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Energy Efficient Digital Hydraulic Power Management of a Multi Actuator System

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Thesis for the degree of Doctor of Science in Technology to be presented with due permission for public examination and criticism in Konetalo Building, Auditorium K1702, at Tampere University of Technology, on the 26th of May 2016, at 12 noon.

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ABSTRACT

One performance measure of hydraulic motion control is efficiency or as looked from a different angle: the amount of losses. The topic of this thesis is to study energy saving potential in multi-actuator applications by means of digital hydraulic innovations.

A dream of the perfect valve has been a driving force behind the development of digital hydraulic valves. Simple question: what makes a valve perfect and how to realize it? One answer is simple: a digital valve system – a DVS. The same questions have been asked considering the hydraulic supply unit. The perfect supply unit supplies exactly the required power for multiple actuators and handles also the recuperation. The chase of this dream has resulted in a digital hydraulic power management system – a DHPMS, or more likely the DHPMS as it is currently the only one of its kind in the world. Combining these two “dream-come-true” pieces of technology together, efficient hydraulic circuit design is made feasible. However, the control of such combination is not trivial; hence holistic control system design is inspected.

The objective of the thesis is to study DHPMS capability to work as an ideal sink/source of hydraulic power. The test setup emulates realistic application and energy efficiency is taken into consideration. The origin of losses is inspected and analysed and solutions are presented. Simulations and measurements are used to prove the potential of the technological advancements. The functionality of the system is considered; power flow within the system lines is matched to the independent demands of different actuators and the power can be routed from an actuator to another- if there is a negative load on one actuator while positive on the other. There is no need to have the same load pressures as the DHPMS has a built in transformer feature.

The results show that the DHPMS works, but in order to gain all the benefits made feasible by of the DHPMS, also DVSs are required. Applying digital hydraulic innovations into a two-degree-of-freedom test system, 40% energy savings are achievable compared with the state of the art system. Similar results are gained from both simulations and measurements. Generalization of the results and future insight is discussed.

PREFACE

Research under the topic “digital hydraulics” has been carried out roughly for a decade at Tampere University of Technology. The author joined the team in fall of 2007 as research assistant and continued as researcher after completion of his master’s degree in summer of 2009.

Research started with digital valves and nowadays the results on digital valves and their control methods have already been utilized in industry. Later on, digitalization was applied to pumps and cylinders and research on new areas emerged. The idea of the DHPMS was founded on the utopia of a universal multiport sink and source of hydraulic power and the possibilities of which seemed outstanding.

In this thesis, the combination of two new technologies, DHPMS and DVS, were united and applied. Mr. Mikko Huova has designed control methods for DVS while Mr. Mikko Heikkilä has concentrated on the DHPMS. The author is in deep gratitude for Mr. Huova and Mr. Heikkilä for co-operation and support. Also the author is grateful to Adj. Prof. Linjama for supervising the research and Prof. Seppo Tikkanen for his guidance on the thesis work. Also thanks to Mr. John Shepherd for language revision and laboratory staff for helping with the hardware.

The thesis is divided in chapters. First chapter introduces the field on which the topic is related and is written for audience not familiar with the area of machine automation. Generalities which are well known to professionals are told.

In the second chapter, state of the art, literature and technology survey is conducted. The chapter is written in certain manner, so that people with general engineering understanding and not just professionals on hydraulics, may follow.

The third chapter draws boundaries of the study and contribution is explained. This is continued with the research plan and explanation of the methods.

The fourth chapter describes the system; hardware and measurement equipment. In the fifth chapter the control system, which is the big brain of the intelligent system, is explained.

The sixth chapter tells about simulations and simulation results. The seventh follows with measurements and measurement results. Discussion about the results comes as the chapter 8 and conclusion ends the thesis.

Nomenclature

Symbol	*Description*	*Unit*
L	Cylinder maximum length	[m]
A_A	Cylinder piston area	[m^2]
A_B	Cylinder rod-side piston area	[m^2]
B	Bulk modulus	[Pa]
b	Viscous friction coefficient	[Ns/m]
b_1, b_2	Cavitation choking coefficient	[-]
B_{eff}	Effective bulk modulus	[Pa]
C_h	Hydraulic capacitance	[m^3/Pa]
E_{DHPMS}	Energy to/from the hydraulic circuit	[J]
E_{in}	Energy to/from prime mover	[J]
E_{out}	Energy to/from hydraulic actuators	[J]
E_{PM}	Energy to/from prime mover	[J]
F	Force	[N]
F_{2ndf}	Force after 2nd order filtering	[N]
F_C	Coulombian friction coefficient	[Ns/m]
F_{fric}	Friction force	[N]
f_{lb}	Lower boundary of a window in nonlinear force filter	[N]
F_{NLf}	Force after nonlinear filtering	[N]
F_{raw}	Raw force data (nonfiltered)	[N]
F_S	Static friction coefficient	[N]
f_{ub}	Upper boundary of a window in nonlinear force filter	[N]
K_h	Hydraulic spring constant	[N/m]

KP	Proportional gain	[1/s]
Kv_1, Kv_2	Valve flow model flow coefficient	[-]
N	Number of valves in a DFCU	[-]
p	Pressure	[Pa]
p_{err}	Pressure error	[Pa]
p_{err_est}	Pressure error estimate	[Pa]
P_{in}	Input power	[W]
P_{out}	Output power	[W]
pA	Cylinder A-chamber pressure	[Pa]
pA_{ref}	Cylinder A-chamber pressure reference	[Pa]
pB	Cylinder B-chamber pressure	[Pa]
pB_{ref}	Cylinder B-chamber pressure reference	[Pa]
pS	Supply pressure	[Pa]
pS_{ref}	Supply pressure reference	[Pa]
Q	Volume flow	[m^3/s]
Q_{est}	Volume flow estimate	[m^3/s]
T	Temperature	[°C]
t	Time	[s]
ts	Sample time	[s]
s	Laplace s	[-]
uA,, uB, uT	Control signals of port valves of the DHPMS	[bool]
u_{DVS}	Control signal array of the DVS	[]
u_{prop}	Control signal of proportional valve	[-]
uPA, uPB, uBA, uBT	Control signals of DFCUs	[bool]

v	Cylinder velocity	[m/s]
V	Volume	[m^3]
V_{A0}	Dead volume of cylinder chamber A	[m^3]
v_{cst}	Output of high level control system, reference to core level control system.	[m/s]
V_{disp}	Displacement of DHPMS piston	[m^3]
v_{est}	Cylinder velocity estimate	[m/s]
v_{ref}	Cylinder velocity reference	[m/s]
v_S	Minimum velocity of friction	[m/s]
x	Cylinder position	[m]
x_1, x_2	Valve flow model exponential	[-]
x_{ref}	Cylinder position reference	[m]
Greek alphabet		[]
Δ	Differential, eg. Δp = pressure differential	[-]
ω	Natural frequency	[rad/s]
ω_{min}	Lowest natural frequency of a system	[rad/s]
ξ	Damping factor	[-]
ξ_{req}	Required target damping factor	[-]
ξ_{sys}	Measured damping factor	[-]
τ	Time constant of a system	[s]

Abbreviations

CAD	Computer aided design
CCW	Counter clock wise
CW	Clock wise
DFCU	Digitan flow contol unit
DHPMS	Digital Hydraulic Power Management System
DOF	Degree of Freedom
DVS	Digital Valve System
EPC	Electro positive flow control
ICE	Internal combustion engine
LS	Load sensing
LUDV	Load independent flow sharing (orig. Germ)
LVDT	Linear variable differential transducer
MBC	Model based control
NFC	Negative flow control
NLS	Negative load sensing

1. INTRODUCTION

Gathering, harvesting and processing materials are the founding parts of a civilization. Materials that are gathered, carried and handled may appear in all phases: gases, liquids, or many types of solid materials like powder, grains, logs or boulders — to mention but a few. Different solutions to move and manipulate various solid materials have been developed. While conveyer belts or pipe lines are useful for some cases, arm-like booms with shovel or a gripper are used in some other operations. Especially solid raw materials like minerals or logs are often picked up by booms. An example of a forestry machine is presented in Figure 1.

Figure 1. *Example of a hydraulic boom on a modern forest harvester (at courtesy of Ponsse Ltd).*

Powertrains, or drivetrains as they are called especially when used for transmission, are classified under three main titles by the applied power transfer method: mechanical, hydraulic or electric. Mechanically operated arm-like-booms have been replaced by hydraulic ones since the middle of the twentieth century. Electrical booms are used in robotics, where speed and precision are a prioritized requirement. Hydraulic actuators have a high power-weight ratio but outstanding torque-weight ratio, which is a valuable feature especially in mobile machinery. Purely mechanical booms are becoming obsolete.

The main parts of hydraulic powertrains are:

- Prime mover, which creates power to be used in the system. This is usually an electric motor or diesel engine.

- A pump, which transforms prime mover angular velocity and torque to pressure and fluid flow.

- Power transfer lines: pipes or hoses.

- Motion control and power regulation: Different valves are used for power routing or for compensation or also for safety functions. Hydraulic transformers are used to change pressure-flow ratio.

- Actuators, which transform hydraulic energy into mechanical work. Hydraulic motors produce angular velocity and torque, while hydraulic cylinders are for linear actions and force.

- A secondary power source and energy storage device is a component or subsystem which makes a system have a "hybrid" power train. For a hybrid the auxiliary source must also be actively controlled, because the same components which are used in hybrid drive trains as the power source/storage can be also used passively as dampeners.

The layouts of hydraulic circuits vary by size and complexity. A generally available state of the art system, a load sensing (LS) system, is presented in Figure 2.

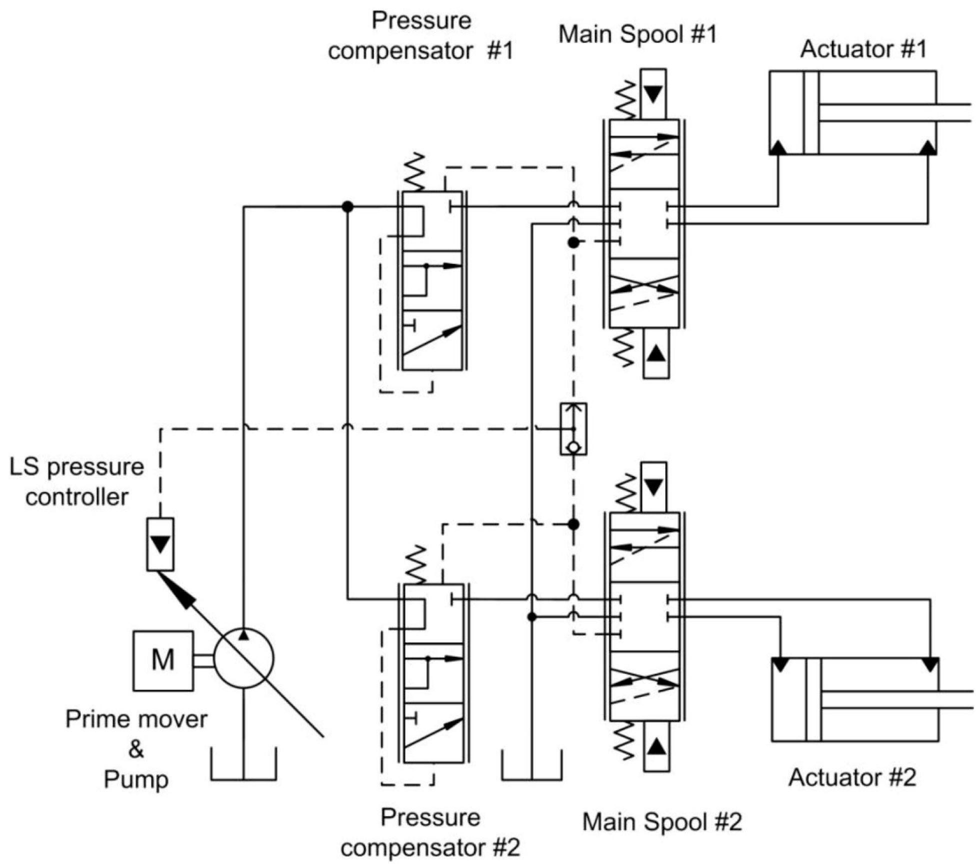

Figure 2. *State of the art hydraulic circuit for mobile cylinder drives*

In a LS-system the pressure is set by the demand of the highest load. The LS pressure is sensed from both actuators, but the shuttle valve selects the highest value. The LS pressure controller principle is to maintain such supply pressure whereby the pressure differential is kept constant over the main spools. This results in the supply pressure being the highest load pressure plus a certain offset. For an actuator which requires less pressure, the pressure is throttled down before the main spool by the pressure compensator. The compensator ensures that pressure differentials at the main spool inflow notches are load independent, which results in a certain control input always causing similar actuator velocity.

The forces and velocities at the actuators can have either positive or negative signs. That is, occasionally energy flow via a powertrain might be reversed. The following illustration, Figure 3, clarifies the possibilities.

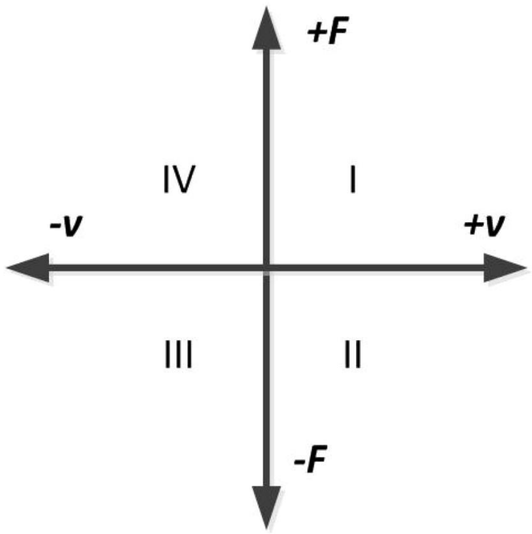

Figure 3. *Operation space and four quadrants. Because of the signs, multiplication produces positive input power to quadrants I and III and negative for II and IV.*

It is quite common that actuators operate in all of the quadrants, but the control valves cannot rout nor can the pump work with negative power (or both). In the LS-system negative power is dissipated by brake-throttling. Figure 4 and Figure 5 explain energy flow via power train in the LS system. Various sources of losses in a typical drivetrain are shown: Idle losses come from friction and leakage of a pump whereas transport losses come from viscous flow in pipes and hoses. Matching losses occur at the pressure compensators and control losses at the main spools. The main spools are also used for brake throttling if load holding valves are not present.

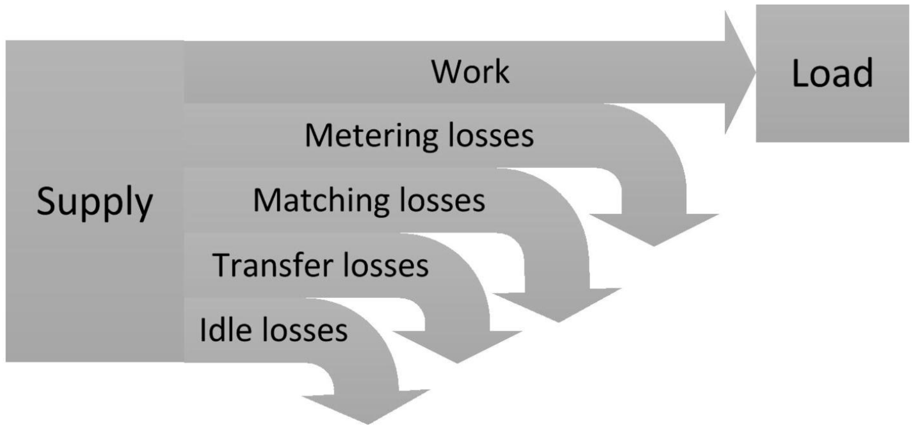

Figure 4. *Generalized power flow of LS-system during positive load.*

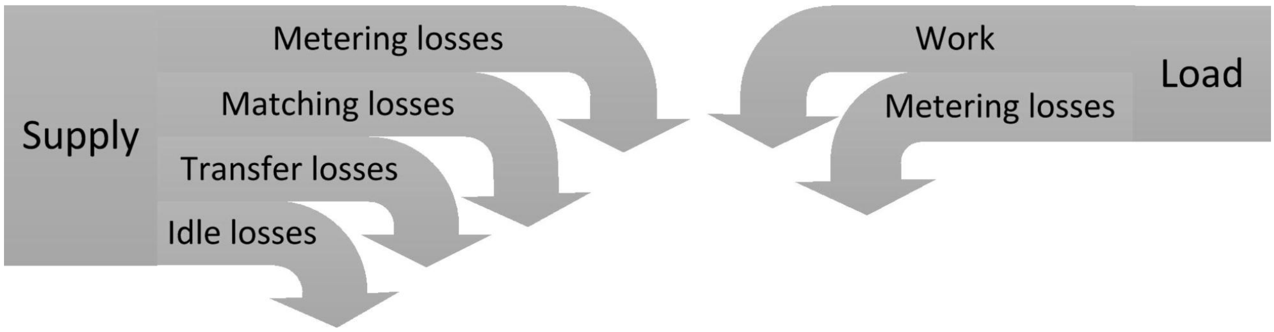

Figure 5. *Generalized power flow of LS-system during negative load*

Other systems and their ability to work in different quadrants are inspected in the "State of the art"-chapter.

1.1. DIGITAL HYDRAULICS

The umbrella term "digital hydraulics" covers a field of hydraulics in which the idea of digitalization is applied to the field of hydraulics (Linjama & Vilenius 2007). Discretizing components and replacing continuously variable (and often mechanically complex) components by intelligently controlled arrays of binary valued base units has been gaining interest and due to this development good results have emerged. A digital hydraulic approach requires electrical pressure measurements and fly-by-wire type controls. While digitalization enables enhanced freedom in the design of control systems, it simultaneously requires multidisciplinary skills from engineers. Knowledge of electronics and programming are required together with traditional hydraulic and/or mechanical engineering.

Digital hydraulics covers basically all hydraulic equipment: valves, pumps, motors, accumulators. Figure 6 shows possibilities. The figure includes analogue and digital versions of:

a) Control edge: Analogue control edge is a 2/2 proportional valve. Digital Flow Control Unit, a.k.a. DFCU consists of parallel connected on/off-valves. The amount of obtainable flow rates depends on the sizing of the valves. If equally sized, the number of possible flow rates equals the number of valves. If sizing follows binary code, then the number of possible flow rates is equal to two powers of the number of valves. Combining four DFCUs a four way valve can be realized, which results in Digital Valve System a.k.a. DVS.

b) Pump: Variable displacement pumps are available, but discretely variable flows can be obtained with parallel connected pumps with by-pass valves. The number of possible flow rates depends on the relative sizing of the pumps, similar to the case with the valve flow rates.

c) The power management system can pump to or motor from its ports or transform power between the ports in case one is motoring while one is pumping. A power management system can be realized with parallel connected analogue over center pumps but with the circuit presented a Digital Hydraulic Power Management System a.k.a. DHPMS is achievable.

d) Variable accumulator is an energy source in which pressure and flow are not coupled as they are in normal gas loaded accumulators. A somewhat variable accumulator can be obtained with analogue transformer and traditional accumulator. The digital counterpart utilizes binary coded annular areas which can be connected either to the system or to a low pressure (tank) line.

e) A variable displacement cylinder is mechanically impossible. However, with a transformer attached to the traditional cylinder it could behave like a variable displacement cylinder. Discretely variable displacement cylinders, however, are on the market and they can be used, e.g. as secondary controlled linear actuators.

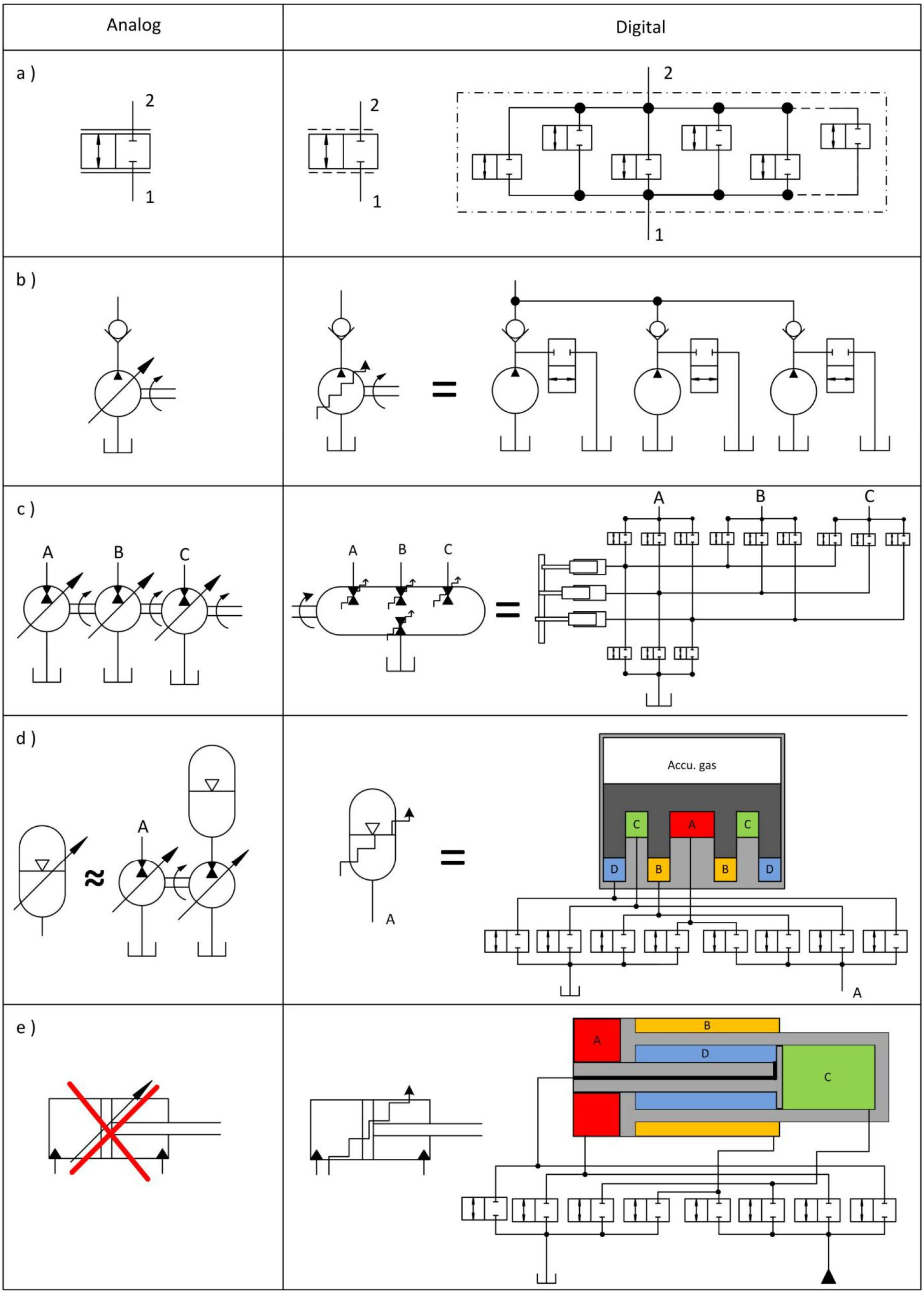

Figure 6. *Examples of digital and analogue hydraulics. a) control notch, b) pump, c) power management system, d) accumulator and e) cylinder*

Comparing the effects of digitalization in hydraulics with the effects seen in the few past decades in electronics might be too optimistic. However, camless ICE is good example of transferring programmability from hardware to software. In the case of an ICE with “virtual cam”, online changeable parameters enable optimal operation in a wider operation range than in case of traditional stiff mechanical control. Similar benefits are obtainable with the digitalization of hydraulics.

2. STATE OF THE ART

The field of the thesis is efficient power management, which is a wide one: it covers roughly everything. Classification into subgroups depending on the complexity of the valve- and pump- system is done. Division is made in the following way and the fields are explained later on (Figure 7):

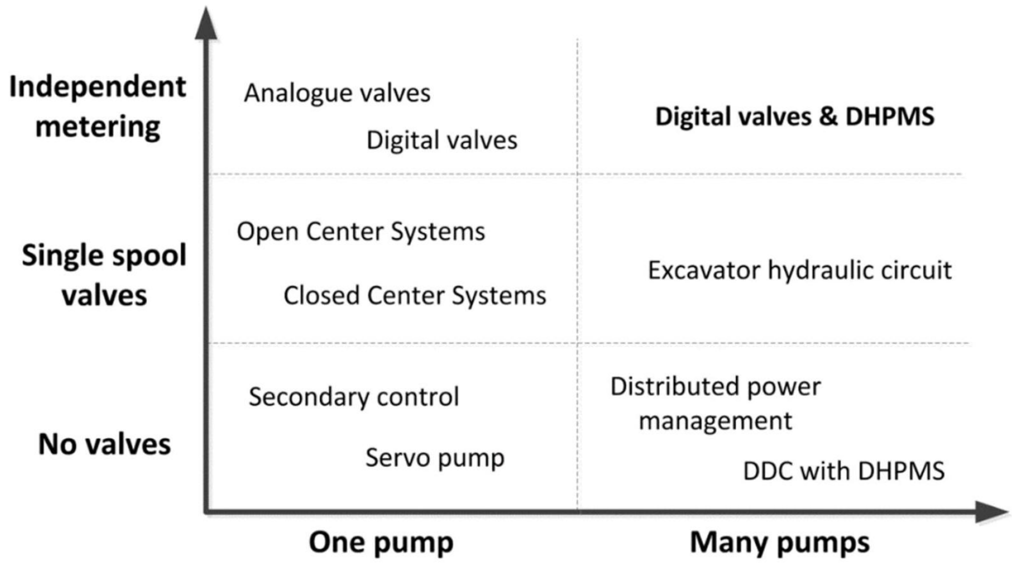

Figure 7. *Mapping hydraulic systems.*

The valves are divided into three groups: ‘No control valves’, ‘single spool control valves’ and ‘independent metering control valves’. A system can include either one or many pumps. In machines with hydrostatic transmission there are often, if not always, independent pumps for transmission and work hydraulics. In this study these are not considered as ‘multi-pump systems’. This is due to the fact that they just contain two independent single-pump circuits, although they share the same prime mover.

2.1. VALVES AND VALVE SYSTEMS

Valves with fixed opening ratio can be optimized only to one loading case per direction. In other loading situations either efficiency, controllability or both will suffer. In order to have an optimal valve during changing load conditions, independent metering (also called SMISMO, Separate-Meter-In-Separate-Meter-Out) is required. Independent metering is studied, e.g. by Janson (Jansson & Palmberg 1990), Hu (Hu & Zhang 2002) and Huova (Huova 2015). Figure 8 illustrates the two variations of independent metering, the right side option of which is capable of both inflow-outflow and differential modes.

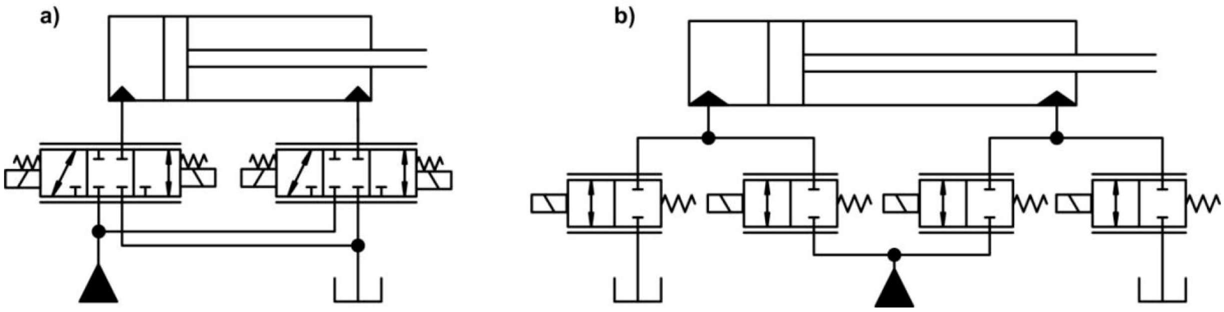

Figure 8. *Two possibilities for independent metering. a) Two mechanically linked notches. b) All notches individually controllable. The latter enables mode switching in software*

In the Vickers CMX valve in- and outflows are independent. Pressure side notches are mechanically connected but tank side notches are independent two way valves. The main stages are pilot operated. The Vickers CMX valve works in traditional LS-systems. Pressure compensation is realized in the main spool, taking advantage of flow forces. The Vickers CMX can be used with hydro-mechanical LS pump (Eaton 1995).

