 MPa while the mass deflection decreases to 1.035 mm and the associated swept volume is limited to only 5 cc under the same 20 MPa applied oil pressure. Deflection distribution is shown in Figure 10 while stress distribution is shown in Figure 11.

To illustrate the dynamic load capabilities to increase the deflection, the oil pressure is applied with mean value of 20 MPa and 2 MPa amplitude variation and 1900 rad/s frequency (same as the natural frequency) according to the equation:

$$p = 20 + 2 \sin(1900t) \text{ MPa}$$

The resulting deflection oscillates between 0 and 11 mm. This means that the 5 mm deflection due to the 20 MPa static pressure has the same magnitude as the 5 mm amplitude due to 2 MPa fluctuating pressure, i.e. the magnification is 10 times.
4.2. Pulsation Suppression

To illustrate the capability to suppress the pulsations, an accumulator of \( m_a \), \( k_a \) and \( c_a \) mass, stiffness and damping coefficient is assumed to be attached between the pulsating flow supply \( Q_{in} \) and the load. The accumulator intervenes with its flow rate \( Q_a \) in order to suppress the pulsations in the load feeding flow rate \( Q_L \). The load is presented as a spring-mass system of \( m_L \), \( k_L \) and \( c_L \) mass, stiffness and damping coefficient, subjected to \( F_L \) external force and driven by a hydraulic cylinder actuator of area \( A_L \).
The mathematical model of the system can be expressed as follows:

\[ Q_L = A_L \dot{x}_L \]  \hspace{1cm} (2)

\[ Q_a = A_a \dot{x}_a \]  \hspace{1cm} (3)

\[ Q_{in} - Q_a - Q_L = \frac{V}{\beta} \dot{p} \]  \hspace{1cm} (4)

\[ pA_a = m_a \dot{x}_a + c_a x_a + k_a x_a \]  \hspace{1cm} (5)

\[ pA_L = m_L \dot{x}_L + c_L \dot{x}_L + k_L x_L + F_L \]  \hspace{1cm} (6)

Where \( V \) is the oil volume trapped in the line, \( p \) and \( \dot{p} \) are the line pressure and its derivative, \( \beta \) is the oil Bulk's Modulus, \( \dot{p} \) is the rate of the change of the oil line pressure, \( x_a, \dot{x}_a, \ddot{x}_a \) is the accumulator displacement and its derivatives, \( x_L, \dot{x}_L, \ddot{x}_L \) is the load displacement and its derivatives, and \( A_a \) is the accumulator equivalent area.

The natural frequency \( w_n \) could be obtained from the finite elements modal analysis then the accumulator stiffness could be obtained from equation 7:

\[ k_a = m_a w_n^2 \]  \hspace{1cm} (7)

It should be noted that the accumulator equivalent mass \( m_a \) equals the attached mass plus one third of the heavy spring mass [5].

To obtain the equivalent accumulator area \( A_a \), the finite elements model is subjected to static pressure \( p_a \) and the corresponding static deflection \( x_{ast} \) is measured. Then the equivalent area could be calculated from equation 8:

\[ A_a = \frac{k_a x_{ast}}{p_{st}} \]  \hspace{1cm} (8)

For verification, the resulting swept volume was measured to comply with the equivalent area of the accumulator \( A_a \) in equation 3.

Then the transfer function between the load flow rate and the input flow rate can be written as:

\[ \frac{Q_L}{Q_{in}} = \left[ (m_a s^2 + c_a s + k_a)A_{L_0}^2 \beta \right] \right] \\
\left[ (V m_L m_a s^4 + V (c_a m_L + m_a c_L) s^3 + (V m_L k_a + c_a c_L + k_L m_a) + \beta (A_{L_0}^2 m_L + \beta A_{L_0}^2 m_a) s^2 + (\beta A_{L_0}^2 c_L + \beta A_{L_0}^2 c_a + V c_a k_L + V k_c c_L) s + V k_a k_L + \beta A_{L_0}^2 k_a + \beta A_{L_0}^2 k_L) \right] 

And the transfer function between the line pressure and the input flow rate can be written as

\[ \frac{p}{Q_{in}} = \left[ (m_a s^2 + c_a s + k_a) \right] \right] \\
\left[ (V m_a m_L s^4 + (V c_a m_L + V m_a c_L) s^3 + (V m_a k_L + V c_a c_L + \beta A_{L_0}^2 m_L + V k_a m_a + \beta A_{L_0}^2 m_a) s^2 + (\beta A_{L_0}^2 c_L + \beta A_{L_0}^2 c_a + V c_a k_L + V k_a c_L) s + V k_a k_L + \beta A_{L_0}^2 k_a + \beta A_{L_0}^2 k_L) \right] 

It can be noted that the term \( (m_a s^2 + c_a s + k_a) \) appears in the two transfer functions' numerators. This means that if the accumulator natural frequency is tuned with the excitation frequency, the response amplitude could be reduced to very small values.
Using the above mentioned accumulator parameters in addition to equations 7 and 8, then the accumulator equivalent parameters are calculated and found to be: the equivalent mass \( m_a = 5.808 \) kg, the accumulator stiffness is \( k_a = 21186949 \) N/m and the accumulator equivalent area \( A_a = 54.98 \times 10^{-4} \) m\(^2\).

Assuming the accumulator damping coefficient \( c_a = 0 \) N.s/m, the load equivalent mass \( m_L = 1000 \) kg, the load equivalent damping coefficient \( c_L = 1000 \) N.s/m, the load equivalent stiffness \( k_L = 0 \) N/m, the load equivalent area \( A_L = 100 \times 10^{-4} \) m\(^2\), the oil volume trapped \( V = 0.5 \times 10^{-3} \) m\(^3\) and the oil Bulk’s modulus \( \beta = 1.4 \times 10^9 \) Pa, then the bode diagram for \( (Q_L/Q_m) \) is shown in Figure 13, and for \( (p/Q_m) \) is shown in Figure 14.

\[ \text{Figure 13. Bode Diagram for } \frac{Q_L}{Q_m} \]

\[ \text{Figure 14. Bode Diagram for } \frac{p}{Q_m} \]

Figure 13 shows that the accumulator is capable of suppressing the load flow rate pulsations over a wide range of frequencies around the accumulator natural frequency. However, its capability to suppress pulsations in the pressure as shown in Figure 14 does not have the same wide range.
Bode diagram and frequency response represent the steady state response, however the transient response is more important in many applications. Therefore, online controllability of the accumulator parameters is important for these applications.

5. CONCLUSION

A new type of second-order hydraulic accumulator is introduced in this paper. The design concepts and the various features are presented, and the ability to online control of the accumulator parameters is illustrated. The various aspects of static and frequency response are analyzed and discussed. The feasibility regarding the developed stresses inside the accumulator component is proven by using Finite Elements Simulation and the capability to suppress the periodic pulsations is proven by using mathematical model analysis.

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