The newer Ultronics valve system has two independently controlled 3/3-proportional spools (for double acting actuator), which both can be controlled independently (case-a in Figure 8). The Ultronics valve has LVDT-transducers at the main spools and the actuator ports are equipped with pressure transducers, which are to be used for control functions, like electronic load sensing (Eaton 2010).

Total freedom of four notch control is available in the Incova valve system, where all four control notches are proportional 2/2-seat valves (case-b in Figure 8) assembled into a package containing integrated on-board electronics to control the opening ratio (Incova Techologies n.d.).

An approach of utilizing discrete valued control notches by using a series of parallel connected on/off-valves, DFCUs (Figure 9) is called a “Digital Hydraulic valve”, or more recently “a Digital Valve System”, DVS (Figure 10). The main idea is to use simple binary components and intelligent control algorithms to produce a digital counterpart to the analogue proportional- or servo valve. If the method is based on 2/2-valves, the digital valve has independent control notches by default (Linjama & Vilenius 2007). DVS can consist of any on/off-valves, but commercially available valves tend to lead physically big DVS’s. Prototype valves for more compact digital hydraulics have been developed: a bistable and bistable&impulse-actuated on/off-valve by Uusitalo et. al. (Uusitalo et al. 2007) & (Uusitalo et al. 2009) and monostable miniature needle valve (Karvonen et al. 2010). A later miniature valve has been redesigned and a laminated block for wide arrays of miniature valves merging to an equally coded DVS has been designed (Paloniitty et al. 2012). Customized control electronics for a wide array of miniature valves has been designed (Tiainen 2014) and integrated into a compact retrofittable package.

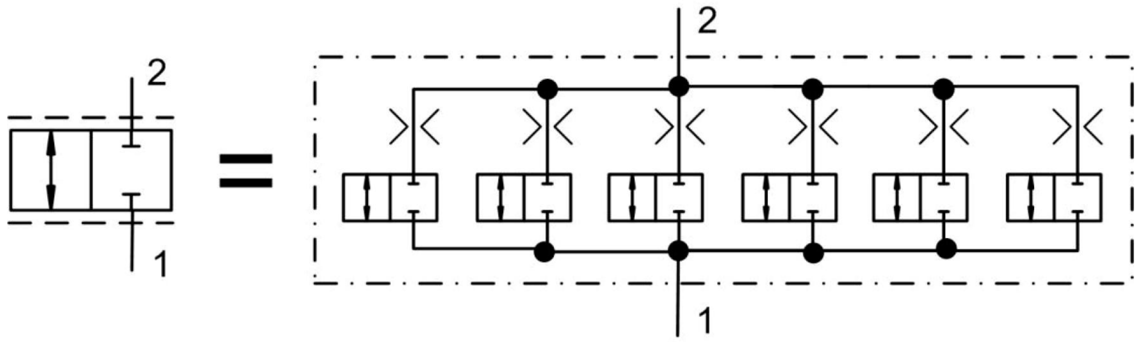

Figure 9. *Digital control notch, aka Digital Flow Control Unit, DFCU. Orifices are used to determine the coding of the valve, e.g. binary series can be used.*

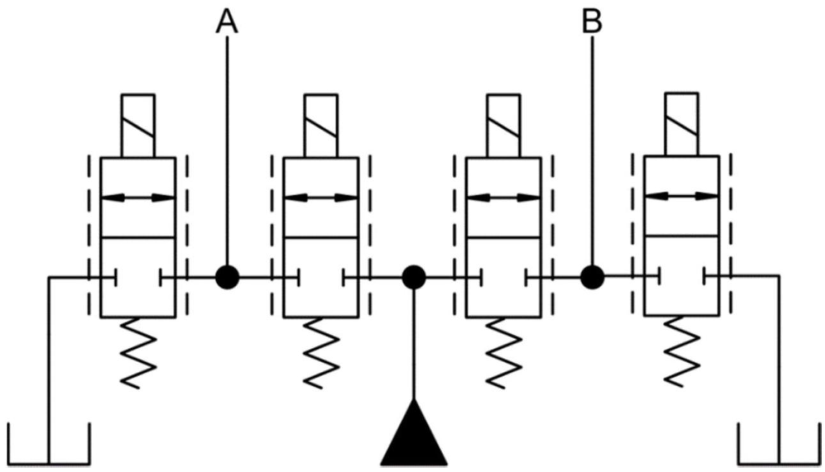

Figure 10. *Four DFCUs merged to a DVS*

Control of a DVS has been researched by Linjama and Huova (Linjama et al. 2007) and the results show that in a seesaw test bench losses were reduced by 40% compared to a traditional LS-valve. DVS has also been tested in the case of a wheel loader and the results show that losses were reduced by up to 63% compared to the traditional LS system (Huova et al. 2010).

In Bosch Rexroth SEC valve control notches are made of 2/2-seat valve elements, but independent metering is not possible due to the actuator principle. In SEC valve one coil opens both PA&BT flows while the other deals with PB and AT (Bosch Rexroth n.d.). BR SEC valve can be driven by ballistic PWM-control principle in order to achieve a reasonably proportional output (Flor, Scheller & Heidenfelder 2012). SEC valves are utilizable in digital hydraulic solutions as a replacement for a servo valve (Fischer et al. 2015). Another valve specially designed for digital hydraulics comes from Bucher Hydraulics (Bucher Hydraulics 2015).

2.1.1. OPEN CENTER VALVE SYSTEMS

In the heaviest, simplest or oldest machines, the most common work hydraulic circuit is based on open centre valves. An open centre system consists of 6/3-control valves in series so that actuations are prioritized (Figure 11). In these systems the load force affects the actuation velocity, giving the operator a feeling of the load forces. The heavier the load, the slower it moves at certain control stick position. The force feedback to the driver is practically useful, e.g. in mining operations, where the driver needs to estimate whether the load on a bucket consists of some loose rocks or maybe a corner of the bedrock. Also it is possible to control heavy loads with engine speed easily by tilting the control stick to its limit and then increasing actuator velocity by pressing on the throttle pedal. Thus, smooth motion is controlled with the valve and from the moment the stick reaches the limit, the system works like a pump controlled one. Based on the author's experience, this is a practical way to drive, e.g. a forklift. The drawback of this style is that high velocities require high diesel rpm, which typically is not the most efficient operation point for a diesel engine. From a driver's point of view this control style is simple, and it is learned fast to be used by instinct, which is essential for effective working.

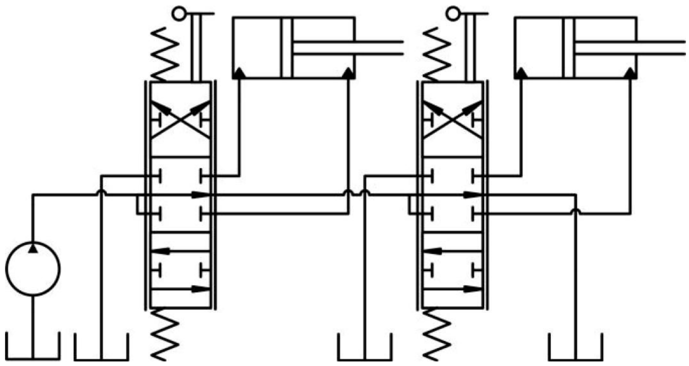

Figure 11. *Traditional Open-center system.*

The efficiency of the traditional open center system may peak up to the levels of displacement control while only one valve is completely open and the driver controls the velocity of the actuator with the gas pedal. The traditional open centre system utilizes a constant displacement pump and therefore the system wastes energy when a heavy load is moved slowly: System pressure is high and extra flow is throttled down to the tank pressure at the center PT-notch.

The losses caused by surplus flow can be reduced if a variable displacement pump with Negative Flow Control-method (NFC) is used. In NFC the pump displacement is reduced if the pressure drop over a fixed orifice at the end of the supply line rises. The losses are reduced as the amount of pumped oil is reduced (Linde Hydraulics n.d.). Negative flow control is used, e.g. in a Caterpillar excavator (Catepillar 2011).

More losses can be reduced if the end of the supply line is blocked after the last valve and pump displacement and valves are electronically controlled. The method is called Electro-Positive-flow-Control (EPC). The control code calculates control values for the valves and pump based on the electrically measured position of the control stick. Cetinkunt et. al. have achieved fuel savings of 7-15% by EPC compared to the traditional open centre system in a medium size wheel loader (Cetinkunt et al. 2004). A commercial product utilizing EPC is documented in web page references (Bosch Rexroth AG n.d.) and (Linde Hydraulics n.d.). Force feedback to the driver, typical of open center systems, is made possible by electronically measuring load pressures and using measured values in the control code. The higher the load pressure, the lower the actuator velocity at a certain control stick position. This method is called Virtual Bleed Off (Bosch Rexroth AG n.d.), which is based on utilization of the valve model in the control code. It is also mentioned in the source that by limiting the pressure and flow rise steepness in the software, diesel engine operation matches better to the required hydraulic power, which should be beneficial to the whole system efficiency.

2.1.2. CLOSED CENTER VALVE SYSTEMS

The closed center control valves are typically of the 4/3-type and they are installed in parallel. A generally accepted state of the art system consists of an LS pump, which produces the pressure level demanded by the highest load pressure. Actuator velocity is made load independent by pressure compensators which maintain the pressure differential over the main spool constant. A Load Sensing (LS) system with proportional valves can be considered to be widely available standard technology for mobile work hydraulics.

The new version of the LS-system has an improvement compared to the standard version: it works logically also when pump output flow is saturated. In a standard version, saturation of flow would cause a situation in which the actuator having the highest load would stop moving. Load Independent Flow Distribution (known also as LUDV. germ.) utilizes pressure compensators after the main spool and all of the compensators use the highest LS-pressure as a compensation signal. This system works so that the flow is always shared to the consumers by fractions depending on the relations of the control signals. The pressure compensators are springless and their throttle depends on load pressure, LS-pressure and main spool position.

Electric load sensing, ELS, utilizes Electronic Displacement Control (EDC) rather than traditional hydro-mechanical pQ-controllers. The control is based on software and the control algorithm may take various inputs instead of only LS-pressure: all pressures, spool positions, pressure compensator positions and control stick position (Djurovic & Helduser 2004). Compared to the traditional LS, ELS enables the possibility of better dynamic behaviour and more flexible control functions and pressure filtering (Luomaranta 1999). Electric displacement control can also be used for LUDV-systems. Compared with the electric pump controller in the LUDV system, it is possible to decrease losses and improve dynamic characteristics (Finzel & Helduser 2008).

Negative Load Sensing (NLS) has been presented in (Erkkilä, Lehto & Virvalo 2009). NLS is based on independent metering (Figure 12). The main idea of the method is to use the outflow control notch for velocity control, and if necessary the inflow control notch is used for pressure compensation. The measured results give signs that the technology has better controllability and smaller losses than the traditional LS system.

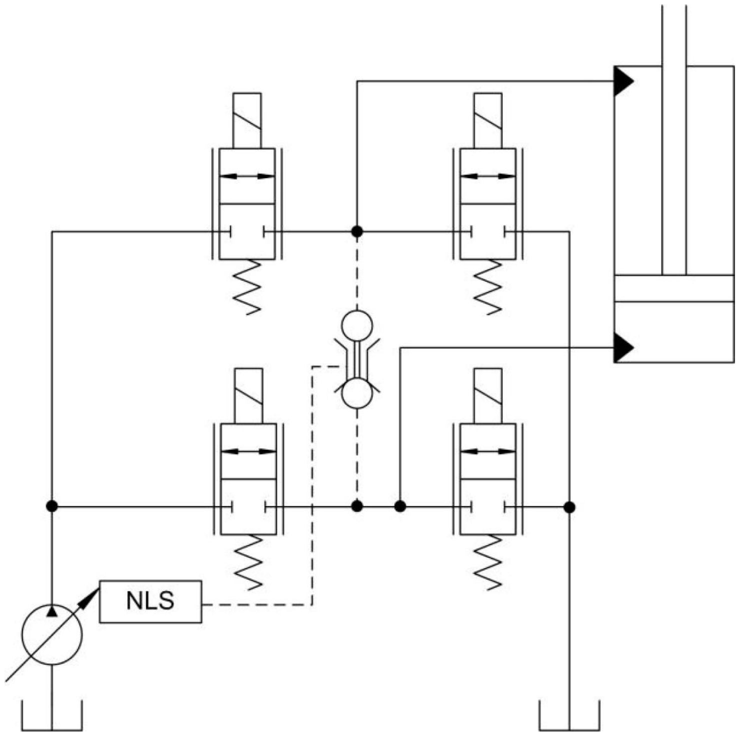

Figure 12. *NLS circuit diagram for one actuator. The middle valve selecting the NLS-pressure is a kind of inverse of a priority valve, but the check-valves are mechanically connected.*

2.2. VALVELESS SYSTEMS

In displacement controlled systems throttling is not used to control routed power, which is why these systems are also considered “valveless”. In the case of displacement controlled systems efficiency is at the same peak level, which is obtainable also with other power transfer methods, mechanical and electrical (Backé 1995).

Displacement control is generally used in hydrostatic transmissions and also in the newest cooling fan drives (Gandrud n.d.), (Bosch Rexroth AG n.d.). Variable speed fan drives are used to optimize combustion engine operation temperature to lower *NOx* emissions and match Tier 4 regulations.

Displacement control in cylinder drive systems can be based on, e.g. the following basic ideas, as presented in Figure 13. The problem with a common single rod cylinder lies in the fact of unequal in- and outflows. Different methods to produce pump controlled cylinder drive in the case of a single rod cylinder are presented.

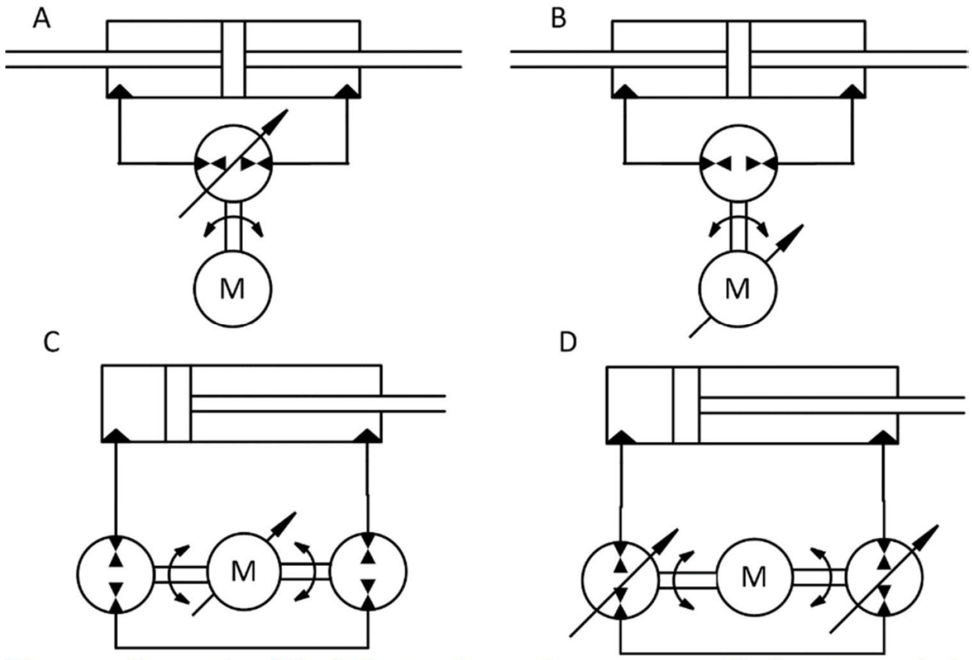

Figure 13. *Some simplified illustrations of pump controlled systems. A single rod cylinder requires two units as the inflow and outflow are unequal.*

The use of two constant displacement pumps driven by AC motors with frequency converters have been studied in (Long, Neubert & Helduser 2003) and the circuit is presented in Figure 14. High dynamic response is reported together with good efficiency. The control system consists of both position- and pressure feedback control. The system, however, requires a state-feedback controller for good stability and leakage of the pumps has been reported to cause a negative effect on performance.

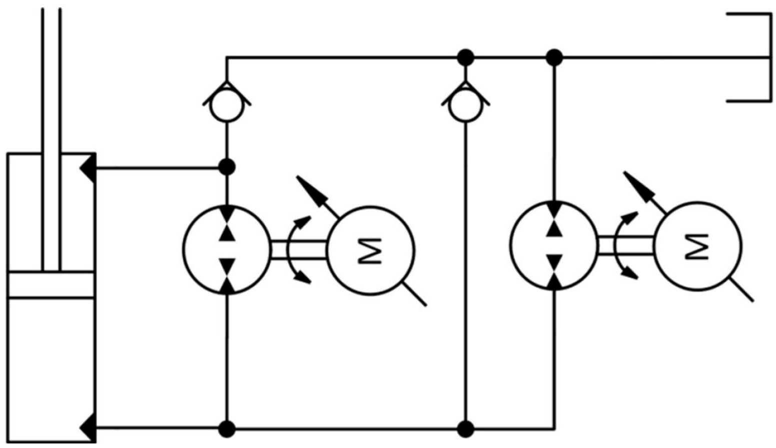

Figure 14. *One possibility to perform pump controlled cylinder drive (Long, Neubert & Helduser 2003)*

An approach based on case C in Figure 13 was also studied by Minav et. al. (Minav, Sainio & Pietola 2014). The approach requires that displacements of the pump/motors are matched to cylinder areas. In real systems matching cylinder areas to pump displacements is impossible because leakage of the pump is not constant and impossible to know accurately. Therefore in the reported system accumulators were attached to the system; one works as a “tank” and the other is attached to the rod side chamber line to balance in- and outflows. Minav’s system has been proven to work by measurements and the possibilities to measure system position from synchronous motor pulse count, and the pressures from motor current are mentioned as pros of the system.

Tikkanen et. al. have proposed a system based on servo motor and high pressure rail. The layout of the system is as follows (Figure 15): The system utilizes synchronous servomotor for velocity control; the rotation speed determines cylinder velocity. The hydraulic power supply line is a high pressure rail. The load pressure level relation to rail pressure determines whether the servomotor is using positive or negative torque. Cylinder power is the sum of the electrical power of the servomotor and hydraulic power to/from the rail. The benefits should lead to downsizing of the electric unit (Tikkanen & Tommila 2015).

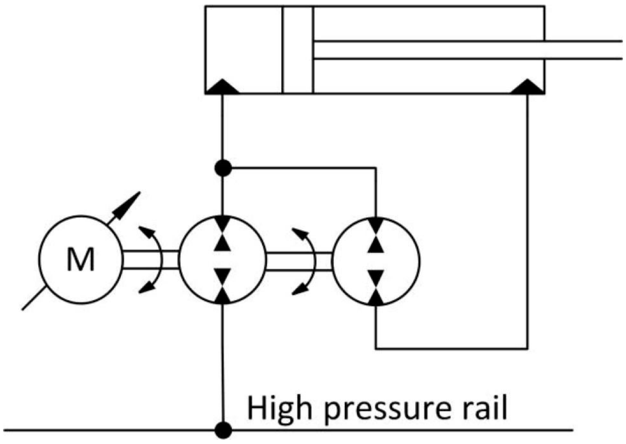

Figure 15. *Simplified schematic of system studied in (Tikkanen & Tommila 2015). The real measurement system has a feed pump to compensate leakage and flushing valve flow in a similar manner typical of closed circuit transmission hydraulics.*

Williamson et al.(2008) studied a small excavator in which Load Sensing hydraulics were replaced with pump control. Performance and efficiency were compared. The hydraulic circuit included one pump-motor for each eight actuators in a way that during digging all needed four actuators: swing, boom, stick and bucket had their own pump-motors. The results showed that 39% less power was needed in a certain load cycle and it consisted of the removal of both matching and metering losses.

Valveless systems also include secondary controlled systems. Secondary displacement control in the case of cylinder drives is a new area of interest. A multi-chamber cylinder works as discrete variable cylinder with arrays of digital logic valves and with intelligent control (Linjama et al. 2009). Diagram of the principle is presented in Figure 6-e. The principle has resulted in the commercial product name NorrDigi™ (Sipola, Mäkitalo & Hautamäki 2012). The topic has also been studied in (Dell'Amigo et al. 2013). A wave power plant has been reported to be one where digital cylinders could be applied. A test of suitability has been carried out (Hansen et al. 2014).

Displacement control is possible with the DHPMS so that the ports of the machine are directly connected to the actuators. This is called Direct Displacement Control, DDC.

Results show good efficiency and open loop tracking. (Heikkilä et al. 2014). The DHPMS ports provide individual pressure and flow and may therefore be used to provide independently controlled ELS-supply lines for various solutions. The machine works as a pump, motor and transformer so that energy can be moved to any direction between the prime mover axle and hydraulic connections. If one port is connected to energy storage the DHPMS works as a hearth of a hybrid circuit. Some possibilities of a DHPMS are illustrated in Figure 16.

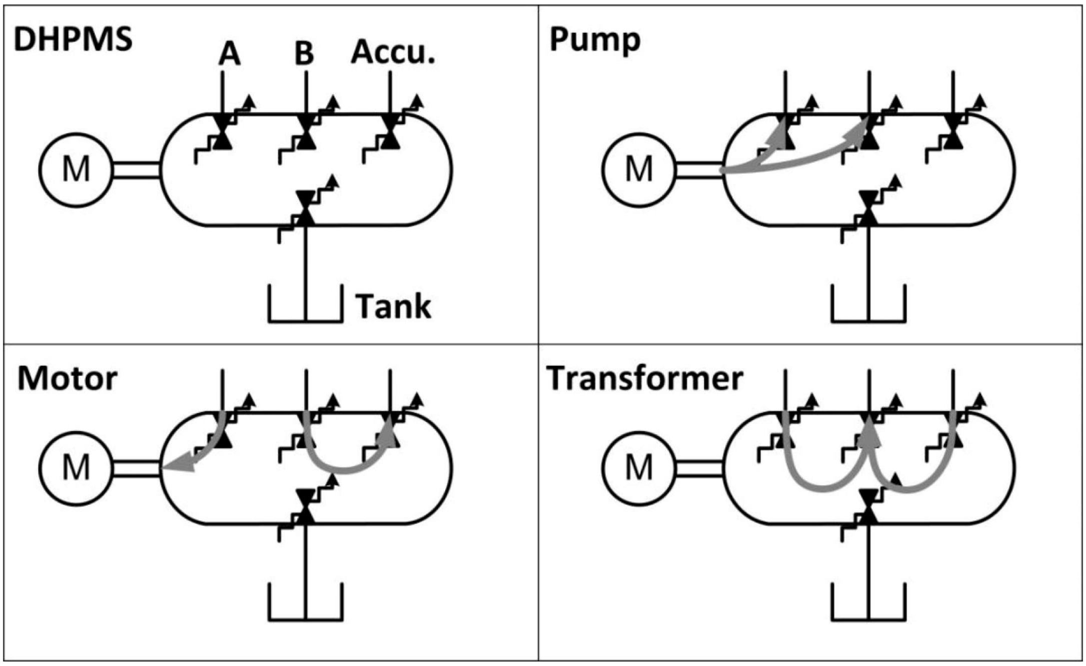

Figure 16. *Energy can flow through DHPMS to and from any lines in any direction during different operation modes. If an accumulator is used, the DHPMS works as a hearth of a hybrid circuit.*

Digital displacement pumps and motors have also been studied in the case of a large scale rotary machine: a wind turbine. A nacelle transmission system from Artemis (Artemis Intelligent Power Ltd. 2012) is based on a digital displacement pump and motor which are used as a hydraulic gear between plant rotor and synchronous generator. A retrofitted digital displacement gearbox in a 2.4 MW wind turbine has been in use since 2013 near Yokohama, Japan.

2.2.1. MULTI PUMP SYSTEMS

Two pump systems are present in almost all tracked excavators. The Caterpillar 336 hydraulic system is based on NFC technology, consisting of two parallel hydraulic circuits with net hydraulic power limiting the hydro-mechanical controller for a two-NFC-pump system (Catepillar 2011). Parallel work hydraulic line duties overlap, so that e.g. the boom can be actuated by both lines, but the bucket cannot and travels, left and right, are in independent lines. The reason may be related to prioritization, which would lead to a quite efficient system if the control functions select circuits for actuators so that matching losses are minimized. In that sense the system pressure levels would work best in the two-actuator-case as two independent supply pressures, but this can be verified only from the control codes which are not available.

Modern forest harvesters also utilize more than one pump and two schematics are inspected. In Ponsse Ergo (Ponsse Ltd. 2014) the work hydraulic circuit is based on the hydro-mechanical LS principle and only one pump is used for all boom functions. Other pumps are used, but they are for independent functions like transmission, harvester head and for auxiliary devices. Therefore that cannot be considered interesting as a multi-pump example. However, John Deere 1270E IT (John Deere n.d.) utilises an open circuit pump also in the transmission circuit, which enables it to be used for other applications. The duties between the two pumps are as follows: While driving, pump#1 supplies transmission and pump#2 can be used for the boom and harvester head. While standing still, pump #1 supplies the harvester head and pump#2 supplies the boom. Knowing that the power source (Diesel engine) peak power is reached when the harvester head chain saw is used, this solution seems quite reasonable. The two pumps probably have different supply pressures, which then have an impact on compensator losses.

In the heaviest excavators reliability demand is high, and therefore even diesel engines may be duplicated. This is the case in one machine studied (Ivantysyn & Weber 2014). The 1 MW reference machine described has four pumps for each of two diesel engines. The paper studies the utilization of displacement control for reduced losses. The paper presents that 35% losses could be reduced with a new system.

An array of pumps with logic valves merge into a digital pump. The idea has been presented, analysed and simulated in (Linjama & Tammisto 2009). Also simulations and analysis have been performed in (Locateli et al. 2014) and a digital pump with digital cylinder has been measured and presented in (Niemi-Pynttäri et al. 2014).

2.2.2. POWER SOURCE EFFICIENCY OPTIMIZATION

Research on the power management system controlling a diesel engine and hydraulic transmission has been extensive. Optimization can be achieved by changing the transmission ratio of the gear or/and by applying hybrid technology. Drive-by-wire technology is required, since the throttle pedal does not meter fuel intake but just gives velocity reference for the high level vehicle controller. Hybridization is also used to increase the prime mover efficiency. This is done by balancing the load, which occasionally leads to downsizing. Electric hybrid cars are available from various manufacturers. Hydraulic hybrid cars are not yet on the market, but have been studied (PSA Peugeot Citroën n.d.), (INNAS n.d.).

Power split in hybrid drive lines or continuously variable transmission (CVT) are power management systems because their use aims to maintain the prime mover at the optimal operation point while serving variable output demand. CVT is used also in tractor power take off (PTO) (Gugel & Tarasinski 2009). A comparison of different kinds of drive line controls in the case of a 120 kW wheel loader has been made (Jähne et al. 2008): Case 1 was a constant engine speed, which is a typical case in tractors when PTO is used. In case 2 controllers measured the throttle, brake and inching pedal positions and controlled the engine speed and transmission rate together. In case 3 power management of both the drive line and working hydraulics was done so that the power reserve for the work hydraulics were not constant in case 3 (it was in case 2). The results showed 6-12 % lower energy consumption in advanced cases compared to the traditional case 1. The origin of the improvement lies in increased prime mover efficiency due to the optimized operation point.

The total efficiency of a system, consisting of diesel engine, hydraulic pump and one or more displacement controlled actuators, can be optimized by controlling the displacements of both hydraulic pumps and motors and also the rotational speed of the diesel engine. This kind of optimization of the driveline required efficiency charts of the parts of the drive train, and the controller is trying to keep the overall system efficiency at the highest possible level. HIL-simulations with this kind of system were carried out by (Ossyra & Ivantysonova 2004) and the results showed 8-33 % fuel savings, depending of the work cycle.

Simulations of system optimization were also presented by (Williamson & Ivantysonova 2010). However, in that case the actuators were four cylinders driven individually by over-center pumps. The displacements and the diesel engine rotational speed were optimized for typical excavator load cycles. The simulation results showed 16.5% reduced fuel consumption compared to the system having constant rotational speed and only varying pump/motor displacements, as was also the case in the previous reference (Williamson, Zimmerman & Ivantysynova 2008).

Simulated investigations of a valve controlled constant pressure system with an intermediate pressure rail have been carried out by Sgro (Sgro 2015). The system is meant for mobile applications, like excavator booms. Hydraulic energy recuperation is possible and improves efficiency, but the key target is to maintain ICE at the operation point of its highest efficiency for a holistically efficient system. Fuel savings of 20-25% are estimated.

A prototype wheel loader, the IHA-machine (Backas et al. 2011), has a hydrostatic transmission based on a pump with on board electronics and wheel hub motors. The work hydraulics circuit is powered from the same axle and a mooring pump is used there. The work hydraulic circuit consisting of two cylinder actuators is controlled by distributed digital valves and a sophisticated two actuator control code (Huova et al. 2010). The valve controller is a model-based one and it calculates optimal control for valves and required supply pressure. The supply pressure demand is related to the load force but also the valve control mode, which can be either inflow-outflow or differential. The over-center pump may work as a motor and negative load on the boom cylinders is routed back to the diesel engine shaft. In the IHA-machine there are no energy storing devices to store energies of the negative load. A model based estimate for power demand is calculated in the model-based controller and it is compared to the power available for the work hydraulic pump from the main shaft. If there is not enough power available, velocity references given by the fly-by-wire control stick are reduced, but both relative velocities are kept as commanded. The estimated power of the work hydraulic circuit is sent to the highest level machine controller, which sums the work hydraulics power demand with transmission power demand and calculates optimal control signal for the diesel engine.

A hybrid circuit is used for a cut-to-length harvester and the target is to reach significant downsizing on the prime mover. Without a hybrid circuit the diesel engine is matched by the power demand of the chainsaw (Einola & Aleksi 2015). Hybrid excavators are on the market. The Caterpillar utilizes hydraulic- (Caterpillar n.d.) and Hitachi (Hitachi n.d.) electric hybrid technology on an excavator swing actuator.

2.3. SUMMARY OF REVIEW

The most commonly used system in small and normal sized systems are single pump and single spool valves. Based on an older source (Vickers 1998), the vast majority of mobile hydraulic systems are open center valves with fixed displacement pumps. A newer edition of the same source claims that the majority of mobile work hydraulics are still based on open centre technology, but Load Sensing (LS) systems are gaining popularity (Eaton Corporation 2010). The state of the art throttling control of mobile systems is considered to be an LS system on the market.

In machines with hydrostatic transmission there are often, if not always, independent pumps for transmission and work hydraulics. In this study these are not considered as multi-pump systems, since they contain just two independent circuits, both having only one pump although they share the same prime movers.

Excavators utilize multi-pump systems for work hydraulics: two or even more. The biggest ones may also have two diesel engines. In forest machinery multi-pump systems are also present, but the idea is not to reduce compensator losses by the separation of pressure levels but to maximize productivity with the least number of components.

Based on the review, three key elements to improve systems are mentioned.

1. Independent metering

    - Independent metering by distributed valves are on the market, though they are seldom applied (or just not reported loudly).

    - Research on online switching between inflow-outflow and differential modes is tested at research institutes. There are challenges, but the technology is emerging.

    - Digital hydraulic distributed valve control has been proven to be a feasible method to utilize independent metering. They are also utilized in the newest paper machine solutions. There are also challenges but new technology is being developed.

2. Independent supply pressures

    - Multi-actuator systems having more than one LS-pump exist. However, the reason seems to be higher total flow capacity rather than reduction or even removal of compensation losses.

    - Large excavators have multiple pumps with two independent pressure lines with duplicated control valves. This kind of circuit is capable of acting as two-pressure system, but it is unclear whether control is preferred to minimize losses or maximize productivity (both are possible by making changes in the software).

    - A multi-actuator system with independent pump controlled actuators has a major flaw: if one pump is replaced with multiple ones but occasionally only one (or maybe two) actuators require all of the total flow, then it would be necessary to replace one big pump with, e.g. four big ones, which is unlikely to be a wise solution.

    - Transformers can be used to produce independent supply pressures for actuators.

    - The recently developed Digital Hydraulic Power Management System DHPMS can be used to produce independent supply pressures for ELS lines.

    - DHPMS is capable of producing its maximum flow to any of its supply ports, while also making it possible to share the maximum flow in demanded ratios to all consumers.

    - Motoring from one port while pumping to another while lines have different supply pressures requires transforming, this is also an integral feature of the DHPMS.

3. Hybrid power trains and intelligent power management

    - Hybridization is a widely studied area, and also commercial applications based on hydraulic hybrid powertrains are available.

    - A hybrid power train contains energy storage, from which peak power can be taken and where, e.g. braking energy can be stored.

- When the prime mover is sized to average power instead of maximum power, it is assumable that the engine stays more often at the most efficient operation point.

Based on the review novelty of the conducted study is proven and the system and the results which are presented in following chapters are clearly pioneering work.

3. SCOPE

In the state-of-the-art system (LS) there is a single supply pressure for multiple actuators resulting in the pressure level being matched only for the actuator currently carrying the highest load. For load-independent controllability the supply pressure is throttled down in pressure compensators (matching) to the level which results in the same pressure differential over all main spools. LS systems have single spool valves which are optimized in just one loading condition, which causes losses during other loads.

A digital hydraulic valve system (DVS, Figure 10) is an independent metering valve realized by digital control edges consisting of parallel connected on/off-valves. A DVS consists of both hardware but also of sophisticated control and it is one alternative of approaching for a “perfect valve”, which may optimize its behaviour constantly. In comparison, the spool geometry is normally in hardware and cannot be changed during operation. In digital valve this can be done but step wisely. As comparison to analogue distributed valve system, a DVS may change its state without transients enabling online mode switching capability. The DVS hardware used for the study is closely similar than already on the market industrial proof versions.

A digital hydraulic power management system (DHPMS, Figure 16) is realization of an ideal hydraulic power sink/source. The DHPMS consist of parallel displacement volumes which in- and outflows are controlled by active port valves. The machine may provide required power independently to the power lines in required pressure and flow ratio and also manage power flow and transformation whenever power is transferred from a line to another. The prototype of the DHPMS used in this study is laboratory proof and good enough for concept testing.

Matching losses, which are feature of multi actuator systems, depend on the difference of pressure levels and flow through the pressure compensator. In two actuator system, if both actuators require the same pressure level, matching losses do not occur. On the contrary, high matching losses occur when big flow rates are heavily throttled down from high pressure. Figure 17 illustrates the case. On the Y-axis there is pressure (~force) and on the X-axis there is flow (~velocity), resulting in the marked areas representing power. In Figure 17 it is assumed that the compensator is fully open (= no throttle) if the pressure level is correctly set for the actuator. Clearly, matching losses are case sensitive depending on the work cycle. However, the potential for energy savings is clear as matching always causes losses; the amount varies in different cases.

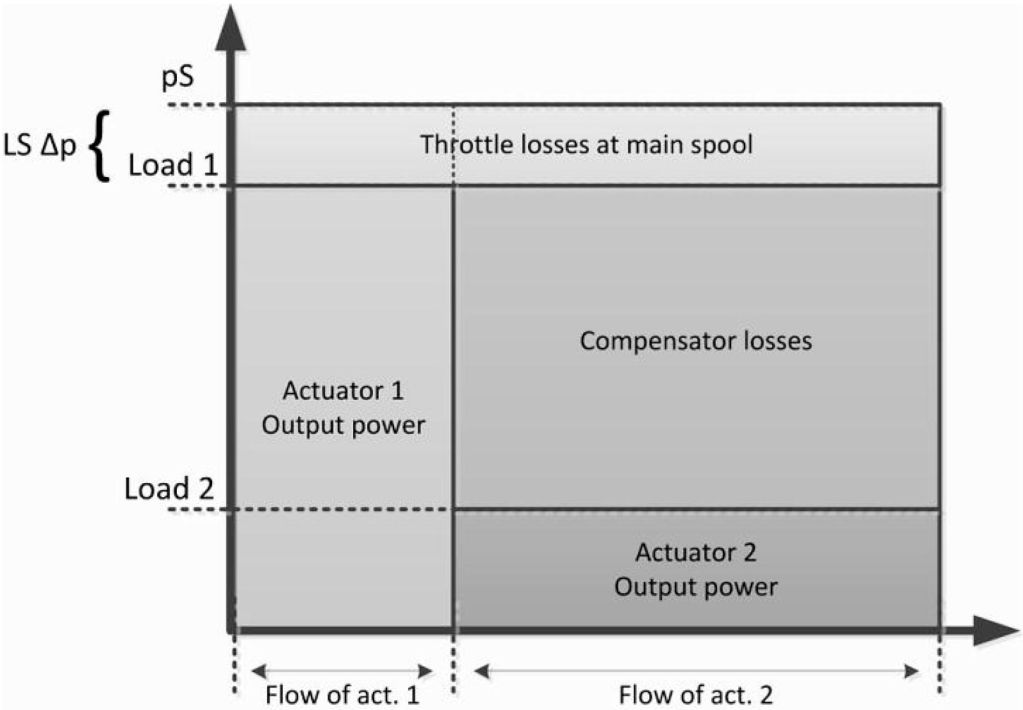

Figure 17. *General power consumption in the case of one supply pressure. Losses at pressure compensator are matching losses.*

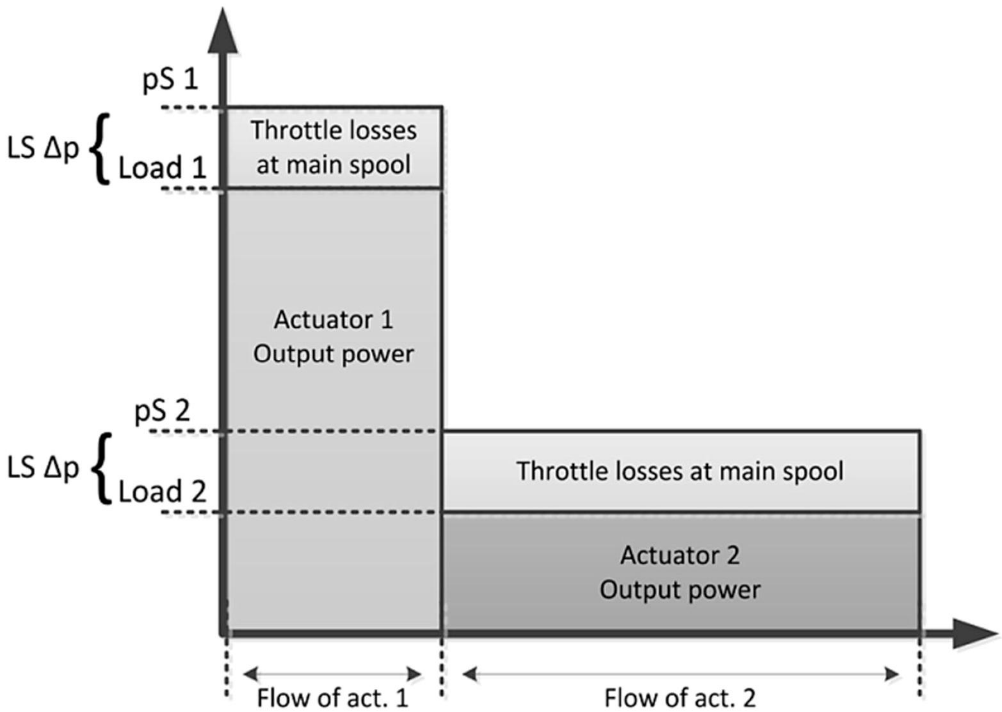

Figure 18. *General power consumption in the case of independent supply pressures.*

When all of the four control notches are independently controllable, the valve is capable of changing the operation mode from inflow/outflow to differential mode, or vice versa. Some of these modes are regenerative. Being able to utilize different operation modes does not necessarily require digital valves, but changing the mode during operation has so far been done only with digital hydraulic valves. The moment of mode switching requires rapid changing of the valve states and system pressure and flow. DVS full amplitude response time is one controller time step if the sample time is set longer than the valve response time.

The digital independent metering valve may switch mode online, but from the supply unit this requires high dynamics as the flow rate and pressure should change stepwise. This is possible with the DHPMS, since it can change its flow rate from one to any other during half a pump revolution. While running on 750 rpm full amplitude response time is therefore 40 ms (from full pumping to full motoring). The DHPMS can motor negative power even while pumping to another line.

Digital valves are controlled with a model based controller, and it produces good estimates for the system states. These estimates are useful for DHPMS pressure control algorithms. Conversely, the DHPMS has outstanding pressure control dynamics, which are required during instants when the DVS changes its operation mode between inflow-outflow and differential modes. With recuperating modes the DHPMS is capable of transferring energy motoring from one supply line while pumping to another.

3.1. AIMS AND METHODS

The aim of the thesis is to design a holistic control system for a multi-actuator plant using DHPMS as a hydraulic power sink/source and DVSs for motion control and compare it to the state of the art system. The reference system is LS system which consists of single supply pressure and proportional valves.

The research questions are:

1. Is the DHPMS capable to work as two port sink/source of hydraulic power which leads to the removal of matching losses in a multi-actuator system?

2. Which kind of control system is required to unleash the potential of the combinatory benefits of DHPMS and DVSs? What are these benefits?

3. How much losses can be reduced and why?

By simulations and measurements the following test cases are therefore required:

1. DHPMS with proportional valves working as a common pressure source for two actuators. This is also a base case which mimics the standard ELS system and is therefore the reference point of the thesis.

2. DHPMS used with proportional valves supplying independent pressure to two supply lines. Matching losses should be removed and the amount of energy saving potential to measured and analysed.

3. DHPMS used with digital valves as a common pressure source for two actuators. The effects of four notch control should be visible.

4. DHPMS used with digital valves supplying independent pressure to two actuators. The losses caused by pressure compensators should be removed and the effects of four notch control should also be visible.

5. Pressurized tank line together with digital hydraulic systems. Recuperating modes are likely to be more often possible with a pressurized tank line. (Simulations only, realization of pressurized tank line is under development)

For the test cases a trajectory or series of trajectories are required in which:

1. Actuator velocities vary by both direction and magnitude. Flow demands vary.

2. Actuator loads (pressure demand) vary by both direction and magnitude. Control modes should change accordingly.

3. There are different pressure levels for the actuator ports and varying load.

For the described purpose a two DOF boom is sufficient. One which consists of lift and tilt functions was found available for the research. When going back and forth with a tilt cylinder, it is possible to utilize all the available control modes, including the recuperating one. If also the lift-cylinder is moving during the recuperating mode, power transfer from a DHPMS port to another should occur.

To measure differences between different systems in the case of energy efficiency and controllability generally, the trajectory has to be somewhat universal. Possible trajectories are infinitely different. Therefore, a circular trajectory is selected only because that is a general case. During circular trajectory both actuators have both negative and positive velocities under varying loads. Also all four velocity combinations are obtainable.

3.2. CONTRIBUTION

Digital hydraulic valves are programmable, thus the functionality is in the program code. Base research about the control of a DVS is carried out by Linjama (Linjama & Vilenius 2007) and Huova (Huova 2015). Industrial pilot projects, related to the field of “Digital valve control” exist and have been reported by Laamanen and Fischer (Laamanen, Aaltonen & Linjama 2010), (Fischer et al. 2015). In those projects only some of the key benefits of the DVS are utilized, like repeatability and reliability rather than energy efficiency. The digital hydraulic power management system, the DHPMS, is also programmable and functionality depends on the control code. Measurements show that the DHPMS works as a pump, a motor and a transformer (Heikkilä et al. 2010). In a multi actuator system, there is potential to improve efficiency by utilizing both DVS and

DHPMS technologies, if the system is properly controlled on the holistic level. Following Venn-diagram graphically illustrates area of contribution (Figure 19).

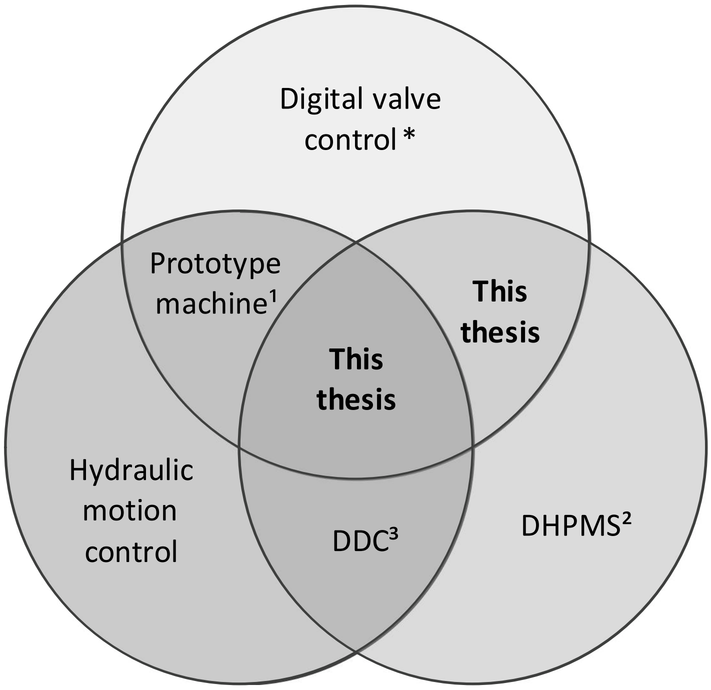

Figure 19. *Contribution chart with the most important references marked on the neighboring fields. References to marked fields are: * (Linjama et al. 2007), [1] (Huova et al. 2010) , [2] (Heikkilä et al. 2010), [3] (Heikkilä et al. 2014)*

The scientific contribution of the thesis is the analysis of the applicability of the DHPMS as a supply unit for a system having two separate supply pressure lines. Furthermore, it is to be shown by measurements, that when used together with independent metering valves the DHPMS can take power from the supply line whenever recuperating valve control is applied. Recuperated power is rerouted by the DHPMS either to another supply line or back to the prime mover and this is verified. The position feedback controller is applied together with DVS- and DHPMS-controllers. The holistic level control system, which merges them together, is designed. The energy efficiency improvement achieved by combined new technologies are measured, analyzed and compared to the state-of-the-art system.

4. SYSTEM

The test system hardware is a small back-hoe/excavator boom having two degrees of freedom (DOF). The test setup is assembled on a stiff platform in a heavy laboratory. The hydraulics of the system is based on closed center electrical load sensing technology. The laboratory has indoor conditions. The system diagram is presented in Figure 20. The proportional valve is isolated from the system when not used because of pilot flow and leakage. Digital valves do not require isolation because seat type valves are leak-free. Valves 11 and 12 are switched if single supply pressure is used instead of independent supply pressure.

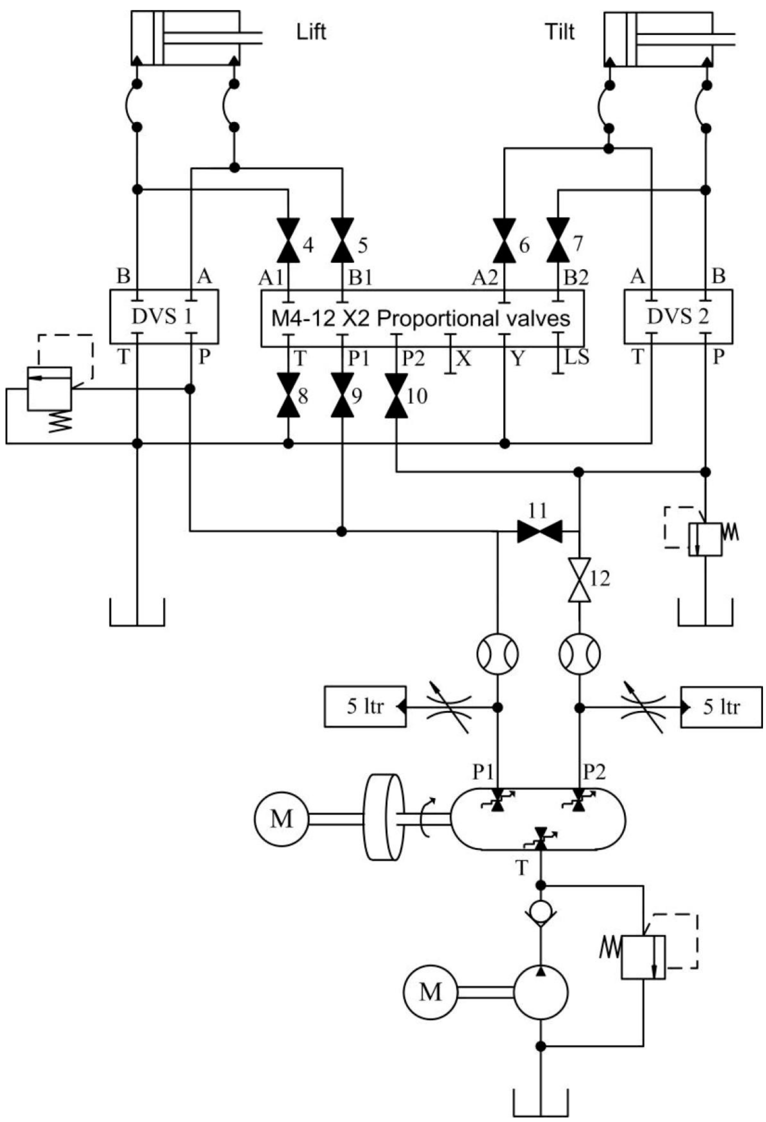

Figure 20. *Hydraulic diagram of the test system. The system includes two cylinder actuators and corresponding control valves. Either digital or proportional control valves can be selected. A DHPMS, which requires a feed pump, works as the hydraulic power supply. 5 litre damping volumes are used for smoother loads.*

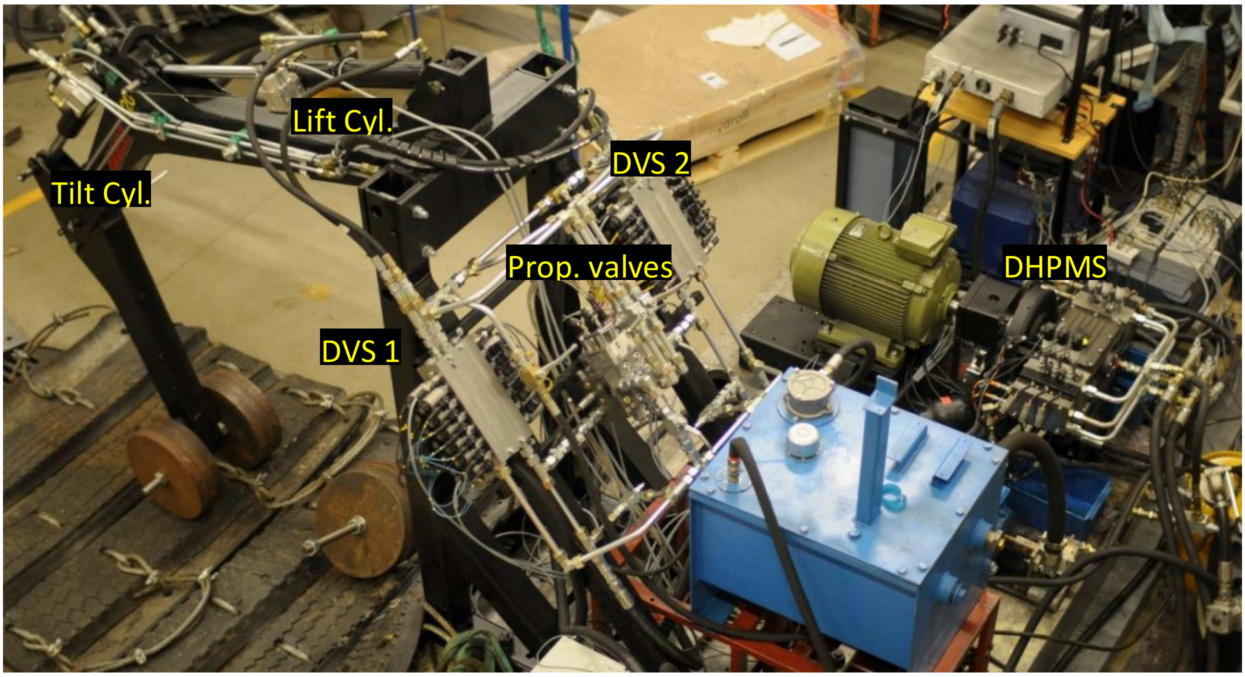

Figure 21. *Measurement setup*

4.1. THE DHPMS

The Digital Hydraulic Power Management System, DHPMS, is a pump-motor-transformer device having pistons connected to actuator lines through actively controlled 2/2 on/off-valves. The controller uses actuator line pressure references and shaft angle as inputs and controls the valves so that the DHPMS either pumps to, motors from, or transforms power between ports. Laboratory tests on the efficiency of the DHPMS are reported in (Tammisto et al. 2010). The efficiency of the component is virtually independent of pressure level. The leakage over the port valves causes volumetric efficiency to drop at partial flows. With proper valves this would not be a problem. Transformation efficiency is reported to be 80 %. Compared to traditional variable pumps with pressure control, DHPMS has outstanding dynamics (Heikkilä et al. 2010).

The prototype DHPMS is based on a boxer-type pump with six pistons. Normally, the pump valve plates contain cap manifolds containing two check valves for each piston. For the prototype these are replaced with custom made manifolds containing actively controllable on/off-valves. For each piston there is one valve for each connection port. The connection ports are for supply line A, supply line B and tank line. That is, both customized end caps contain nine valves. The valves are fast and have good flow capacity (~20 l/min @ 0.5 MPa), having both internal and external leakage.

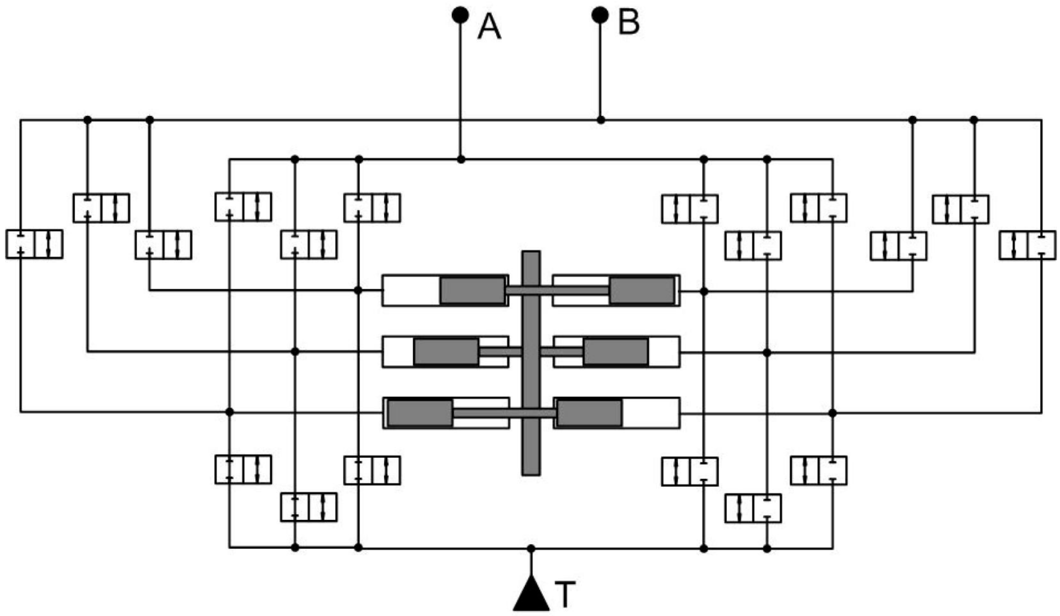

Figure 22. *Boxer type DHPMS schematic. The real machine contains cylinder wise pressure relief valves and instrumentation.*

The prototype DHPMS is capable of producing two independent supply pressures which can be used as ELS lines for proportional valves or for DVSs. However, a six-piston-machine, supplying two lines, may produce too much pressure pulsation so that it affects the actuator tracking performance, causing velocity ripple. The DVS controller utilizes software-based pressure compensation and the bandwidth depends on controller sample time and filtering. Hydro-mechanical pressure compensators which are integrated in proportional valves typically have a high bandwidth. The bandwidth of the control system is not broad enough to actively compensate DHPMS-caused pulsation of pressure and therefore possible visible effects are clearer with digital valves. This is to be analysed.

4.2. DIGITAL VALVES

Digital valves contain six on/off-valves per control edge. All the control edges are independent. The schematics of the digital valve were presented earlier (Figure 9 and Figure 10). Valves are normally closed type valves from Bosch Rexroth (KSDER-type). The return springs of the valves are changed to be stiffer to increase the repeatability of the closing times. Uncertain switching times increase the risk of major pressure peaks (Laamanen 2009). Serial orifices are installed in the DVS manifold. Coding is based on binary coding of the flow rates, but the biggest “bits” have almost the same flow rate.

The orifice diameters were decided by measuring the flow rates of various possibilities and initially selecting the closest match to the binary series. Verification measurements were done and the orifices were changed if any flow step between adjacent states greatly exceeded the flow step between states zero and one. The measurement method and flow series are presented later in section 5.4. All the DFCUs selected were identical.

The valves are driven with 4 ms 48 V boost voltage and 12 V hold voltage. The switch times of the valves were measured to be 10 – 15 ms. The differences between opening and closing times were compensated by delaying faster operation in the controller. This resulted in a reduction of the pressure peaks.

4.3. PROPORTIONAL VALVES

The proportional valves are Bosch Rexroth M4-12-X2 mobile proportional valves. Independent supply pressures are possible with the selected manifold end units. The spool opening ratios were selected by simulations; though obtaining an exact match was not possible in a realizable time-table. The proportional valves are driven with Rexroth Bodas RC-module, which in this case works only as voltage-current amplification. All the control codes are in the measurement computer, although some could be driven in the RC-module.

4.4. MEASUREMENT & CONTROL EQUIPMENT

The measurement setup consists of two independent measurement and control systems: one for the DHPMS and one for boom motion control. The upper level controller for motion control contains valve controllers and, among other outputs, it also provides pressure and flow references for the DHPMS controller. The DHPMS controller provides required pressure and flow to the supply lines. The following table presents a list of equipment:

Table 1. *Pieces of measurement setup*

Pieces of measurement setup	**Type**
Pressure transducers (supply line, chambers, tank)	Trafag NAH 250 bar
Supply line flow meters	Kracht Volutronic VC1
Oil temperature sensors	PT 100
Cylinder position (and velocity) encoders	Pepperl+Fuchs RVI58N
Torque and angular velocity sensor	HBM T40B
Motion controller board	dSpace DS1103
Motion controller time step	Meas. 1 ms, motion control 40 ms
DHPMS controller board	dSpace DS1005
DHPMS controller time step	Meas & Ctrl. 50 µs

5. CONTROL SYSTEM

The goal of the control system is to control the tip of the boom in Cartesian space. A closed loop trajectory is used so that different systems can be compared. The controller contains high level cylinder motion controls and core level valve controls. The DHPMS controller is in a separate measurement computer.

An overall picture of the control system is presented in Figure 23. The first block contains look-up-tables for tip position coordinates: x- and y-coordinates since the reach space is planar. Through inverse kinematics and trigonometry the references are transferred to cylinder position references. The upper level position control is a closed loop position controller and creates a control system velocity reference (*v_cst*). The *v_cst* can also be manually given from the control stick. Either a digital valve system (DVS) or proportional valves are used. DVS is controlled by a model based controller (MBC) and the proportional controller is a simpler one. Both valve controllers provide supply pressure reference (*p_ref*) and flow estimate (*Q_est*) in addition to actual valve command signals. Supply pressure references and flow estimates are used in DHPMS pressure control algorithms, but basically the *p_ref* is used in a closed loop and *Q_est* behaves like a feed forward. The valves react to the control signals and generate flow to the actuators, the positions of which are measured. The tip position depends on the actuator states.

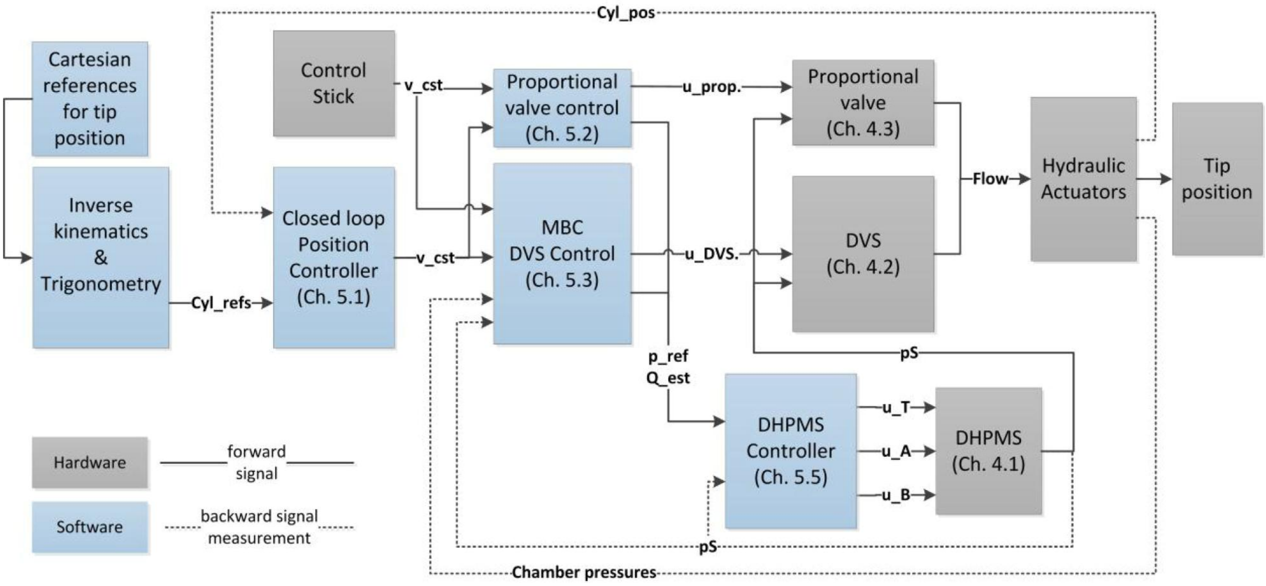

Figure 23. *Overall picture of the control system. The light grey blocks represent software and the darker grey ones hardware; the solid lines are forward signals (commands) and dotted lines backward signals (measurements & feedback). Different valves are complementary.*

5.1. UPPER LEVEL POSITION CONTROL

The upper level position controller consists of velocity feedforward and position feedback controllers. The reference for feedforward is a numerical derivative of position reference and a feedforward gain of 0.6 is used.

The system consists of two actuators, which are linked together by the boom structure. Actuation of one cylinder moves the load, which changes pressures also in the other cylinder. As the oil is compressible, also states of the other cylinder are affected. Clearly, the system is a Multi-Input-Multi-Output type (MIMO). In such a system actuation of one actuator is a disturbance to the other and thus robust control methods are applied. In (Linjama 1998) a robustly stable controller for a hydraulic multi actuator boom has been designed, resulting in a fairly simple control principle for a stable feedback system: P-control with first order lag with the pole at significantly lower frequency than the lowest system natural frequency.

The controller is designed in the following manner: Firstly, the lowest natural frequency of the system, from any input to any output, whichever it is, is measured. The lowest natural frequency varies as a function of tip position and the load. Therefore, the values were measured at various different operation points in the reach space with 200 kg load mass. Measurement was done by driving the boom manually with steps by using a control stick. Frequencies were inspected from the pressure signals. The lowest natural frequencies in the reach space are shown in Figure 24. The lowest value was found to be about 2.6 Hz (17 rad/sec).

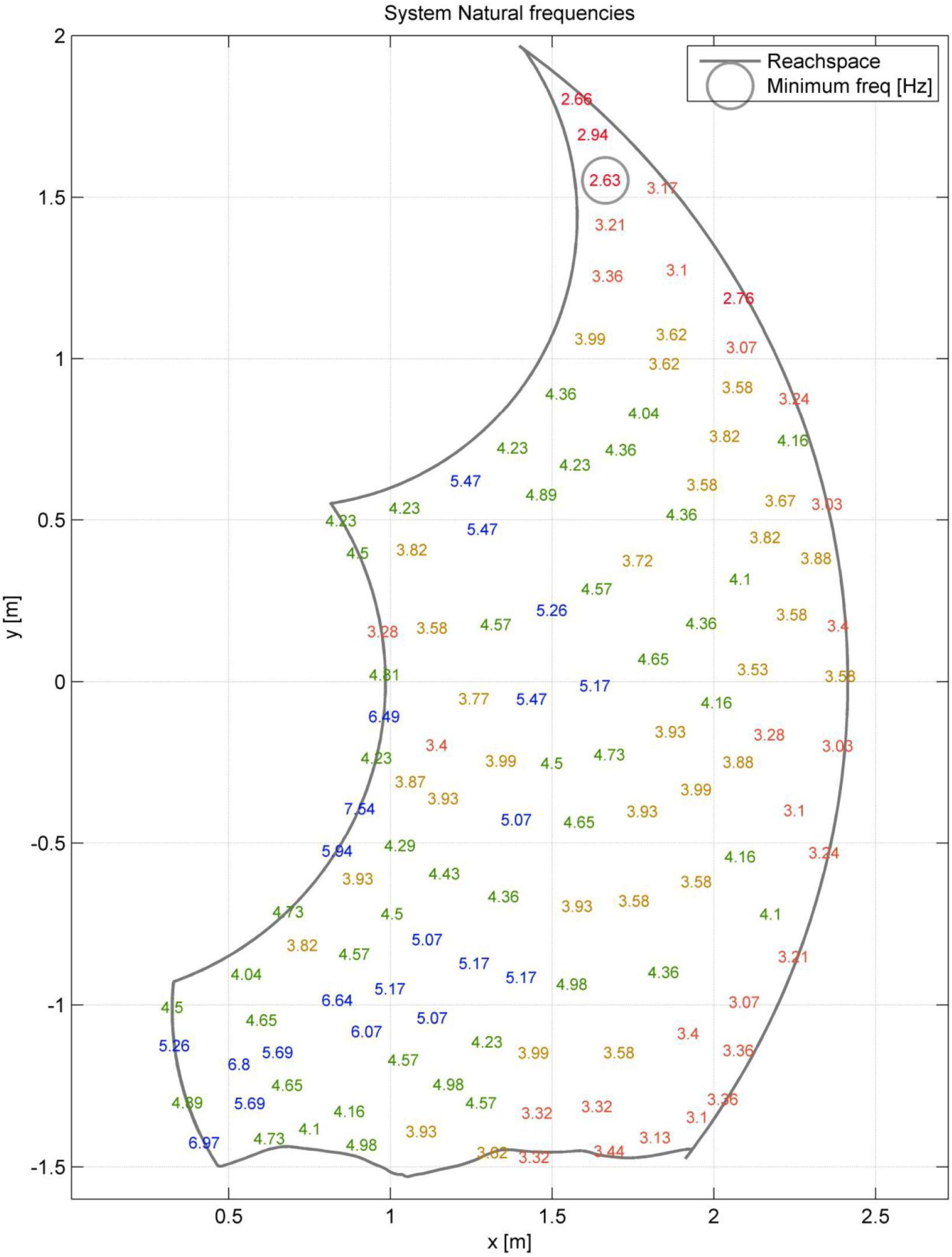

Figure 24. *Map of natural frequencies in Hz in the reach space, where the bottom level is limited by the floor. The lowest frequency marked with a circle. 200 kg*

The second phase of control design is to select a target damping for the closed loop system assuming the system behaves like a 2nd order system. A damping ratio of 0.7 was selected. Substitution of the target damping and the lowest natural frequency into equation eq.1 gives feedback controller gain, which is used for both controllers. In the equation ξ_{pref} stands for the preferred damping ratio, and ω_{min} is the lowest natural frequency.

$$K_P = 1/\left(4 \cdot \xi_{pref}^2 \cdot (2/\omega_{min})\right) \quad (1)$$

The substitution results in feedback controller proportional gain of 4.22. A block diagram of discrete filtered P-controller with velocity feed-forward is presented in Figure 25.

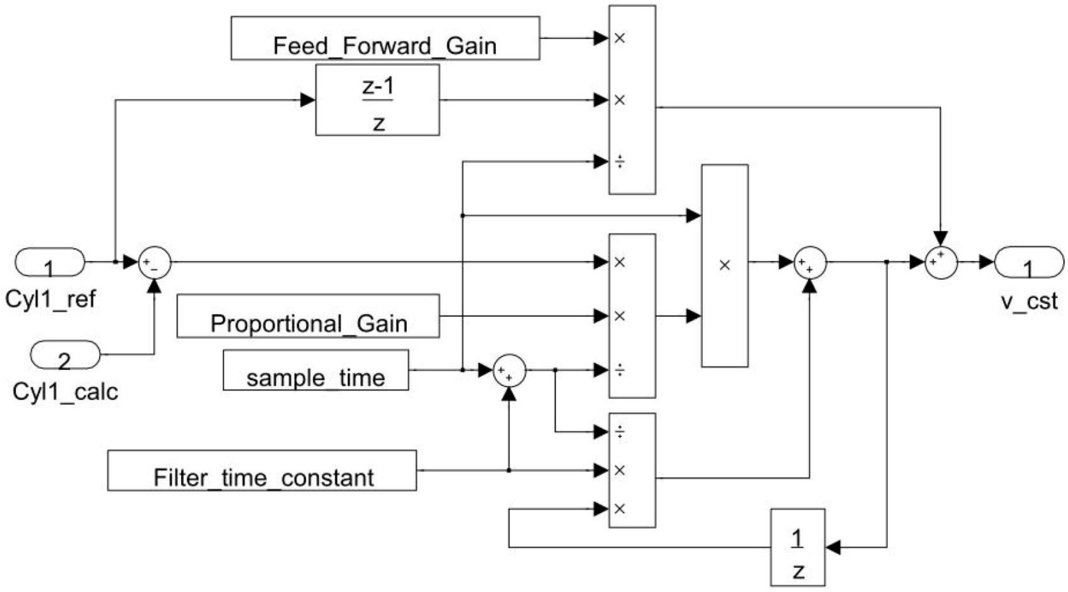

Figure 25. *High Level Position Controller: Discrete PT1 type with velocity-feed-forward. Discretization method: Tustin's approximation*

5.2. PROPORTIONAL VALVE CONTROLLER

The input for proportional valve controller is v_{CST} and output is the valve command voltage. The control law contains two-dimensional charts which are used to map valve commands and corresponding actuator velocities. Dependency between the signal and response are measured in-situ in a practical manner instead of by clinical measurements in a separate valve test bench. Roughly the proportional valve controller contains the inverse of the spool geometry, which can be considered to be a simple model based control method, but no internal model estimates are calculated. Flow estimate in case of proportional control is purely based on v_{CST} and cylinder dimensions. The dependencies between control signals and cylinder velocities are presented in Figure 26.

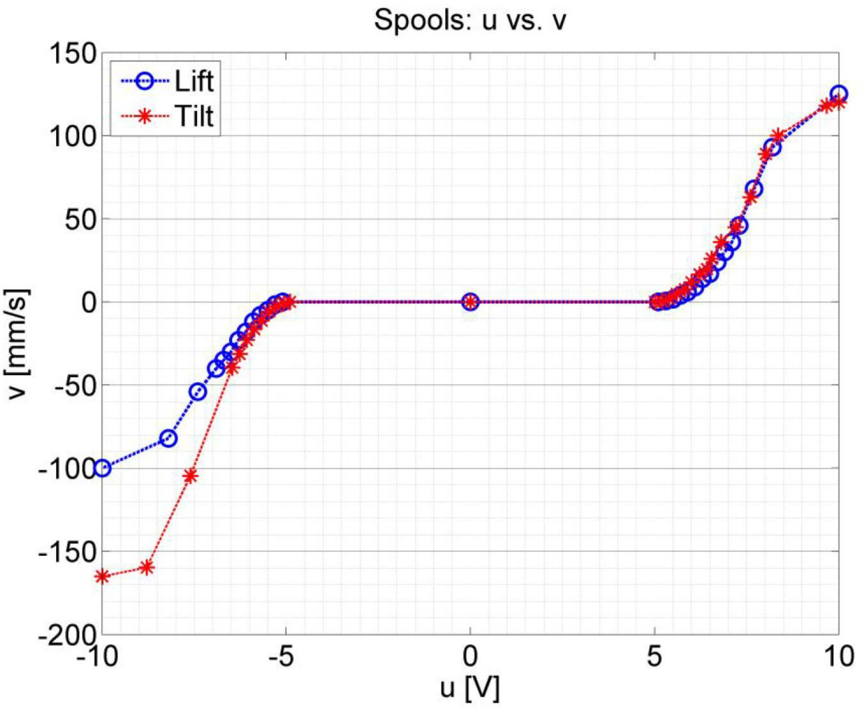

Figure 26. *Cylinder velocities vs. valve control voltage (spool position)*

5.3. MODEL-BASED DVS CONTROLLER

The model based DVS controller (and sub functions) is based on the control system engineering presented in (Linjama et al. 2007) and (Huova 2015). The layout is shown in Figure 27. The DVS controller has pressure signals and v_{CST} as inputs and valve model parameters are used in calculations. The output of the DVS controller contains valve states, pressure reference and a flow estimate. Filtering and signal processing is required on the measured values used in the controller.

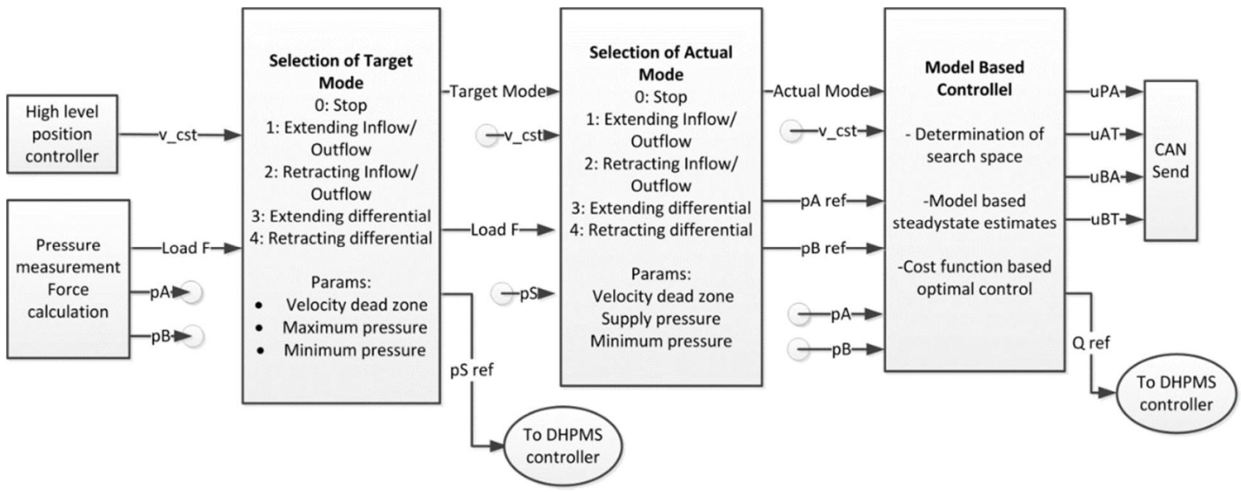

Figure 27. *Model based DVS controller architecture*

5.3.1. SIGNALS PROCESSING

Multiple measured signals are required in the control functions and filtering is required. For most of the signals 1st or 2nd order linear (digital) filters are used, but one exception exists. The load force requires a special nonlinear filter. In theory, for stable operation linear filters may be sufficient, but in practice the system behaviour might be too slow. This if the load suddenly changes, which is the case, e.g. when motion starts. The filter is described in Luomaranta et. al. (Luomaranta 1999) and for the study it is recoded into a Matlab script.

The load force is calculated from raw cylinder chamber pressure signals and then filtered. The filtering method is described in the following bulleted sentences and graphically explained in Figure 28. The filter code is presented in Appendix A. The filter algorithm is as follows:

1. Window boundaries are set (as presented in Appendix A) (*F_high* and *F_low*)

2. Output of the nonlinear filter is the mean value of the filter window boundaries. Averaging is the filtering method.

3. While the measured signal stays within the window, the window narrows as time passes with the behaviour of a 1st order system.

4. While the measured signal stays within the window, the window narrowing time constant decreases. That is, while the signal remains within the window, which occurs, e.g. during dampening oscillations, the narrowing rate becomes faster as time passes. This reduces the settling time.

5. When the input signal crosses the window boundary, the crossed boundary of the window follows the input until a peak value is reached. This results in a reduced phase shift.

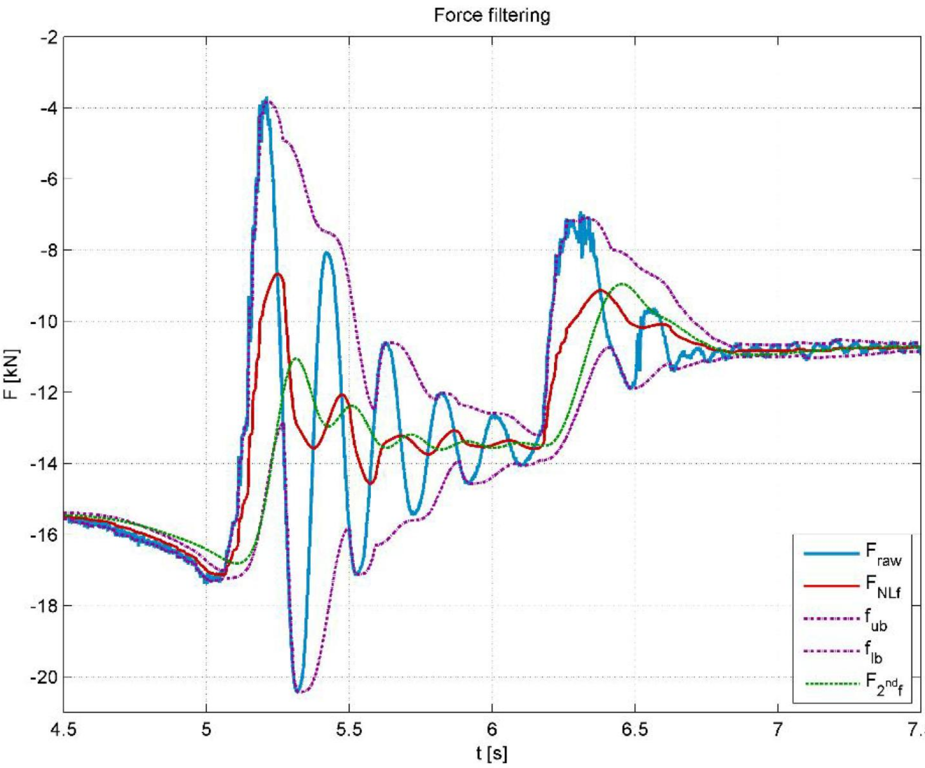

Figure 28. *Force filtering with non-linear (NL) and 2^{nd} order linear filter. NLf window boundaries, upper and lower, the average of which the F_{NLf} is, are also shown.*

In Figure 28 external load force is changing at times of about 5 s and again at 6.25s. The NLf causes less phase shift than linear 2^{nd} order filter but is somewhat more oscillatory. In the measurement system NLf is used only for load force. Other signals are filtered with 2^{nd} order filters because the best stability was empirically reached in that way. A sample time of 1 ms is used in signal processing and the filtering parameters are listed in the following table (cf. Appendix A):

Table 2. *Filter parameters*

2^{nd} order filter frequency:	10 rad/s
2^{nd} order filter damping:	0.7
NLf rate	$1 \cdot 10^{-4}$
NLf increase rate	$3 \cdot 10^{-5}$
Filter sample time	1 ms

5.3.2. MODE CHOOSING LOGIC

DVS consist of 4x6 on/off-valves which results in 2^{24} state combinations. Calculating model based estimates for all states is computationally costly, but fortunately it can be avoided by intelligent search space reduction methods. A mode choosing algorithm is the first step of the search space reduction. Ideally, a DVS can be driven in the following modes (Huova, Linjama & Huhtala 2013) which are listed in Table 3. Direction and load force determine the usable modes. The flow directions of modes through "primary" DFCUs on both A- and B-side are marked with "→"-symbols. The modes include both inflow-outflow and differential, both recuperative and "normal" power flows:

Table 3. *Modes of DVS.*

Mode	Direction	Load force	A-side	B-side	Type
PTe	Extending	Restrictive	P→A	B→T	Inflow-Outflow
TPe*	Extending	Overrunning	T→A	B→P	Recuperative Inflow-Outflow
PPe	Extending	Restrictive	P→A	B→P	Differential
TTe*	Extending	Restrictive	T→A	B→T	Differential
PTr*	Retracting	Overrunning	A→P	T→B	Recuperative Inflow-Outflow
TPr	Retracting	Restrictive	A→T	P→B	Inflow-Outflow
PPr	Retracting	Overrunning	A→P	P→B	Recuperative differential
TTr*	Retracting	Overrunning	A→T	T→B	Recuperative differential
* Requires pressurized tank line					

Pressurized tank line is not applied in this thesis, which reduces the number of feasible mode candidates to four: PTe, PPe, TPr & PPr. The mode choosing logic is divided into two parts: Target Mode and Actual Mode.

Motion is allowed when the velocity reference is larger than velocity threshold 1 and stays allowed while the reference stays larger than velocity threshold 2 (threshold_1 > threshold_2). Hysteresis is essential for avoiding over-intensive mode switching into and out of "STOP"-mode at low velocities.

Figure 29.

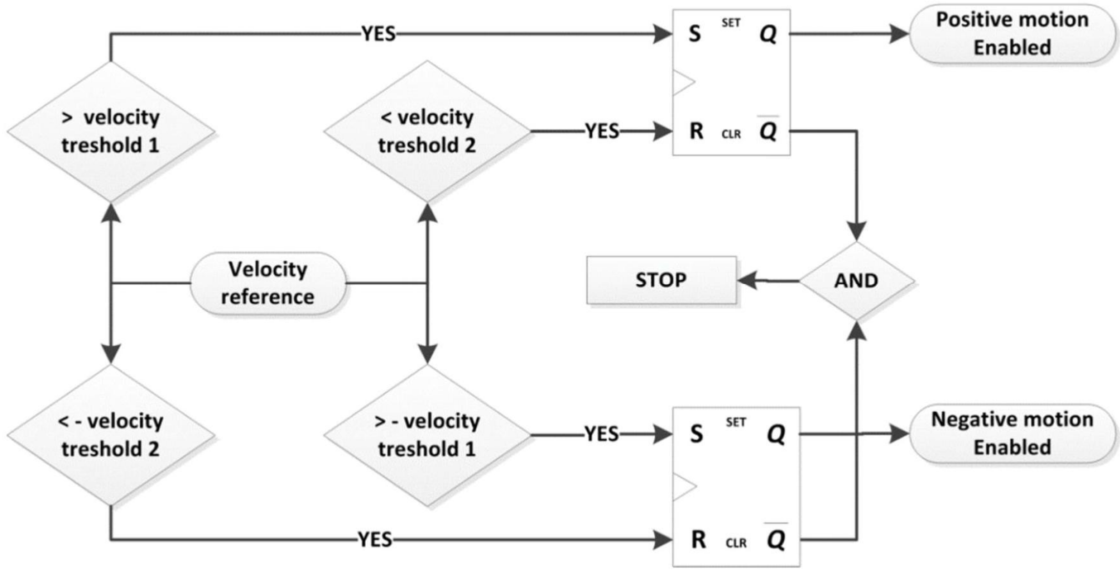

Figure 30. *Velocity hysteresis. Start from velocity reference*

The target mode is chosen based on the reference velocity and the measured load force. That is, the target mode is defined by determining what is possible within the system limits. The outputs of the target mode choosing logic are 1) the target mode and 2) the supply pressure reference. Generally, differential modes are efficient, but since every mode change consumes energy during transition it is not efficient to change the mode too often. This is taken into account in the algorithm.

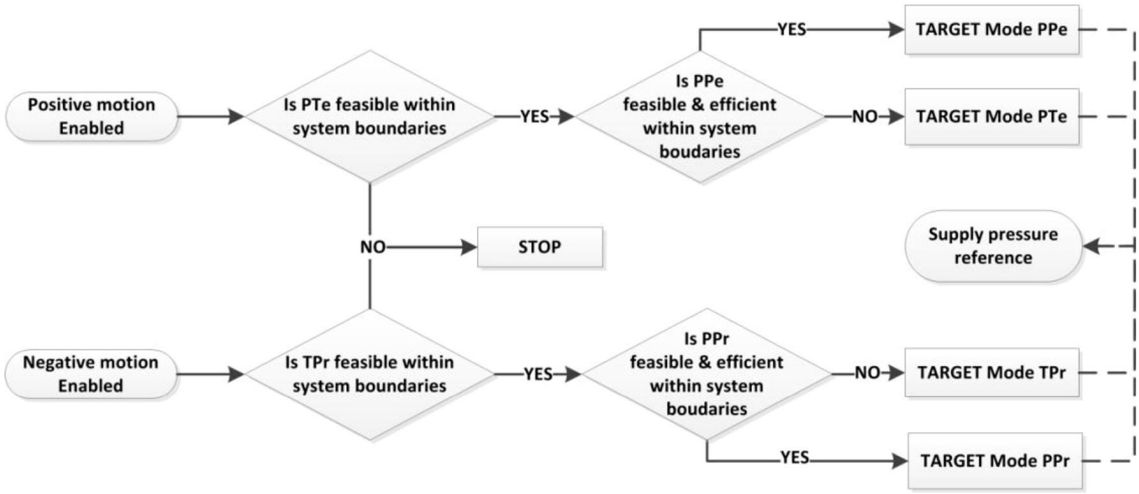

Figure 31. *Target mode choosing logic principle. Start from motion enabled*

The actual mode choosing logic sets the target mode as actual if that is possible in the current system state. A common behaviour is that the actual mode “waits” for the supply pressure to reach the reference and when conditions for safe load handling are obtained, motion is enabled. Inflow-Outflow mode can be selected instead of differential if it is feasible. In addition to the actual mode, cylinder chamber pressure references for the model based controller are calculated.

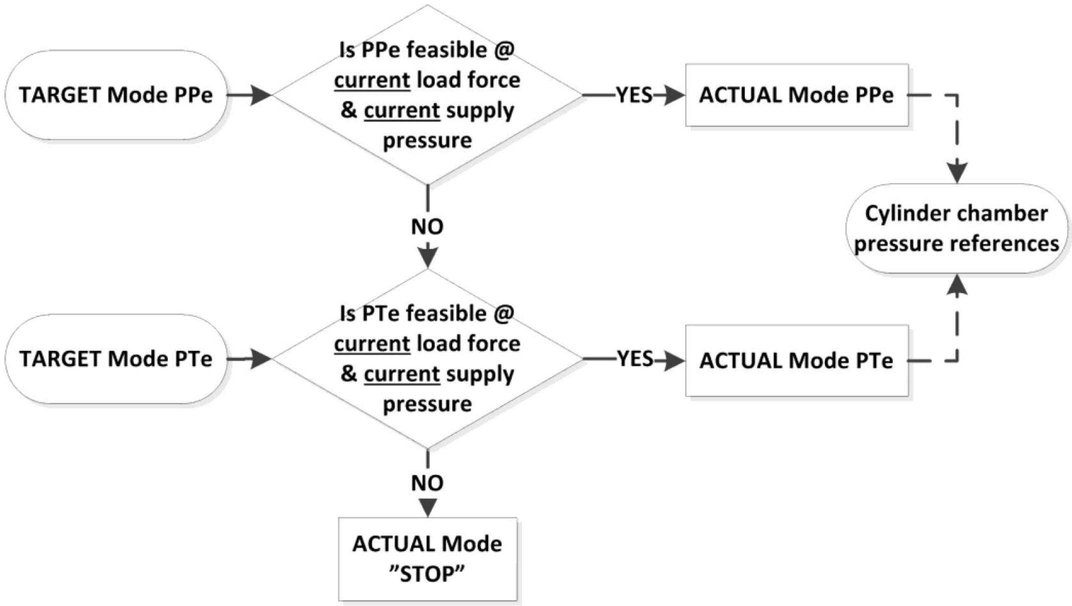

Figure 32. *Actual mode choosing logic for extending modes. Retracting is similar.*

After the selection of the actual mode, the controller calculates optimal states for cylinder A- and B-side DFCUs. Cylinder chamber references are required in those calculations.

5.3.3. SELECTING THE OPTIMAL DFCU STATES

From the selected mode the optimal controller determines which of the DFCUs are used as "primary" and which as "secondary" in motion control. Opening also secondary DFCUs causes cross-flow, which increases flow resolution. Optimal control includes two main phases: direct and non-direct calculations.

The flow rates of A- and B-side DFCUs are directly calculated with the "valve flow model" (Section 5.4.1) by using measured supply and tank pressures and cylinder chamber pressure references (previously calculated from actual mode and load force). These two sets contain 2^{2N} values per each, where N is the number of valves at any single control notch, as sum flow is taken account. Cylinder areas are known and chamber vice velocities (A- and B-sides are handled independently in the first phase) are calculated from the flows. The cross flow increases energy consumption because of power loss via secondary DFCUs which is calculated. Cost function, consisting of terms for velocity error and power loss, is minimized. A fixed small number (five in this case) of states having the least penalty are selected from both A- and B-side DFCU; others are disbanded. The final search space contains a combination of subsets of A- and B-side candidates, but it is augmented with the current state, resulting in 25+1 state candidates.

Cylinder velocity and chamber pressures are calculated for each combination of final search space. The model consists of the set of equations, meaning that indirect calculations are required. Because flow model is nonlinear, the Newton-Rhapson iterative method is used. The solution of the set of equations consists of chamber pressure estimates (pA_est, *pB_est*) and velocity estimate (v_est). Cost function values are calculated for members of the final search space. The optimal DVS state, which minimizes the cost function, contains states for each DFCU (*uPA, uAT, uBA, uBT*). Model based flow estimate is calculated and sent to the DHPMS controller.

5.4. OBTAINING PARAMETERS FOR MBC VALVE CONTROLLER

The parameters can be initially set heuristically and by simulations and fine-tuned by measurements. The parameters define, e.g. the target pressure differential (*ELS Δp*), mode switching hysteresis, cost function parameters, etc. which affect the system. The target *Δp* value must be high enough so that pressure ripple caused by the pulsating flow of the DHPMS does not trigger stop-mode, which occurs if the *Δp* is lower than the minimum allowed value. This value is essential because the controllability diminishes if the pressure differential is too low.

The minimum allowed velocity is the important value in the sense of stability. Initial value can be calculated from the target pressure differential and from the flow capacity of the smallest valve and by assuming that the utilization of cross flow halves it. The calculated value is 4.2 mm/s and the value of 4 mm/s was found to be good in the measurements. For the minimum velocity there is hysteresis in a way that the movement is allowed to start when v_{cst} is bigger than 4 mm/s but remains in motion until the v_{cst} is smaller than 2 mm/s.

The model based valve controller utilizes data from every individual valve and orifice pair. In total, there are 48 of these (2 DVS containing 4 DFCUs per each consisting of 6 on/off-valves). The valve flow model in the MBCs is based on an empirical model, which is a generalized version of the commonly used square root model.

5.4.1. VALVE FLOW MODEL

A traditional way to model an orifice is based on a square root based characteristic curve model. A more accurate two-input-six-parameters flow model has been published in (Linjama, Huova & Karvonen 2012). The flow model is the following (eq 3):

$$Q(p_1,p_2)=\begin{cases} Kv_1\cdot(p_1-p_2)^{x_1}, & b_1\cdot p_1 < p_2 \le p_1 \\ Kv_1\left(\left(1-b_1\right)\cdot p_1\right)^{x_1}, & p_1 \le b_1 p_1 \\ -Kv_2\cdot(p_2-p_1)^{x_2}, & b_2\cdot p_1 < p_2 \le p_1 \\ -Kv_2\left(\left(1-b_2\right)\cdot p_2\right)^{x_2}, & p_2 \le b_2 p_2 \end{cases} \tag{3}$$

The parameters of the model are direction dependent and the cavitation choking effect is taken into account. *Kv* and *x* define the flow rate and shape of the curve. The parameter *b* determines the critical pressure ratio. The subscript number defines the direction of the flow. It is worth noting that value $x = 1$ equation can be used for laminar flow but with $x= 0.5$ it represents the square root flow model. Cavitation choking is not "activated" if value $b = 0$ is selected. The parameters of the valves are presented in Appendix B. An example of the general shape of the function is presented; a flow surface of one of the bits of one of the control notches is presented in Figure 33.

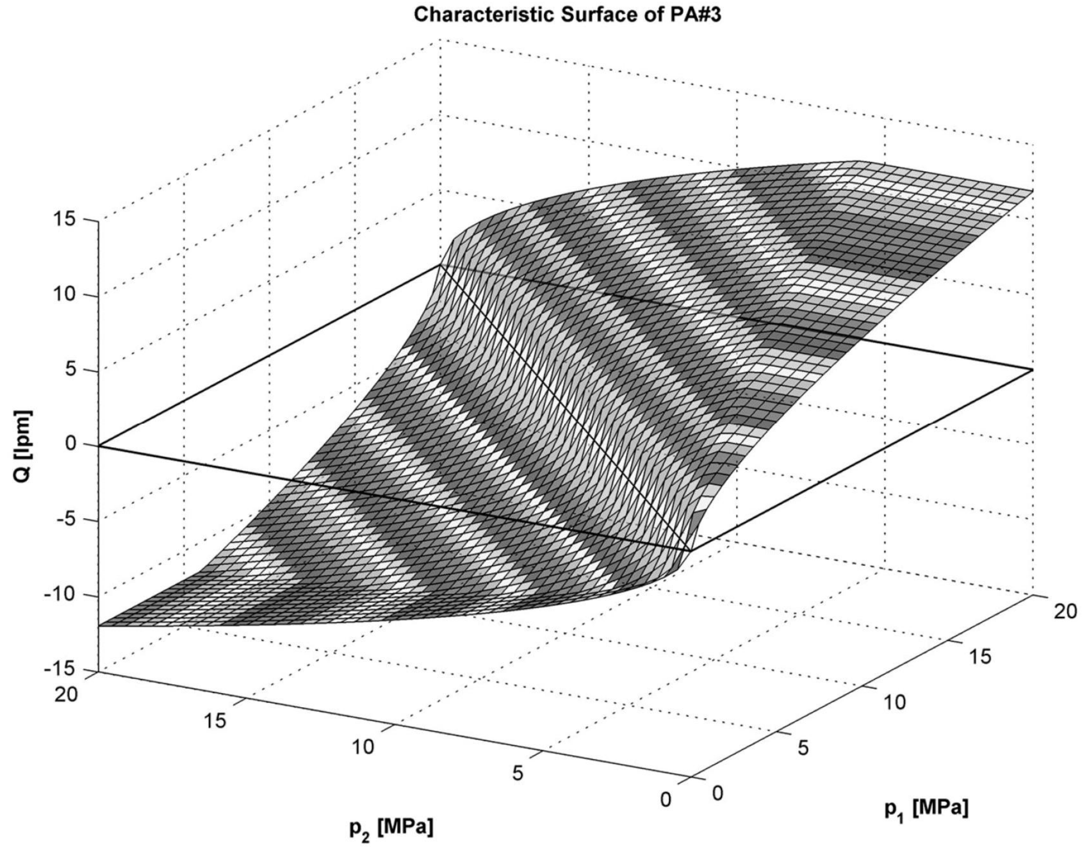

Figure 33. *Characteristic surface of PA notch, bit 3. The colour scheme is selected to visualize the shape of the surface, especially the effect of cavitation choking visible at the flat areas.*

5.4.2. MEASURING AND DEFINING VALVE PARAMETERS

Measurements are required for obtaining the values of the six parameters. Two tests are made:

- Upstream pressure is kept at constant value. Downstream pressure ramp from certain value to zero and back to the value. This is called "p_2-ramp" later on.
- Upstream pressure ramp from zero to certain value and back to zero. Downstream pressure is kept at zero. This is called "p_1-ramp" later on.

Figure 34 contains an example of a measurement of bit#3 of the PA of notch of the lift DVS.

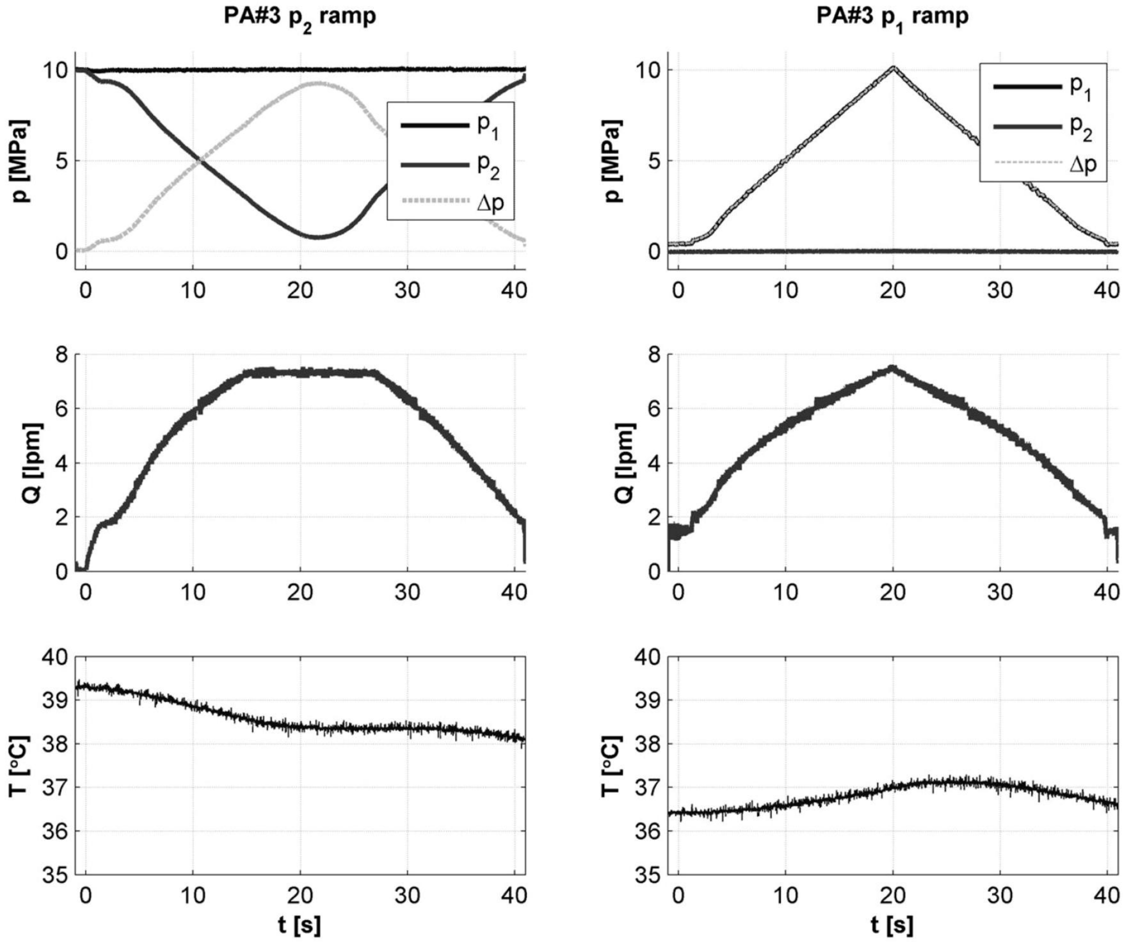

Figure 34. *Example of pair of measurements containing both inlet and outlet pressure ramps. Both up- and downstream pressures are closed loop controlled. At the p_2 ramp the upstream pressure is kept constantly high and at the p_1 ramp downstream pressure constantly at one atm.*

Figure 34 shows the importance of different ramps. If only the p_1-ramp is measured, the cavitation choking effect cannot be detected. However, the fact is that cavitation choking is occurring all of the time, as seen in Figure 35. The cavitation choking parameter can only be identified from p_2-ramp measurement and identification process should start from there. Later on, the values for *Kv* and *x* can be obtained. A good initial value for *x* is 0.5. The initial value for *Kv* can be calculated from any value with the square root model.

In Figure 35 the measured p_2-ramp (red line) and fit to it is shown (green line). However, if those parameters are used in the valve model, and the model is tested with p_1-ramp, the model output (magenta line) and measurement (blue line) differ. An improved fit to the p_1-ramp can be obtained, but the cavitation choking parameter should be still obtained from the p_2-ramp measurement. Other parameters can be re-tuned, which results in a better fit (cyan line). Again, the model should be tested against measurement, and now p_1-tuned parameters are tested with the p_2-ramp case (dotted black line). Also now a slight model error is visible. Clearly, fluid flow is a complex phenomenon even in the simplest case. In practice the 6-parameter model is found to be usable.

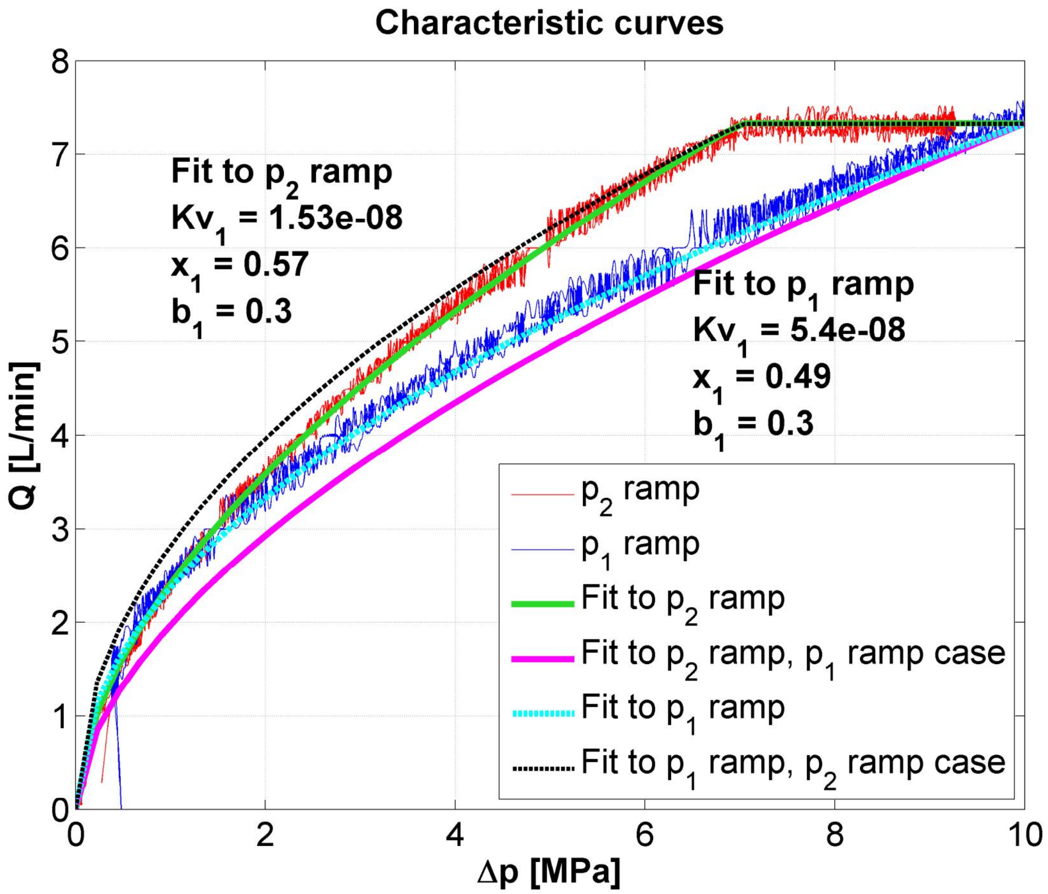

Figure 35. *Defining parameters from measurements and flow model validity*

Parameters for the valve models should be obtained from the p_2-ramp measurements, but at the tank notches towards the tank line parameters they should be taken from p_1-ramp measurements if there is a zero pressure tank line in use.

In the valve test bench ISOVG46 grade oil was used, but in test setup the oil is ISOVG32. This was a known feature and compensated by doing measurements in different oil temperatures: Valves were measured in 35-40 °C and the system in 25-30 °C. However, oil temperature was neither constant nor directly controllable, as it seldom is in real applications.

5.4.3. FLOW SERIES

The valve orifices generate semi-binary coding; the orifice diameters are [0.6, 0.8, 1.2, 1.6, 2.3, 2.4] mm. Hence, the DFCUs have 64 states, but many of the states have flow capacities close to some other. In order to reduce the computational load the search space was pre-reduced in the following manner: From state 13 on, the smallest valve remains open and from state 22 also the second smallest bit remains open. This results in the resolution of the valve decreasing as the states increase above those levels, but instead of 64 states of the DFCUs there are only 27.

The valve parameters are known so the flow series can be calculated. In Figure 36 the PA control notch flow rates for both lift and tilt valve blocks are presented at 2 MPa pressure differentials. The upper figure shows the flow rates of the states and the lower presents the resolution. The resolution reduction is seen as bigger steps following the states 13 and 22. In an ideal binary series the steps would be identical, but realistically the flow characteristics are as presented. All DFCUs are identically coded, but as clearly visible in reality, some differences occur.

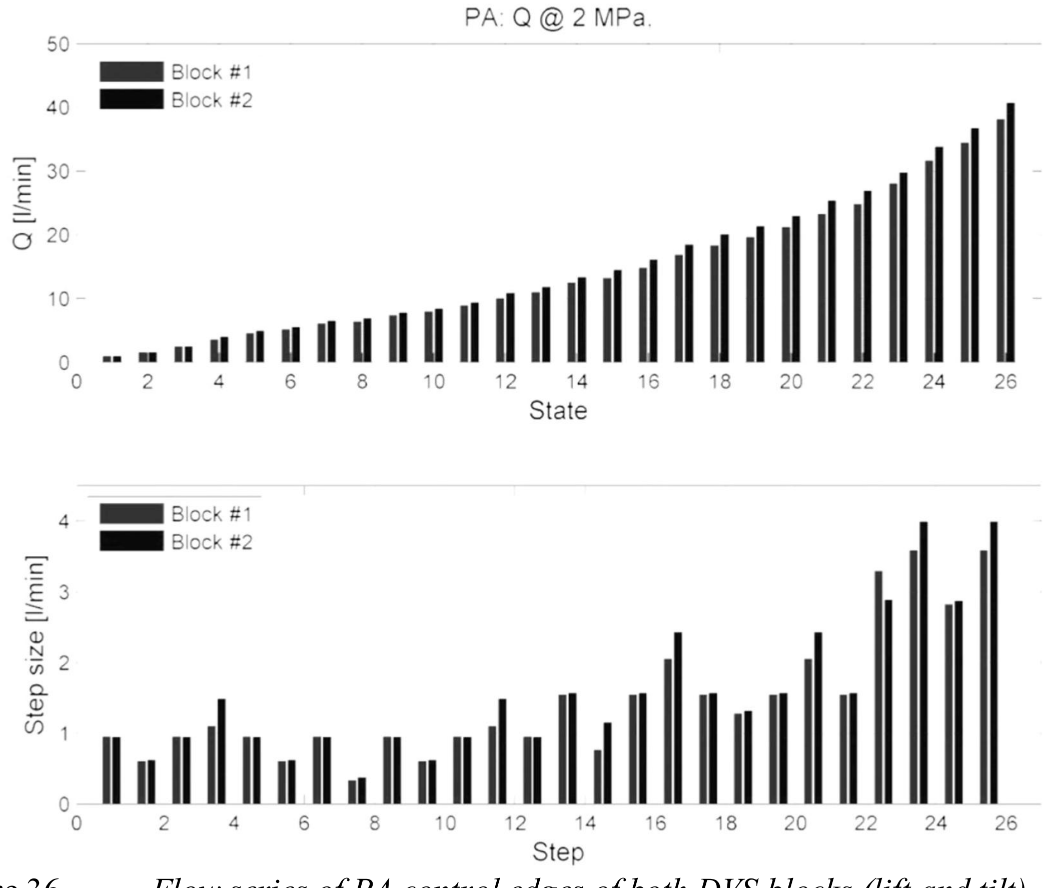

Figure 36. *Flow series of PA control edges of both DVS blocks (lift and tilt). All flow series are technically similar but minor deviation is natural.*

5.5. DHPMS CONTROLLER

The control of the DHPMS seeks to maintain pressures in the actuator supply lines at target values, which are set, e.g., according to ELS-functions. In the studied boxer pump unit, six mode decision instants occurred during one pump revolution, and at each of them the pumping mode was selected for one piston and the suction mode for the other (Heikkilä et al. 2010). Evidently, in the pumping and suction phase, the pistons could be connected to either the actuator supply line or to the tank line, resulting in three possible pumping and suction modes (T, A, B) for each piston. Active digital valves were further controlled according to the piston modes.

Figure 37 shows the pressure control logic of the DHPMS. The block diagram presents a model-predictive mode selection logic for a pair of pistons with opposite phases. First, a change in the supply line fluid volumes is estimated against previously selected modes and actuator flows. Then, linear extrapolation, as a function of shaft angle, is used to determine an additional fluid volume due to yet uncompleted strokes. As a result, the vector $\boldsymbol{\Delta V}_{1\_est}$ = [$\Delta V_{A\_1\_est}$, $\Delta V_{B\_1\_est}$] is formed. Similarly, the actuator flow volumes $\Delta V_{A\_2\_est}$ and $\Delta V_{B\_2\_est}$ are extrapolated till the stroke end by assuming that the actuator flows remain unchangeable. Actuator flow estimates $\boldsymbol{Q}_{2\_est}$ = [$Q_{A\_2\_est}$, $Q_{B\_2\_est}$] can be calculated from actuator velocity estimates owing to the rapid pressure response in transient states.

Table 4. *Possible mode combinations and their effect on supply line oil volumes*

Pumping	**Suction**	$\boldsymbol{\Delta V}_{\mathbf{A\_mode}}$	$\boldsymbol{\Delta V}_{\mathbf{B\_mode}}$
T	*T*	0	0
A	*T*	V_{disp}	0
B	*T*	0	V_{disp}
T	*A*	- V_{disp}	0
A	*A*	Not used	
B	*A*	- V_{disp}	V_{disp}
T	*B*		-V_{disp}
A	*B*	V_{disp}	-V_{disp}
B	*B*	Not used	

For all mode combinations, errors in the supply line pressure are calculated considering the measured pressures $\boldsymbol{p}$ = [p_A, p_B] and their target values $\boldsymbol{p}_{ref}$ = [p_{A_ref}, p_{B_ref}] when the change in the fluid volumes $\boldsymbol{\Delta V}_{est}$ = [ΔV_{A_est}, ΔV_{B_est}] and hydraulic capacitances $\boldsymbol{C}_h$ = [C_{h_A}, C_{h_B}] are known. In this study, the supply line capacitances were considered constant, and the equation can be written as follows:

$$\left|\boldsymbol{p}_{\mathrm{err}_{\mathrm{est}}}\right| = \left|\boldsymbol{p}_{\mathrm{ref}} - \boldsymbol{p} - \frac{(\boldsymbol{\Delta V}_{\mathrm{est}} + \boldsymbol{\Delta V}_{\mathrm{mode}})}{\boldsymbol{C}_{\mathrm{h}}}\right| \qquad (4)$$

where $\boldsymbol{\Delta V}_{mode}$ = [ΔV_{A_mode}, ΔV_{B_mode}]. The possible values of ΔV_{A_mode} and ΔV_{B_mode} are shown in Table 4, where V_{disp} is the geometric piston displacement. Because we excluded the mode combinations of pumping and suction chosen for the same supply, the optimal mode combination was a search amongst seven candidates by minimizing pressure errors. Optimal modes were then routed to the valve switching controller, and correct valve timing was determined in relation to pressures and angular velocity to optimize pre-compression and pressure release times and to compensate for valve delays (Heikkilä et al. 2010). The sample time of the DHPMS controller is 50 µs.

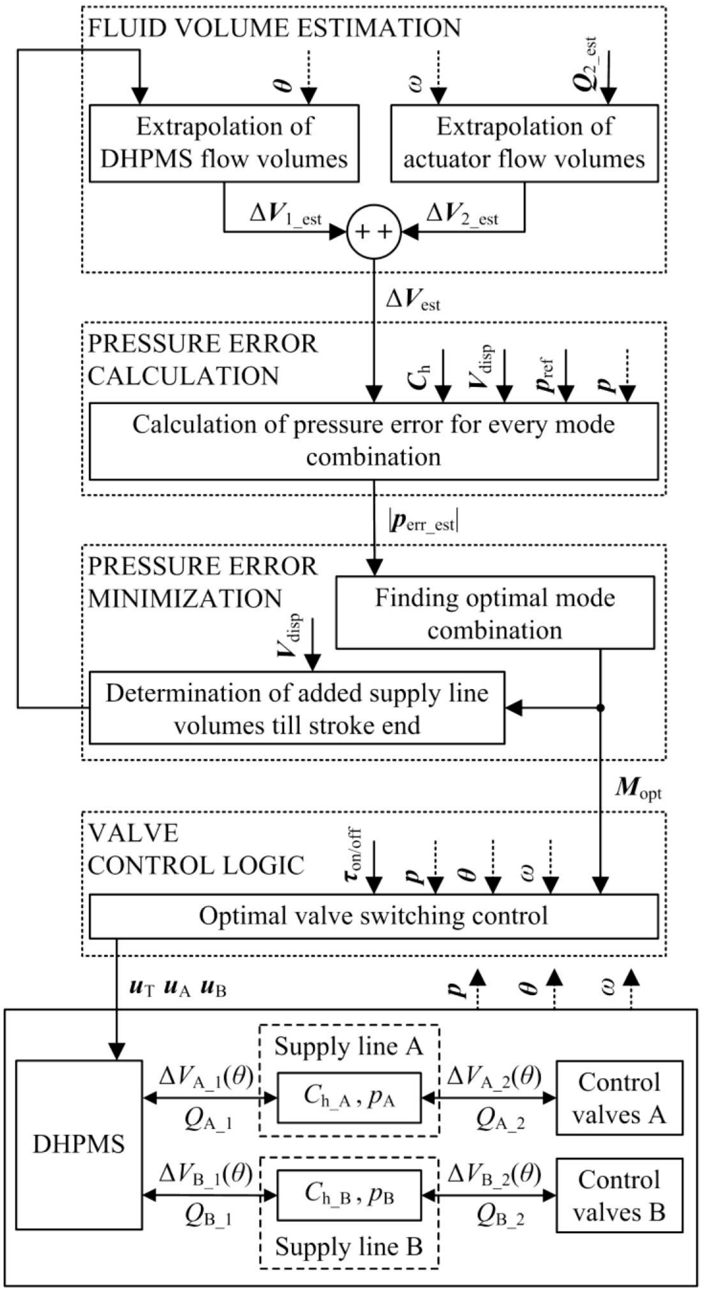

Figure 37. *Mode control logic of the DHPMS (Karvonen et al. 2014)*

6. SIMULATIONS

Simulation provides a sophisticated tool to analyse ideas and test control system designs. MATLAB Simulink has been used as the simulation software. Two independent simulation studies have been performed. The most important results of all the simulations were in control system design, but also estimated improvements to the efficiency were determined. The designed control system was implemented into the measurement setup. Also the pressurized tank line was tested by simulations, but it was not possible to perform measurements on that technology in the case of the selected plant.

In the first simulation study published by the author (Karvonen et al. 2011), the applicability of a DHPMS to work as a two pressure source was studied. The required control codes were designed and tested with a simulation model. In the study ideal pressure compensated proportional valves were used. The LS-supply pressure source was modelled by a simple second order transfer function with rate limiter and saturation. Reference pressure was determined as traditionally done in LS-circuits. The simulation study shows that DHPMS is capable of providing separated ELS supply pressures. The results also indicate that by reducing matching losses by means of independent supply pressures; about 20 % energy savings should be achievable.

In the second simulation study, the effects of different control algorithm modifications of a DVS, when implemented together with the DHPMS, were studied. This was done to reveal holistic level phenomena and verify the optimal control method of DVSs in the test system. This study resulted in upgrades to the DHPMS controller and the final version of DVS controller for measurement setup was determined. The study is published in (Karvonen et al. 2014). Pars of the author's accepted manuscript of the paper is allowed to be presented in this thesis by courtesy of Taylor & Francis. The simulation chapter therefore consists partly of that.

In the second study the digital valve model emulated the proportional valve, producing a discrete proportional valve controller. Virtual spool geometry was selected so that cavitation free actuation was achieved. For the tested model based controllers, three different mode choosing-logic methods were tested.

6.1. SIMULATION MODELS

The simulation models were made with MATLAB/Simulink and SimMechanics. The equations are shown in the chapter and the parameters are listed in Appendix C. Chronologically, the simulations were made before the measurements and therefore, e.g. measured valve parameters were not used but rather estimates. Also a nonlinear filter for force was used only in measurement as the simulation model worked well enough with linear filters.

6.1.1. MECHANICAL MODEL

Figure 39 shows a CAD image of the machine. The inertia ellipsoids of the mechanical model are shown in Figure 40. The cylinders were not modelled as bodies but only as force sources. The mechanical model's cylinder connection points were used to calculate directions for the forces applied. In addition, cylinder lengths and velocities were calculated from the connection point coordinates.

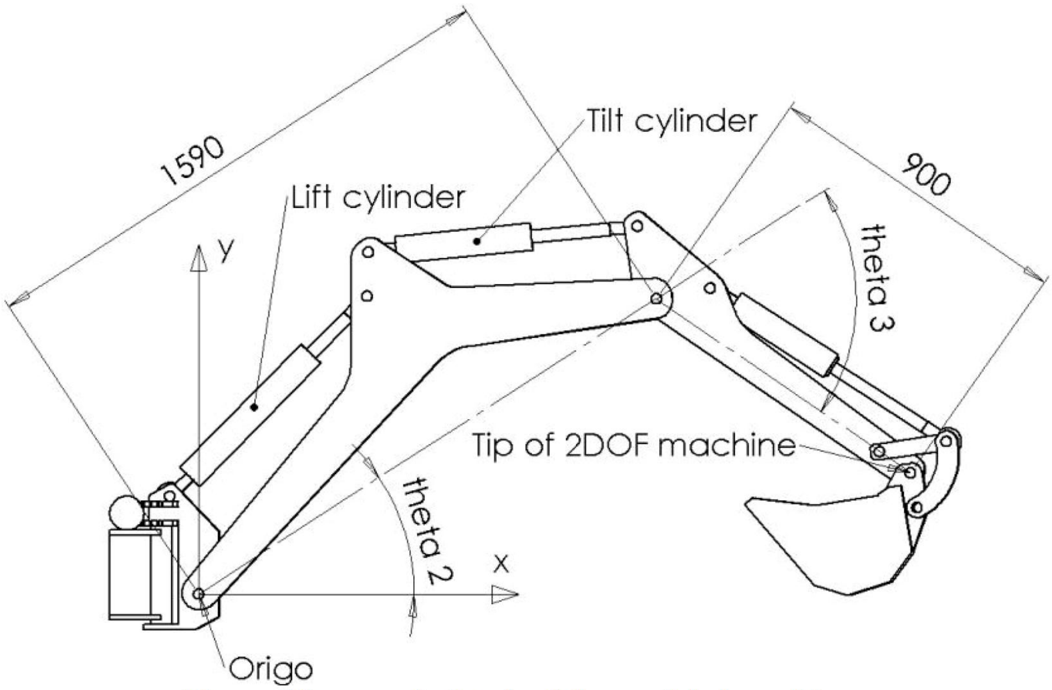

Figure 39. *A sketch of the modeled machine*

Figure 40. *Inertia ellipsoids from SimMechanics. Cylinders were not modeled as bodies. Cylinder piston positions, velocities, and generated torques were analytically calculated.*

6.1.2. HYDRAULIC MODELS

The cylinders were modelled as two volumes linked together so that the volumes of the cylinder chambers corresponded to reality. Eq. 5 shows an equation for the pressure dynamics. Cylinder friction was modeled with a friction model based on a hyperbolic tangent, as shown in Eq. 6.

$$\frac{dp}{dt} = \frac{B_{eff}}{V}\left(\sum Q - \frac{dV}{dt}\right) \tag{5}$$

$$F_{fric=} \tanh(K \cdot \dot{x}) \cdot \left(F_C + (F_S - F_C) \cdot e^{-(\dot{x}/v_S)^2}\right) + b \cdot \dot{x} \tag{6}$$

On/off valves were modelled with the empirical equation of a turbulent orifice:

$$Q = sign(\Delta p) \cdot u \cdot Kv \ \cdot |\Delta p|^x \tag{7}$$

where u represents the binary control value. On/off valve dynamics were modelled only with a delay and a rate limiter.

6.1.3. THE DHPMS MODEL

Our digital hydraulic power management system is based on volume models that change according to the crank shaft angle and orifices that connect the chambers to one of the actuator supply lines or to the tank line. When the piston's trajectory is sinusoidal, its position can be solved from Equation 8,

$$\mathrm{x(t)} = \frac{\mathrm{s}}{2}\sin(\omega \mathrm{t} + \theta) + \frac{\mathrm{s}}{2} \tag{8}$$

where ω is angular velocity, θ the phase shift, and s the stroke of the piston. The instantaneous pressure of each cylinder chamber can be integrated from Equation 2. The DHPMS was modelled by combining in parallel six piston units with a phase shift of sixty degrees. Supply lines A and B were modelled as static 6.6 dm^3 volumes overall with DHPMS and actuator flows as inputs. In the supply lines, a rigid volume of 5 dm^3 was added for better pressure control resolution (Heikkilä et al. 2010) with the hose volumes (1.6 dm^3) also included in the overall volume. Supply line capacitance determines the amount of pressure rise caused by one pumping stroke. In simulations smaller pistons are easily parameterized, but empirically possible methods are used also in simulations for easier model verification.

6.1.4. DFCU PARAMETERIZATION

The digital valve models were based on an empirical orifice model with diameters set in a series of [0.6, 0.8, 1.1, 1.6, 2.4, 2.5] mm. The diameters were chosen for the series to provide fair resolution, high enough flow, a small variation in step size, and authenticity to real components. A flow characteristic estimate can be calculated by assuming that a square root model is valid. All the control notches were similar. The DFCU containing six valves had 64 states, but state space was pre-reduced to minimize the calculation load in the model-based controller. Pre-reduction was done as presented in an earlier chapter.

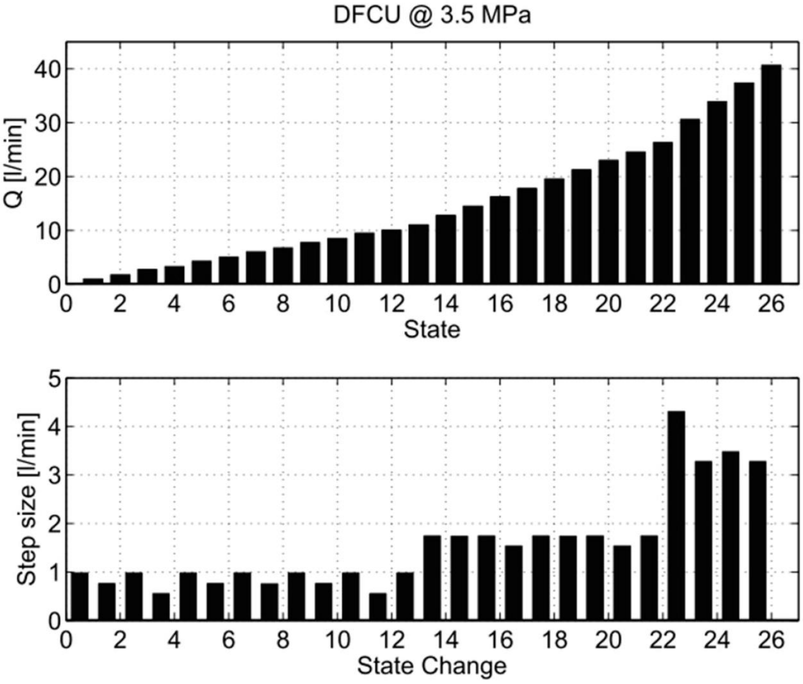

Figure 41. *Static characteristics of the DFCUs. Upmost is the flow vs. state, middle the step size vs. state change, and at bottom the characteristic curves of the valves. N.B. Due to pre-reduced search space, state 26 stands now for opening vector [1 1 1 1 1 1], not for vector [1 1 0 1 0], as it would with a full search space.*

The valve dynamics were modelled with a delay and a rate limiter such that the opening delay was 6 ms and the closing delay 10 ms. The armature movement time was 4 ms open and 5 ms close, resulting in switching times of 10 ms open and 15 ms close, which approximates the values of a certain commercial valves driven with proper valve drivers. Because the controller time step must be selected so as to secure safe state transitions before states are updated with new values, we selected position and valve controller time steps of 20 ms.

6.1.5. DISCRETE PROPORTIONAL VALVE CONTROLLER

The simplest digital hydraulic valve controller consists of proportional gains for each control notch such that two control notches are controlled simultaneously, as is done in a traditional single spool proportional valve. The laws controlling the notch states are expressed below in Eq. 9:

$$
\begin{aligned}
&u_{PA} = K_{PA} \cdot (x_{ref}\text{-}x)\ ,\ 0 \le u_{PA} \le (2^n - 1) \\
&u_{AT} = \text{-}K_{AT} \cdot (x_{ref}\text{-}x)\ ,\ 0 \le u_{AT} \le (2^n - 1) \\
&u_{PB} = \text{-}K_{PB} \cdot (x_{ref}\text{-}x)\ ,\ 0 \le u_{PB} \le (2^n - 1) \\
&u_{BT} = K_{BT} \cdot (x_{ref}\text{-}x)\ ,\ 0 \le u_{BT} \le (2^n - 1)
\end{aligned}
\tag{9}
$$

where *u* is the control signal (state) for a notch, *K* is proportional gain, and *x* is the cylinder position. *n* stands for the number of valves in a DFCU. The *K*-gains are fixed and affect the valve similarly to spool geometry in a traditional proportional valve. The *K*-gains are tuned to guarantee cavitation free actions with all but the highest load mass. Henceforth, this controller is referred to as the "Emulated Proportional" as it mimics the behaviour of a proportional valve. The following table lists the gain values.

Table 5. *Virtual spool geometries for control notches. The value defines the state used if the position error is one meter.*

Flow gains	**Lift [1/m]**	**Tilt [1/m]**
K_{PA}	220	220
K_{AT}	180	180
K_{PB}	180	180
K_{BT}	86	180

Because this controller contains no pressure compensation functions, an ideal pressure compensator was modelled in the valve model. This mimics a digital valve with traditional pressure compensator. Since the valve parameters are known, the flows of the states at a certain pressure differential can be calculated and the states put in order. A selector block is used to find the row that corresponds to the control value, and the binary state vector in that row represents the desired state.

The above controller has limited functionality, but it can still "do the work." Its functions are simple enough for formal methods to be used to guarantee its functionality, which would be tricky with a complex model-based controller. Parallel use of a simple "safe" controller and a complex optimal "not-guaranteed-to-be-safe" controller was studied in (Huova & Linjama 2012), but in that study the pressure compensator was in the software.

6.1.6. MODE-CHOOSING LOGIC FOR INDIVIDUAL ACTUATORS

A single actuator has four possible actuator modes. In the following list, the letters P, A, B, and T are paired to indicate flow direction in the order of the letters. The control notches used can be seen from the letter pairs. Whenever the letter P, which represents a supply pressure line, is the latter, it indicates the regenerative mode because the flow is towards that port. This mode choosing logic was finally used in measurements.

- **Mode 0**, Stop mode

- **Mode 1**, Extending inflow/outflow (PA&BT)
- **Mode 2**, Retracting inflow/outflow(PB&AT)
- **Mode 3**, Extending differential (PA&BP)
- **Mode 4**, Retracting differential (AP&PB) (recuperative mode)

With this mode-choosing logic, both actuators are operated independently, and the MBCs produce their own pressure references to the DHPMS controller. With two independent supply pressures, pressure references are used as they are. With only one supply pressure version, the highest pressure reference is used. Unnecessary pressure, if present, is throttled down at the outflow control notch while the pressure differential at the inlet control notch is to remain at the target Δp of the control valve. This modification of the DVS controller is referred as *MBC_{Orig}*.

6.1.7. MODE-CHOOSING LOGIC FOR TWO ACTUATORS SHARING THE SAME SUPPLY PRESSURE

DVS mode-choosing logic can be turned into a cleverer version of the same logic, selecting modes for both actuators and producing a pressure level reference to optimize a two-actuator system pressure reference to minimize the power requirement. The modes possible for a single actuator are the same as in the previous case. The trick behind this logic is to calculate the model-based estimates into a power requirement for all possible mode combinations of two actuators. The combination estimated to require the least power is then selected. This controller is usable only for single pressure, multi-actuator machines. This controller has 16 different mode combinations, and the combination most likely to minimize the energy consumption estimate is selected. This modification of the original MPC is henceforth referred to as the *MBC_{TwoActMod}*. In case of independent supply pressures, this modification would result same result as *MBC_{Orig}* and therefore that case is not simulated.

6.1.8. MODE-CHOOSING LOGIC FOR A PRESSURIZED TANK LINE

If flow is available from the tank line, which requires a pressurized tank line, more modes are usable:

- **Mode 4**, Extending the pressure side differential (PA&BP)
- **Mode 3**, Extending regenerative inflow/outflow (TA&BP) (recuperating mode)
- **Mode 2**, Extending the tank side differential (TA&BT)
- **Mode 1**, Extending inflow/outflow (PA&BT)
- **Mode 0**, Stop mode
- **Mode -1**, Retracting inflow/outflow (AT&PB)
- **Mode -2**, Retracting the tank side differential (AT&TB) (recuperative mode)
- **Mode -3**, Retracting regenerative inflow/outflow (AP&TB) (recuperative mode)
- **Mode -4**, Retracting the pressure side differential (AP&PB) (recuperative mode)

Note that the recuperative capabilities are greatly increased due to the new modes. Modes 4 and -4 were not used because they cause a discontinuous supply pressure reference, which does not improve the efficiency of single actuators. This controller, henceforth marked as the *MBC_{PresTankMod}*, has been described in detail in (Huova 2012).

6.1.9. SIMULATION TEST CASES

The circular reference trajectory was driven with five different loads and five different velocities, and a total of 25 combinations were simulated for four different controllers together with single and independent supply pressures. The fastest trajectory was set so as to fulfil the maximum flow demand with the six-piston DHPMS prototype. DHMPS parameterization was based on a real machine.

Load masses of 0, 75, 150, 225, and 300 kg, and trajectory times of 10, 15, 20, 25, and 30 seconds are used. Peripheral velocity is kept constant, and no start- or end-smoothing functions are used. This result in somewhat jerky behaviour at end and start instances, but it simplifies trajectory generation. Also jerks at the start and end are insignificant in terms of energy efficiency. Furthermore, it is important that the system maintains stability also in case of sudden changes in references.

The circular trajectory had the same starting and ending point, and because the load mass was kept constant during simulation, no actual work was done. At a certain time step, output power was then calculated from the product of the cylinder net force, which is the sum of the products of pressures and areas multiplied by velocity. Hydraulic input power was obtained from the pressures and flows of the DHPMS and mechanical input power from the angular velocity and torque of the DHPMS. Work from the power was obtained by a trapezoidal integration method. Figure 42 through Figure 48 illustrate the 15-s trajectory and 300-kg load mass case for each simulated system (Table 6).

Table 6. *List of figures concerning simulated cases*

Controller modifications	Single supply pressure	Independent supply pressure
Emulated proportional	Figure 42	Figure 43
MBC_{Orig}	Figure 44	Figure 45
MBC_{TwoActMod}	Figure 46	N.A.
MBC_{PresTankMod}	Figure 47	Figure 48

Figures contain matrices of subplots with the left column for the lift cylinder (*Cyl 1*), and the right column for the tilt cylinder (*Cyl 2*). The topmost row displays cylinder position and position reference. The second row is for velocity and velocity reference, the output of the high level position controller. The third row for mode has signals for both target and actual modes. As the emulated proportional valve controller used only inflow/outflow modes, no mode selection logic was needed. The DFCU states in the fourth row are control notch openings. The following rows stand for chamber pressures and supply pressure and its reference. The last row indicates hydraulic power consumed by the line and the output power of the actuator. Negative output power means overrunning load, and negative input power occurred if the DHPMS was motoring. Input work for a line is shown in the middle of the subplot. The last graph has bars for total input, hydraulic input, and output energies summed over all simulated cases with varying trajectory time and load mass.

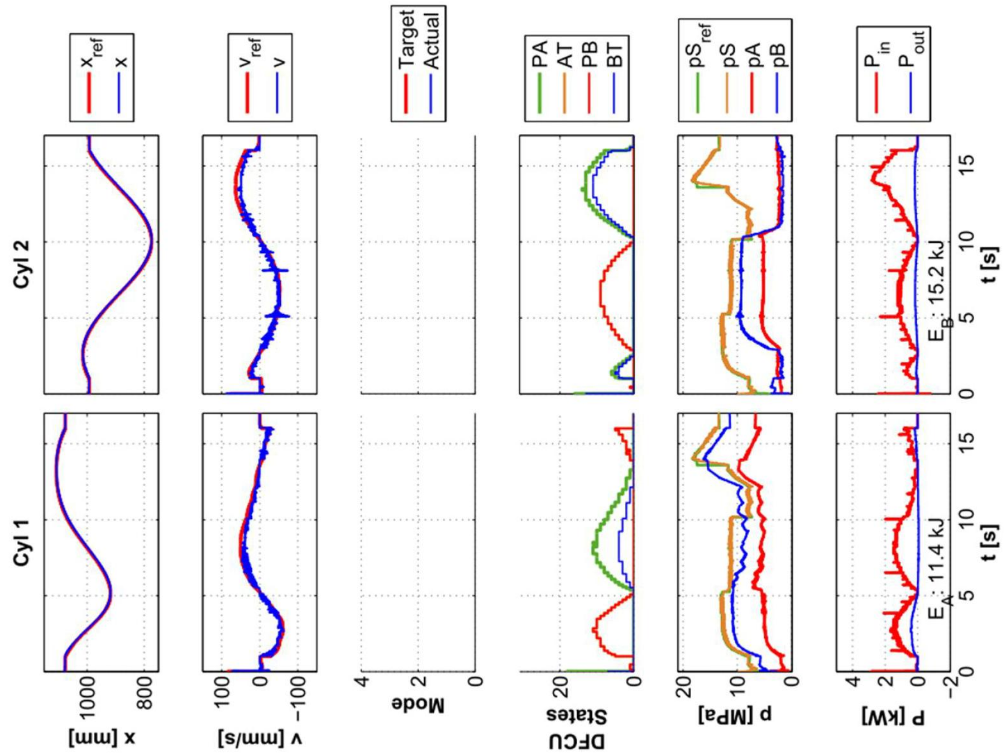

Figure 42. *Emulated proportional valve with common supply pressure for both actuators. Ideal pressure compensator models were used.*

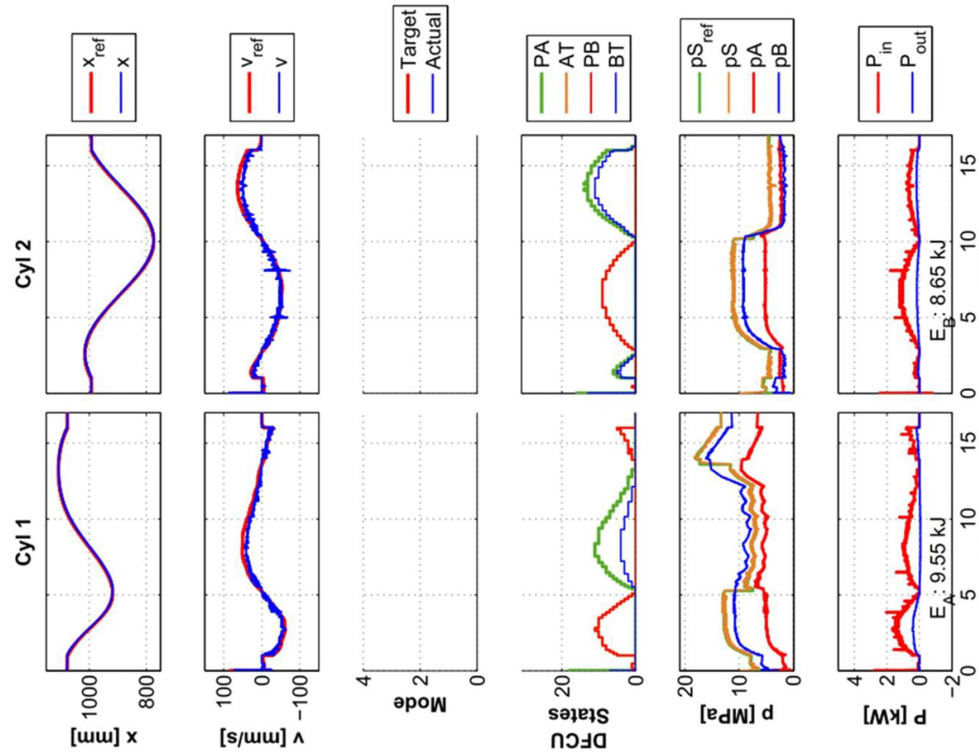

Figure 43. *Emulated proportional valve with two independent supply pressures. Ideal pressure compensator models were used.*

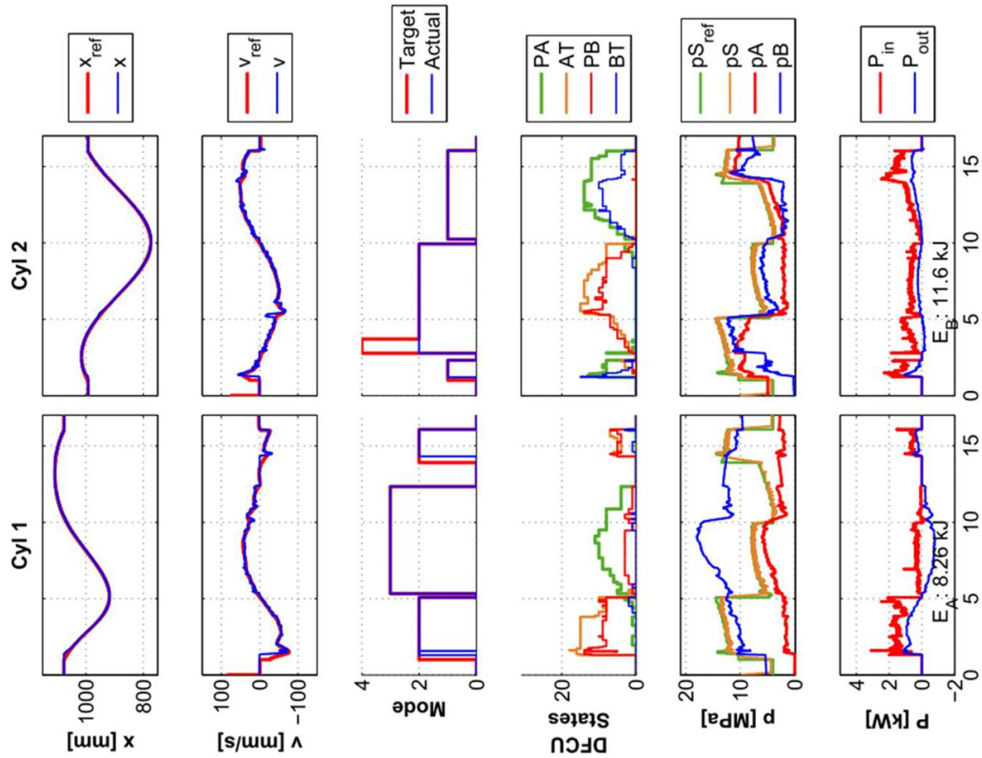

Figure 44. *MBC_{Orig} with common supply pressure*

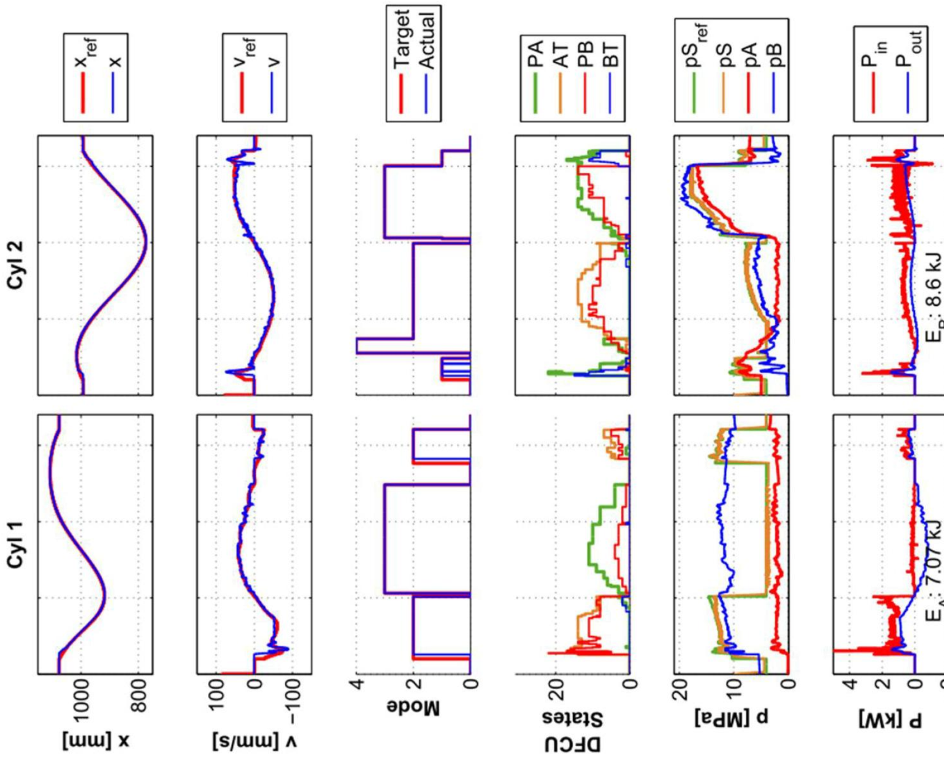

Figure 45. *MBC_{Orig} with two independent supply pressures.*

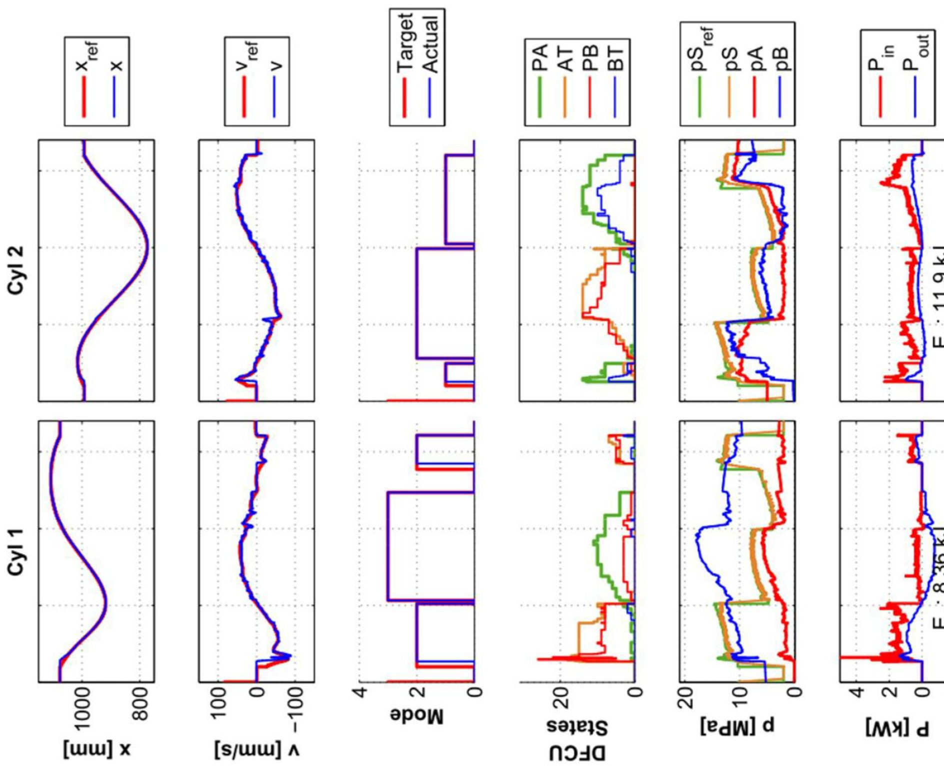

Figure 46. *MBC_{TwoActMod} with common supply pressure*

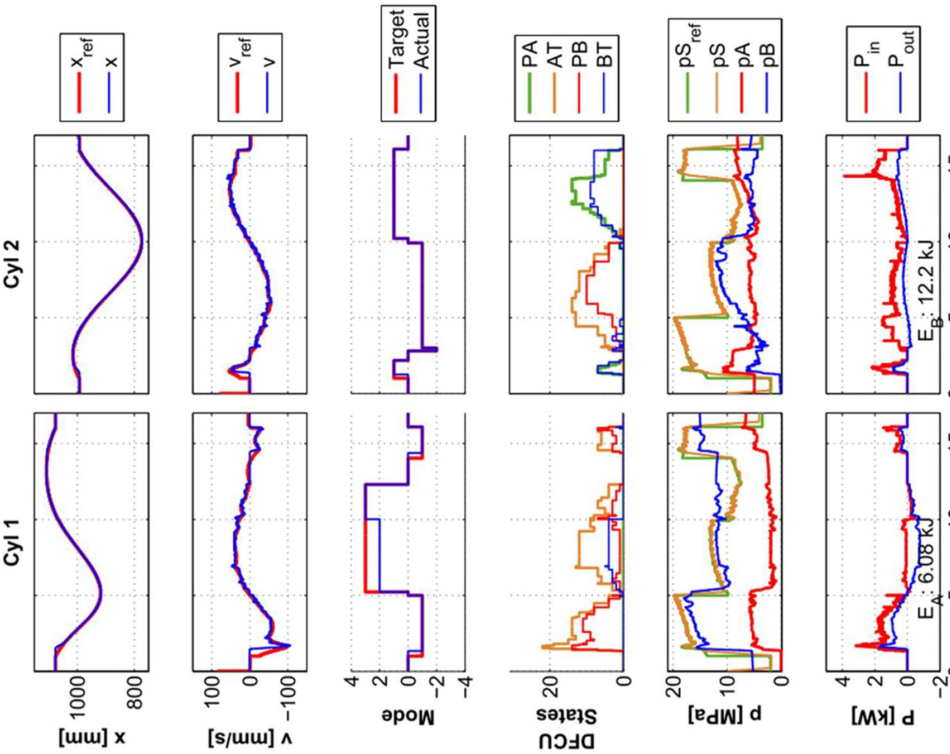

Figure 47. *MBC_{PresTankMod} controller using a pressurized tank line with common supply pressure.*

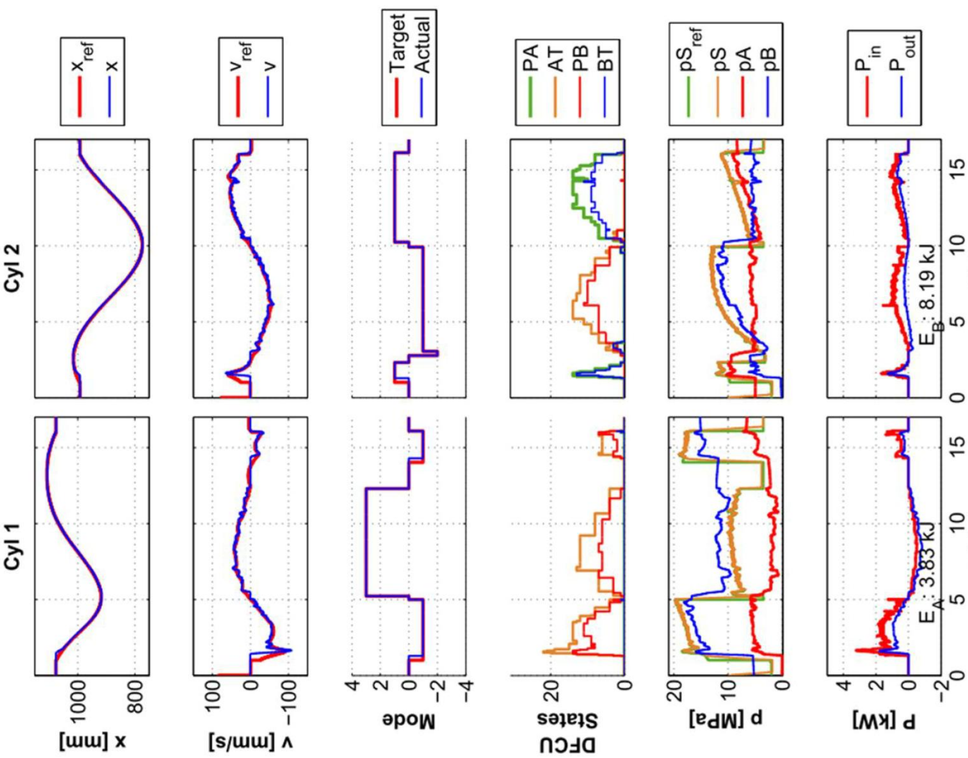

Figure 48. *MBC_{PresTankMod} controller using a pressurized tank line with two independent supply pressures.*

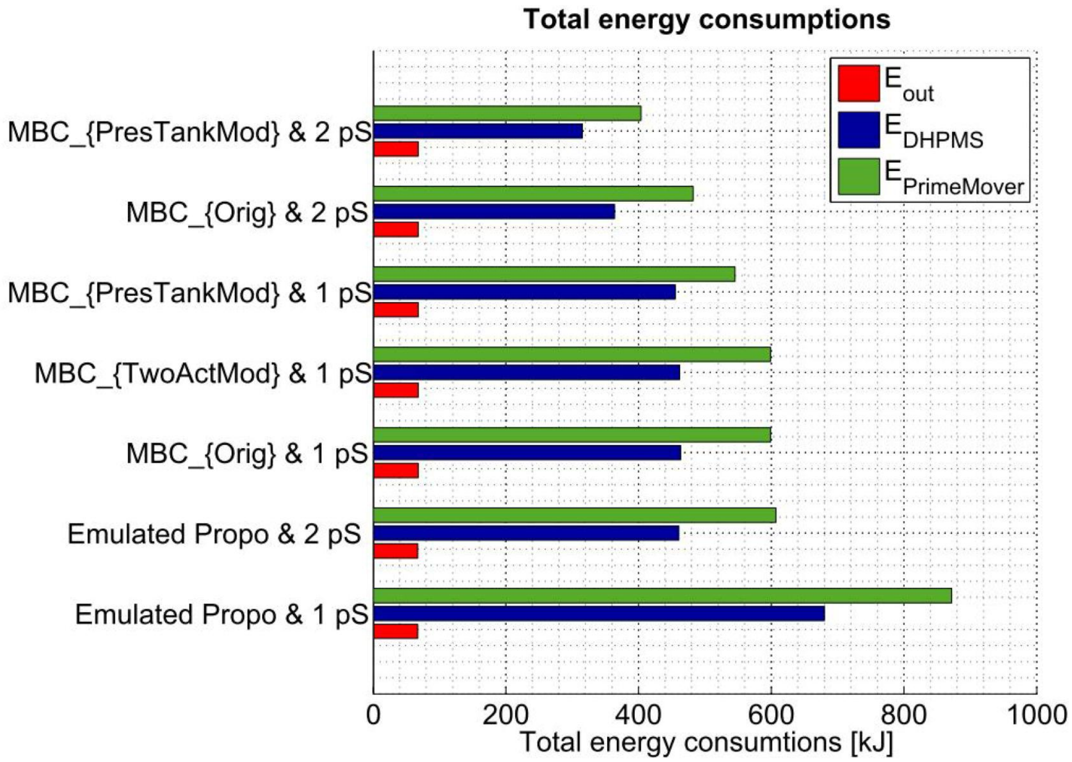

Figure 49. *Total energy consumptions of all 25 test cases. E_{DHMPS} is hydraulic energy taken from the ports of the DHMPS. $E_{PrimeMover}$ is the amount of energy taken from the prime mover axle.*

6.2. ANALYSIS OF SIMULATION RESULTS

The results in Figure 49 show that the most efficient way is utilizing independent pressures and a pressurized tank line. The biggest losses occur with the least intelligent controller with only one supply pressure, as suspected. All losses in the simulations are throttling losses only. Losses caused by cylinder friction are considered as part of output work. The trajectory is conservative and therefore no actual work is done.

Friction forces and leakage were not modelled in the DHPMS, its losses are then only throttling losses in the port valves. The prototype's total efficiency measured 70 – 85 %, and it remained at this level during the pumping, motoring, and transforming functions (Heikkilä et al. 2010). The simulated results show similar total efficiency, indicating the DHPMS's most significant losses as being throttling losses.

The supply pressure signals show that at the start and end, actuator 1 demanded higher pressure and in the middle actuator 2, creating pressure compensation losses in the actuator with too much available pressure. The two-supply pressure system required no pressure compensation at the inlet notch.

During moments of cavitation (occurring only with the highest load), the pressure differential over the valve dropped to zero, causing a velocity error. In addition, the cavitating cylinder became less stiff, as seen in Eq. 10.

$$K_h = \frac{A_A^2 \cdot B_{Aeff}}{(x \cdot A_A + V_{A0})} + \frac{A_B^2 \cdot B_{Beff}}{\left((L - x) \cdot A_B + V_{B0}\right)} \tag{10}$$

Here, A stands for area, B for bulk modulus, m for reduced mass, L for cylinder stroke, and V for dead volumes (Merritt 1967). While chamber is at cavitation stage, its bulk modulus is virtually zero. That is, one of the terms of the equation becomes zero, which clearly reduces the hydraulic spring constant and thus stiffness of the cylinder itself. Because the stiffness of one cylinder affects the system's natural frequencies, the system may at some operation point become unstable.

The controller is tuned to guarantee stability at the system's lowest natural frequency and is, because of filtering, robust against high frequency perturbations. Designed controller does not guarantee stability if the system's natural frequency drops, as can be seen in the simulation results on the emulated proportional cases, where cylinder 1 chamber A cavitates for some time. During cavitation, close to mid-simulations, the system turned momentarily unstable, as evidenced by the strongly oscillating chamber pressure. A model-based controller which applies independent metering can actively control chamber pressures and avoid cavitating cylinder chambers.

The more complex mode-choosing logic of the controller *MBC_{TwoActMod}* seemed to have no positive effect on the results. On further inspection, some simulation results show that the benefits, if any, are case sensitive. The hypothesis was that the *MBC_{TwoActMod}* should have resulted in fewer losses than the *MBC_{Orig}* with one supply pressure but not less than the *MBC_{Orig}* with two supply pressures. The hypothesis seems correct, though there is no significant difference in the net energies used between the single pressure cases of these controllers. This, together with the fact that the controller comes with more complex mode-choosing logic, does not support using this method for this application and trajectory.

The system with independent supply pressures enables use of the most efficient modes more often. Compare the modes in the results of the tilt cylinder in the case of the *MBC_{Orig}* at a time of about 3 s (Figure 44 and Figure 45): With only one supply pressure, the mode-choosing logic proposes the regenerative differential mode. Because the supply pressure is too high, by the demand of the other actuator the mode cannot be chosen as the actual one. In case of independent supply pressures, the pressures are at the correct levels as determined by the target modes, and the target mode can be selected as the actual mode. The same functionality can also be seen in the *MBC_{PressTankMod}* with actuator 1 at a time 5 to 10 s (Figure 47 and Figure 48). The proposed target mode is regenerative inflow/outflow, but with only one supply pressure, this mode cannot be selected as actual, and thus normal inflow/outflow is used instead. With two supply pressures, the pressures are set by the demands of both actuators, and the target modes can be chosen as actual more often.

The control algorithm of the DHPMS works well. By testing also different DHPMS controller modifications parallel to the DVS controllers, it was found out that accurate velocity estimate obtained from a model-based valve controller can be used to calculate a flow estimate for the DHMPS controller. As a result, a smoother pressure signal could be obtained by controlling only pressure by the feedback of the measured pressure signal. The flow estimate is thus used in a fashion similar to the general use of feed-forward in controllers.

7. MEASUREMENTS

In the measurements, state-of-the-art technology is compared to new methods. The measured results consist of three main parts: Verification of the controller design is performed and also the results were presented in short in (Karvonen et al. March, 2014). Validating the system functionality is the second phase and the results were presented in (Karvonen et al. June, 2014). The last part of the measurements is to measure reduction of losses by applying the proposed methods. The results on this have been published earlier in (Karvonen et al. September, 2014). For the thesis, a new set of measurements was done in a similar manner.

7.1. VERIFICATION OF CONTROLLER DESIGN

Controller design is verified by performing step- and trapezoidal response tests. The first reveal the performance and stability of the feedback controller. The second reveal the behaviour when both feedback and feed-forward controllers are working together for good velocity and position tracking.

During step response tests the load mass was 200 kg. The steps were performed in two independent coordinates in Cartesian reach space: In the area of highest [0.9, - 1.1] m and lowest [2.0, 0.8] m natural frequencies (the frequency map was presented earlier in Figure 24). The steps were done towards the selected point from distances of ± [2, 4, 8, 16] mm in actuator space. Both controllers were active during the test, but references were given to only one actuator at time. In Figure 50 the step responses are presented. The results are inverted for the sake of clarity, so it looks as if the steps are done from the same point, but that is not the case: rather the opposite.

From the step responses it can be verified that the controller is stable in both tested operation points with both valves and with both actuators. The proportional valve has some leakage, which causes minor drifting of position, but on the other hand it increases damping, which is well visible from the responses. Settling times are shorter in positions of higher frequencies.

The positioning steady state errors lie at predicted values: As previously presented in Figure 30, movement is enabled when a certain tolerance is crossed and it stays enabled until a smaller tolerance is crossed. The tolerance values are 4 mm/s to start movement and 2 mm/s to stop movement. With a gain of 4.22, velocity is enabled when there is 0.95 mm position error and it stays enabled until position error is less than 0.5 mm. However, in the case of the proportional valve, only a smaller tolerance is used without any hysteresis, which is possible in the case of an analogue valve. The calculated values correspond well with the measured results.

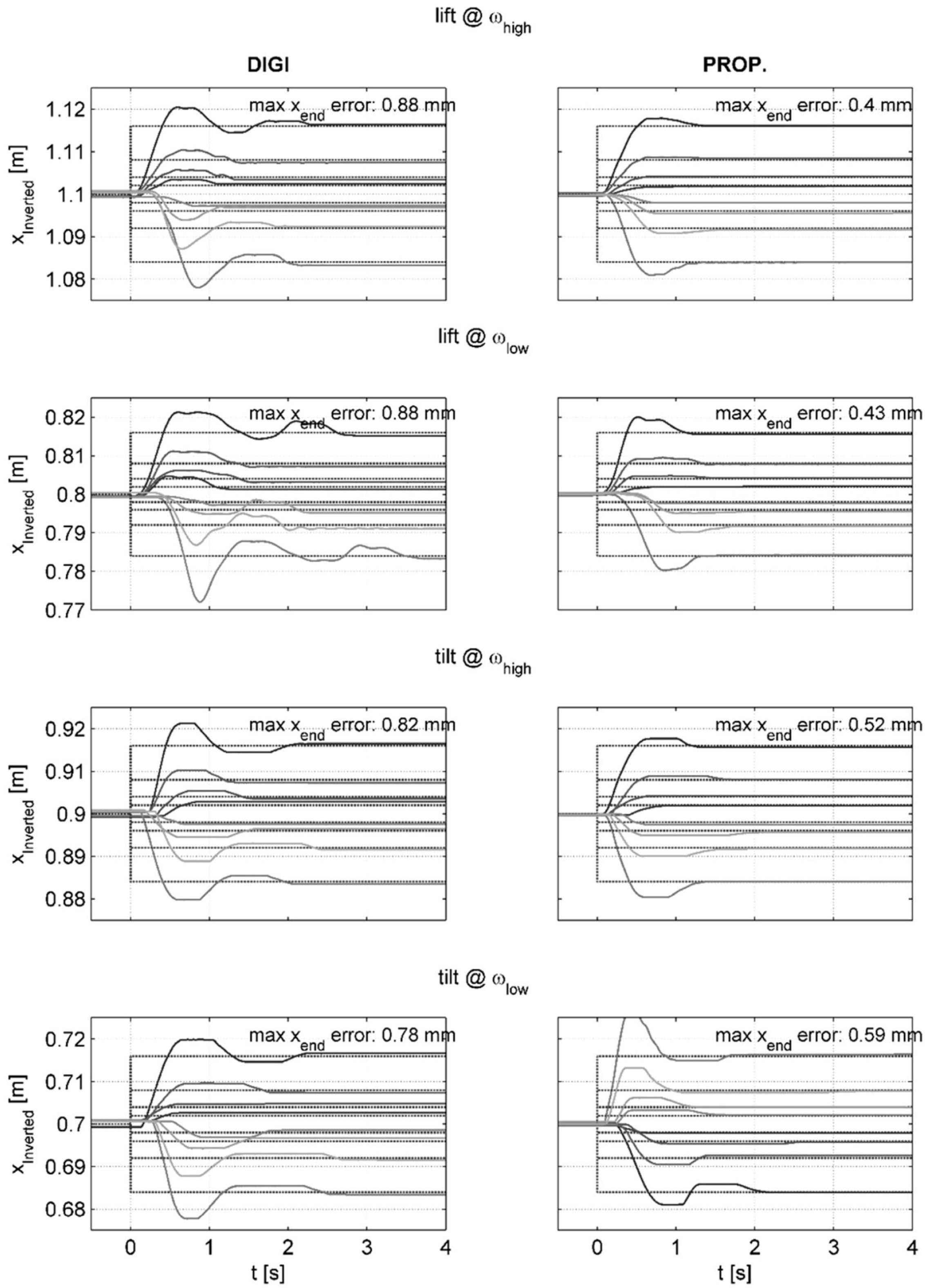

Figure 50. *Step responses of two actuators in two operation points in Cartesian space, locations on minimum and maximum system natural frequency, with digital and proportional valves. The plot is inverted for improved clarity; in reality steps are done towards the point, not away from it.*

Trapezoidal responses give insight into the velocity tracking performance. The tests are made in two operation regions; the starting points are the same and were targets during the step response tests. Velocity tracking was acceptable in both test areas, but better around the area of higher natural frequency. Figure 51 presents an example of the differences in digital and proportional control in the case of trapezoidal trajectory.

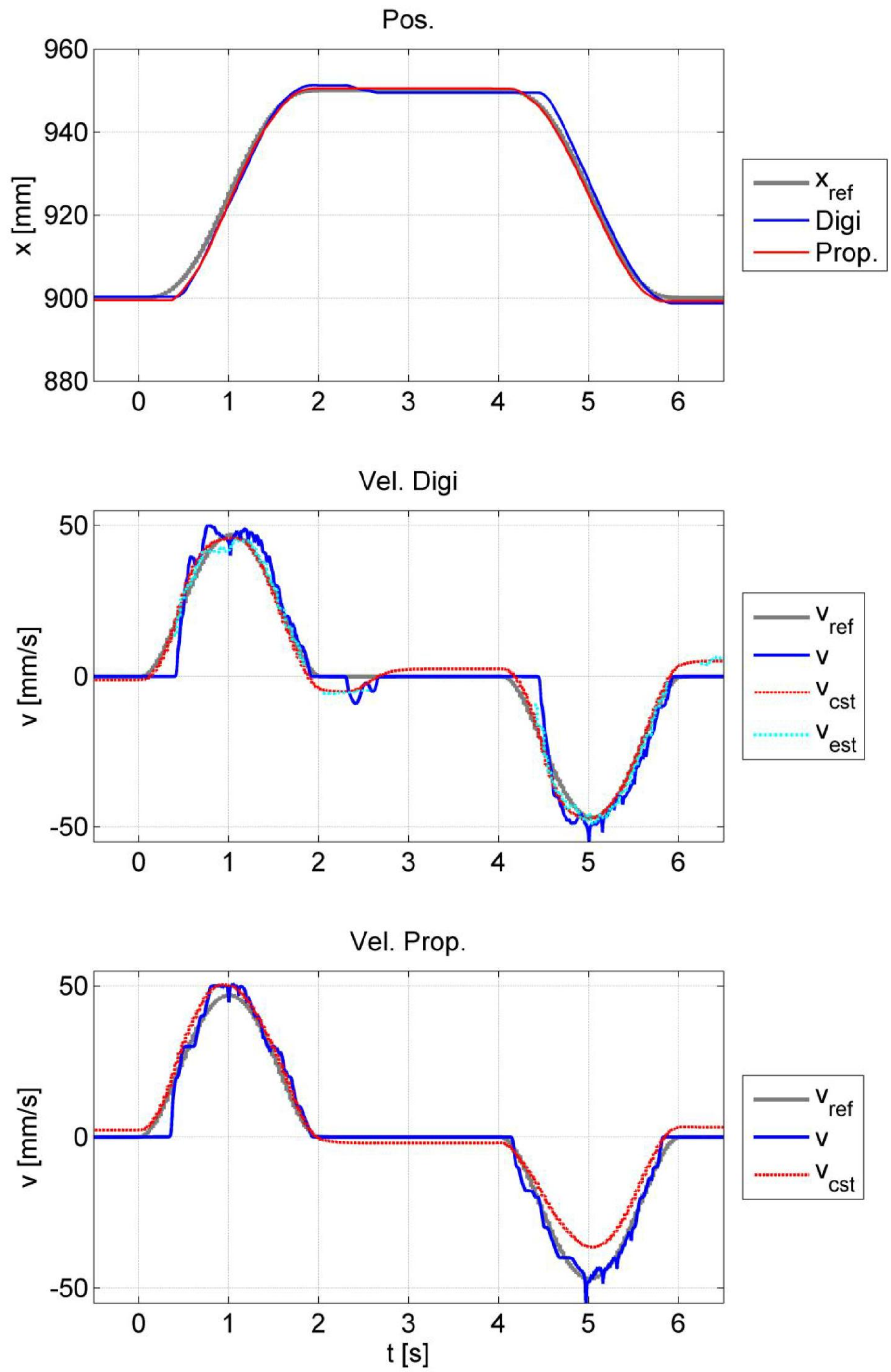

Figure 51. *Trapezoidal response test*

The velocity tracking characteristics of the proportional- and DVS controlled systems have differences. The digital system has lower damping, as discussed earlier, and it is seen from the overshoot. Also the velocity threshold is bigger in the DVS case, which is seen from the larger delay: Motion is enabled when v_{CST} crosses a magnitude of 4 mm/s and stays enabled until the magnitude drops below 2 mm/s. The DVS controller calculates the velocity estimate, which is *NaN* while motion is not enabled. During motion the velocity matches well with the estimate. This behaviour is visible from subplot 2 of the figure.

The velocity tracking in different directions is similar in the DVS case, but in the case of proportional valves the velocity tracking is in different directions. During this test the load force direction was restricting of positive movement and overrunning with negative. The direction of the load force does not affect the DVS system as it utilizes independent metering, and optimal opening ratio is used constantly to match the load. In the proportional case the overrunning load causes the cylinder to move faster than it should, which then causes P-term in the feedback controller to decrease the value of the v_{CST}, which is seen in the lowest subplot in Figure 51. Both systems work well enough with trapezoidal tests.

7.2. VERIFICATION OF THE SYSTEM FEATURES

The system provides independent supply pressures to the actuator lines and during recuperation power is transferred and transformed from one supply line to another. When used with DVS this should enable power to be transferred and transformed in the DHPMS when there is a positive power flow to one supply line while there is negative from the other. The described functionality is presented with various signals in Figure 52 and by the power flows in Figure 53. The presented data does not give perfect tracking or otherwise "optimal" results, but demonstrates well the functionality and differences between the systems. Explanations about the required functionality and related effects are followed after the figures.

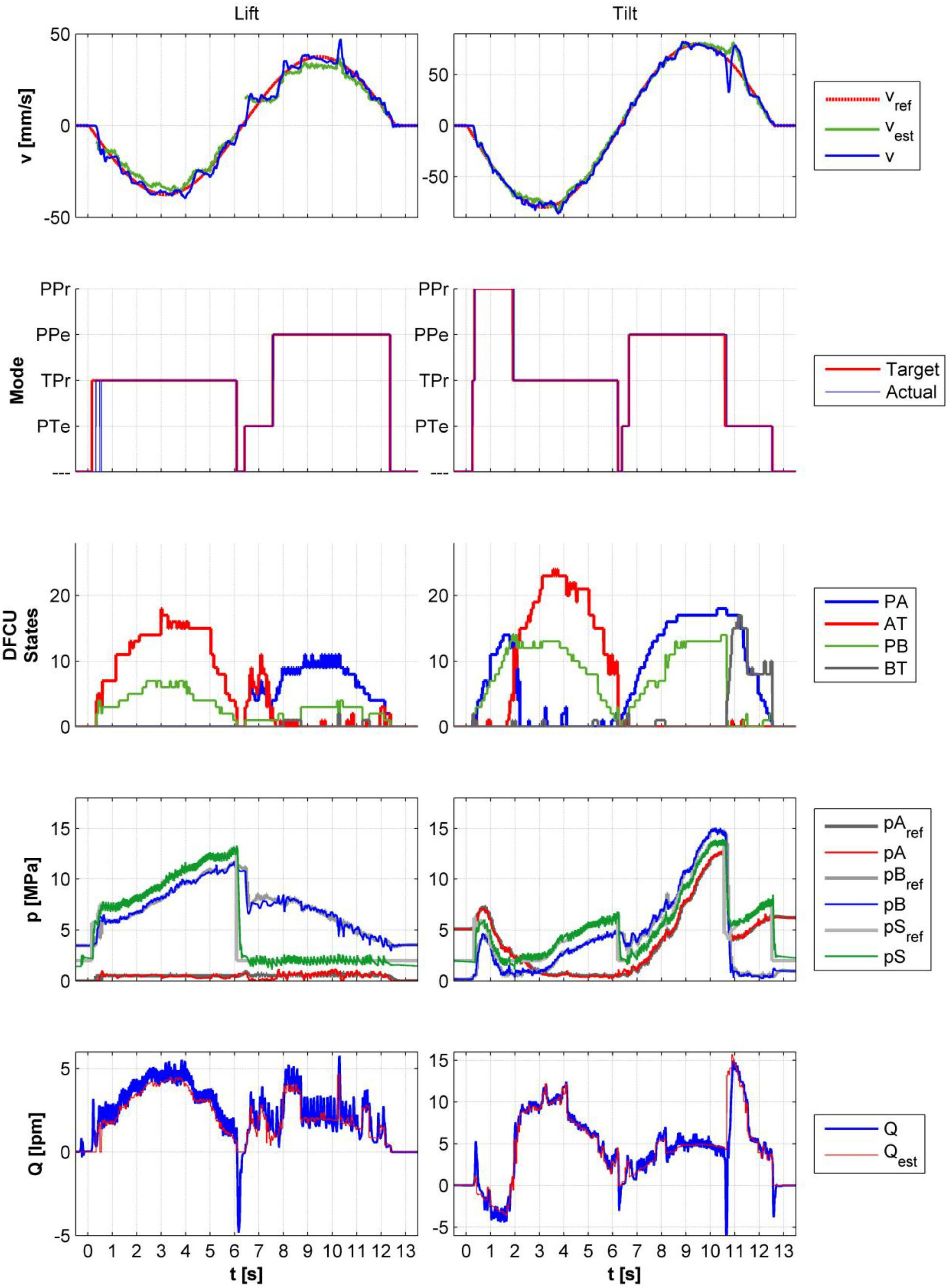

Figure 52. *Sinusoidal test trajectories in the case of DVS & independent supply pressures*

1. The 1st row of subplots presents the actuator velocities and references. The velocities follow the reference quite accurately most of the time. The oddness at the tilt actuator at the time of 11 s is explainable: the sudden change in flow reference at the instant of mode switching and DHPMS dynamics affect the performance (details follow).

2. The 2nd row shows in which modes the DVS works. The target and actual modes are both presented. The lift DVS works in three different modes while the tilt DVS works in all four modes.

    a. The lift cylinder has a constantly overrunning load towards the extending direction; thus it uses TPr for retraction, but extending movement starts with PTe because acceleration of the inertial load creates a high load force. Quite soon the mode is changed for PPe.

    b. The tilt cylinder has loading which changes in both magnitude and direction. The trajectory starts with overrunning loading towards the retracting direction while also the reference velocity is towards the retracting direction. That is, recuperating PPr mode is feasible and applied after a short acceleration in the more forceful TPr mode. The DVS stays in recuperating mode until pressure difference over the A-to-T control notch reaches the allowed minimum pressure differential. This happens when the load force magnitude drops, as it does when the load which moves in a pendulum-like path approaches its position of minimum potential. When recuperation is not feasible anymore, TPr mode is used again and soon after it is the only valid option as the load direction changes to restrictive.

       Towards the extending direction the tilt cylinder at first has an overrunning load, and differential mode (PPe) is used after a short acceleration in inflow-outflow mode (PTe). The load force changes direction and magnitude during motion. For demonstration purposes the maximum pressure is limited to 15 MPa, and when the cylinder cannot generate enough force at the maximum pressure in the PPe mode, the mode is set to PTe.

3. The 3rd row shows individual DFCU states within a DVS. From the signals the variable opening ratio is visible, as are also online mode switching and minor utilization of cross flow.

4. The 4th row tells about system pressures. Supply pressure references are set by the target modes. Clearly, supply pressures are independent and follow their own references with good accuracy. Chamber pressures and their references (or estimates) are also shown and clearly the measurements match the references, which indicate that the model based optimal controller performs well. Step like changes in supply pressure at the instants of mode switching cause no oscillation of the supply pressure.

5. The 5^{th} row contains flow references and flow rates in the supply lines. The flow estimate from the DVS controller is used as flow reference for the DHPMS. The flow rates follow the reference well. The only differences occur at the instants when supply pressure level is lowered while the velocity reference remains constant (e.g. when entering stop mode). This is natural as only the flow through the DVS is considered by the model based DVS controller.

   The DHPMS controller takes account of supply line capacitance and therefore there is no overshoot when controlling pressure and flow. During the mode switches the flow rate changes rapidly. An especially big change occurs when the tilt actuator changes mode from recuperation to normal. During recuperation the flowrate is negative, which means the DHPMS is motoring from the line. At the same time the flow rate required by the lift actuator is positive, but there is a different pressure level so transformation occurs and power flows from one actuator line to another. During the flow rate transient there is a difference between the flow and flow estimate as is also the case with velocity estimate and velocity.

The power flows of the results in the previous figure are independently presented in Figure 53. The input-, hydraulic, and output powers are presented and analysed. Figure 53 shows clearly the recuperating operation at the beginning of the run on the tilt actuator. The level of recuperated power almost matches the level required by the lift actuator and therefore the prime mover power is temporarily almost at the level of idling (about 0.4 kW, idling power seen before and after the reference). The power graphs show that there is a good portion of negative power at the lift actuator during positive velocity. Recuperation from extending movement is not possible without the pressurized tank line which was not available for the measurement setup.

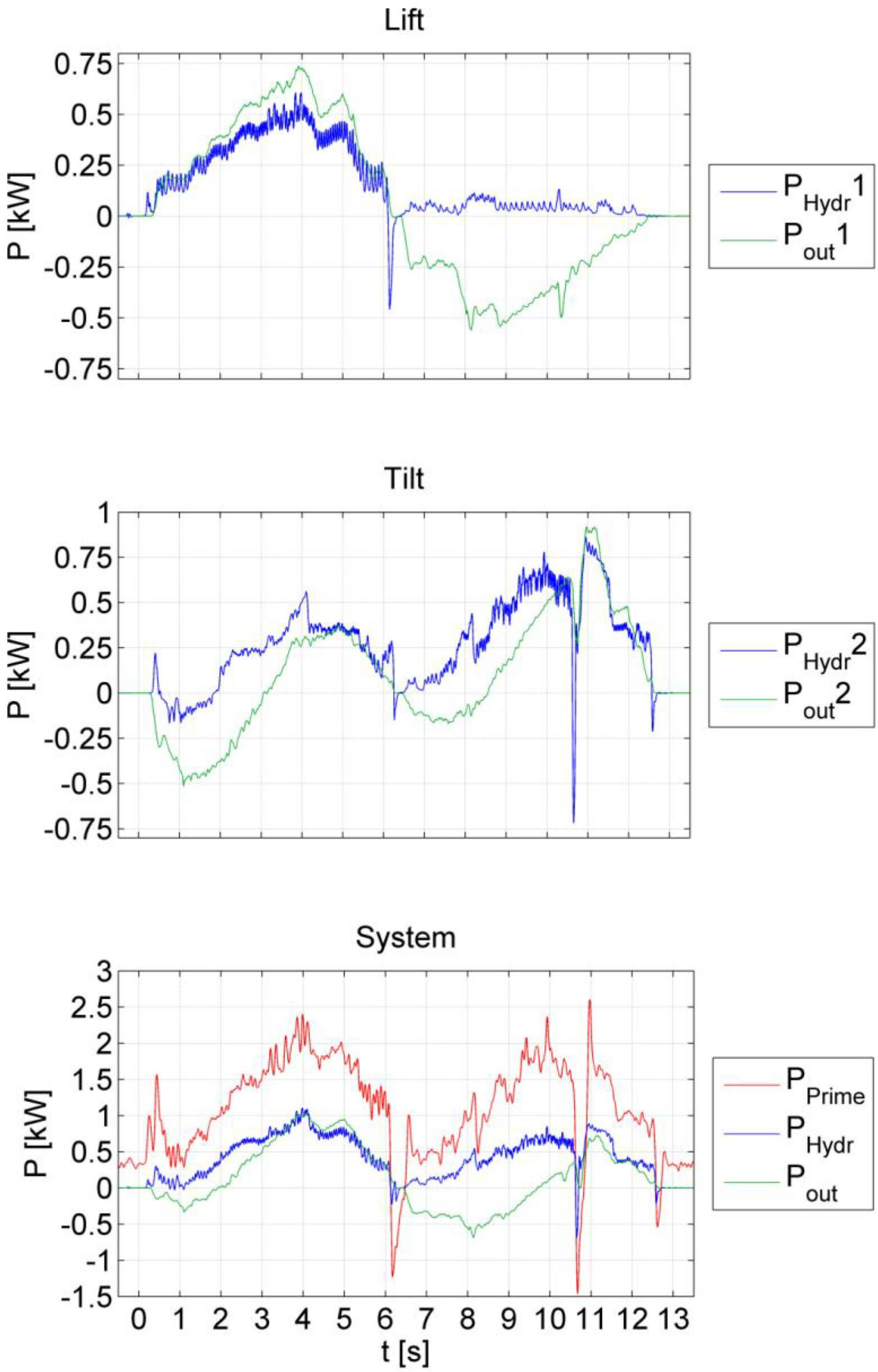

Figure 53. *Power flow in the case of DVS and independent supply pressures. Recuperation and transformation occur at the beginning of the trajectory.*

The previously presented case had independent supply pressures. The same trajectory with common supply pressures is driven and the results are shown in Figure 54. Analysis follows after the figure.

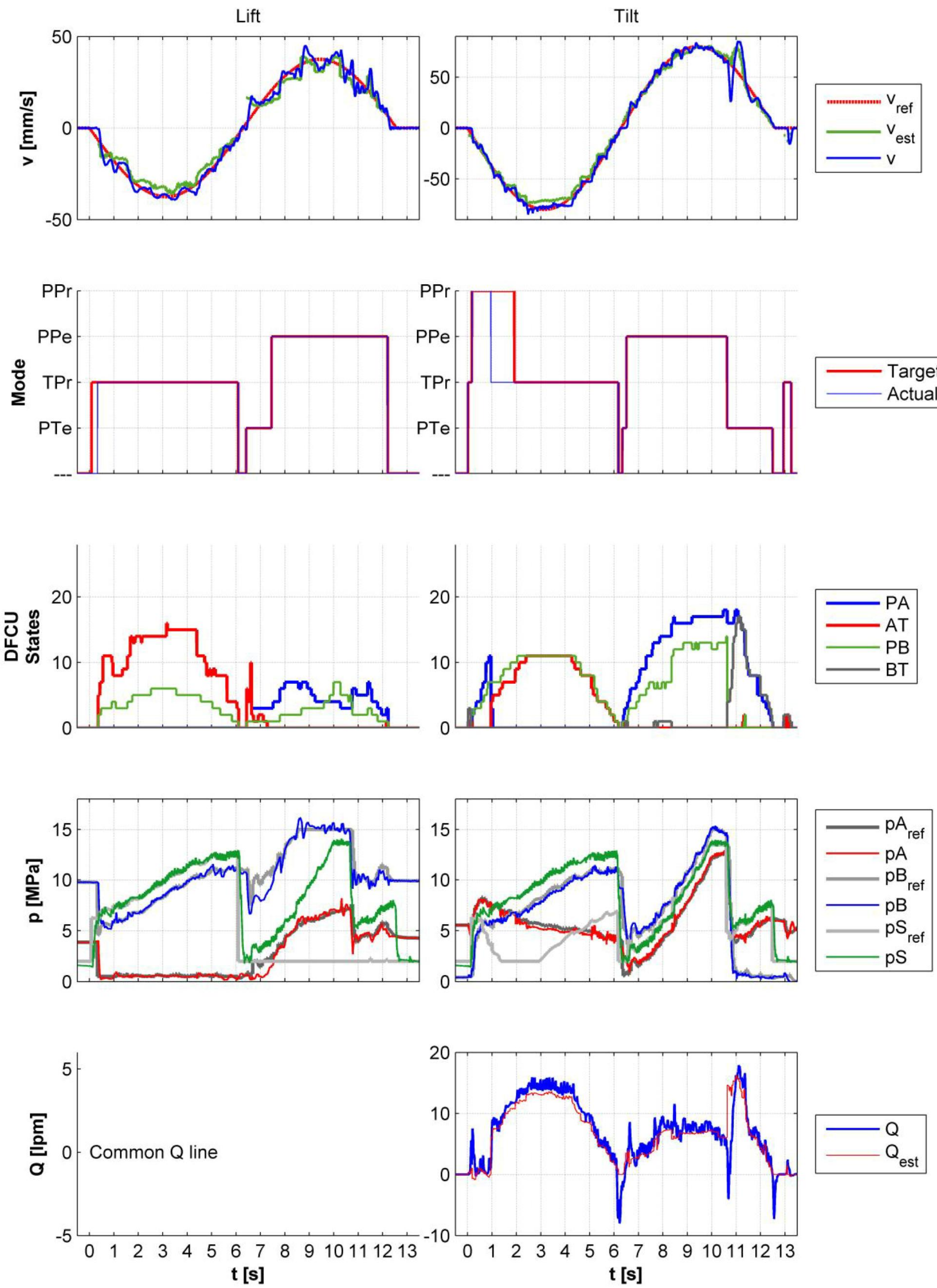

Figure 54. *Sinusoidal test trajectories in the case of DVS and common supply pressures*

The common supply line is supplied by highest demanded pressure level and pressure compensation is occasionally required. The DVS applies independent metering and pressure compensation is realized by changing the opening ratio. This feature is seen by comparing DFCU states in independent and common supply pressure cases. The common supply pressure reduces the feasibility of recuperating mode at the tilt actuator, which is seen by comparing the mode plots of the systems. The target mode is not necessarily feasible on the actuator, which has lower supply pressure demand and a different mode is chosen as actual. Generally speaking, both systems seem to work as supposed to, as all signals follow their references quite well, except for the supply pressures that always follow the higher reference because of the LS principle. As a result, it can be stated that utilization of all the potential of DVS independent supply pressures is required. The same result was also found from the simulation study presented earlier.

The same trajectory is driven also with proportional valves in both the independent and common supply pressure cases. The case of independent supply pressures is shown in Figure 55 and the case of common supply pressures in Figure 56. Analysis follows after the figures.

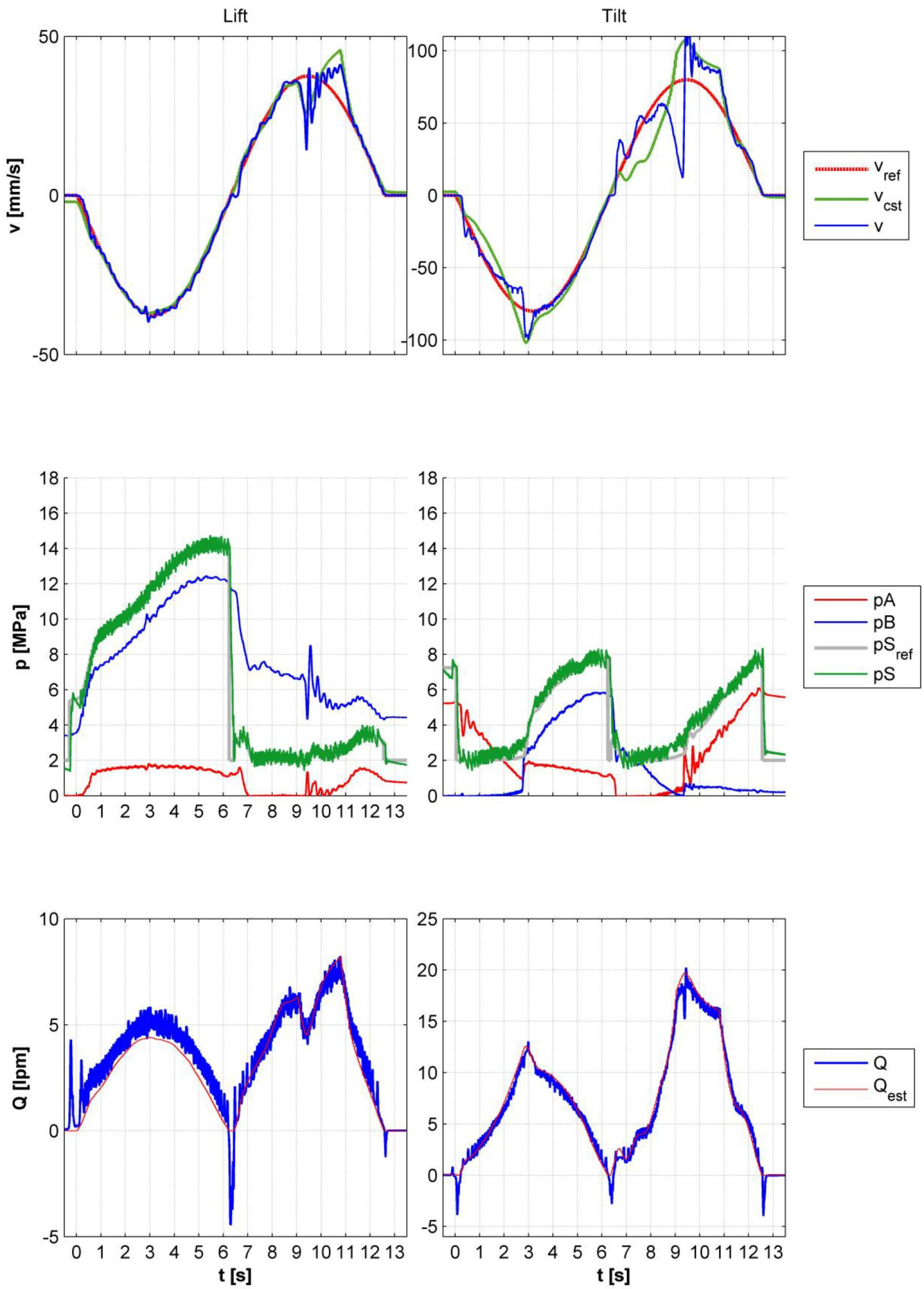

Figure 55. *Sinusoidal trajectory in the case of proportional valves and independent supply pressures*

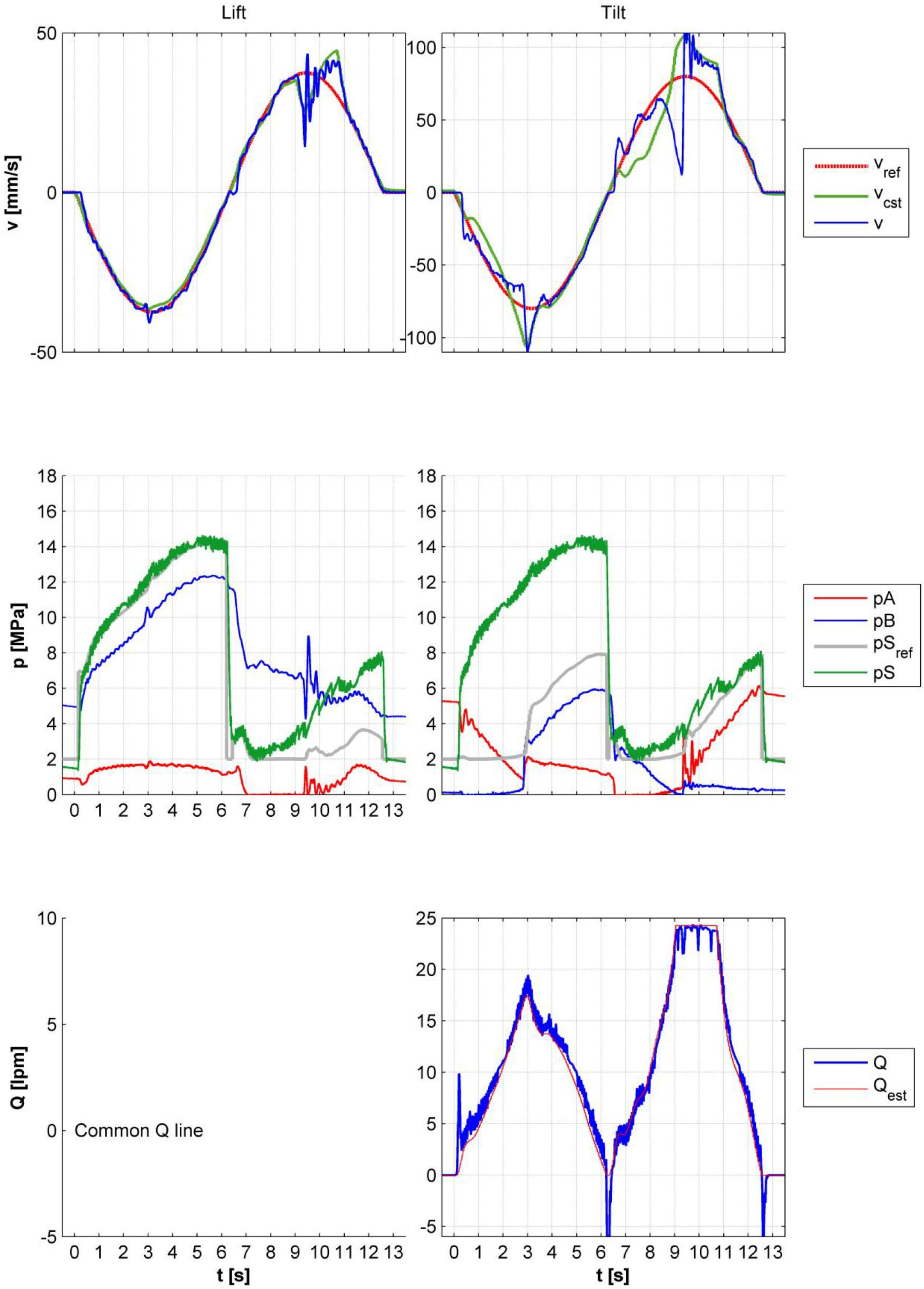

Figure 56. *Sinusoidal test trajectories in the case of proportional valves and common supply pressure*

The proportional valves have high bandwidth hydro-mechanical pressure compensators as it is meant to be used in multi-actuator LS-systems. The result that there is no difference in system behaviour between different supply pressure cases can be explained by this. When comparing different valve systems the differences are clear. The proportional valves are not controlling chamber pressures but only flow and spools should be matched to the worst case situation. However, it was not possible to obtain the wanted spools, but something "suboptimal" was obtained, with cavitation occuring in the cylinders (pressure drops to zero). This leads to controllability problems, especially with heavily overrunning loads during which cavitation occurs. When the load is restrictive controllability is good: even better than with the DVS, the behaviour of which, however, is quite immune to load force magnitude and direction due to the independent metering capability.

The flow saturation limit is reached with proportional valves but not with digital valves. The control functions include a flow-sharing function which is seen as a drop of v_{CST} at the lift actuator. The effect is in use also on the tilt actuator, but it is not visible as the load is temporarily out of reference and therefore the v_{CST} is highly affected by the feedback P-term of the controller.

The DVS is utilizing differential modes, which require less flow than the inflow-outflow modes used constantly by the proportional valves. This result indicates that the possibility to change operation modes in valves could also bring the possibility to downsize a pump, especially in systems in which high force and high velocity are not required simultaneously. Downsizing may provide a reduction of losses as typically efficiency on partial loads is relatively lower than on the nominal output. The functionality of the system provides sufficient method to test the efficiencies on different loads and trajectories.

7.3. RESULTS ON ENERGY CONSUMPTION

The measured energy consumptions are calculated by means of trapezoidal integration of the power, which is calculated from certain measured signals.

1. Mechanical input power is measured on the prime mover axle. Torque sensor outputs both angular velocity and torque.

2. Hydraulic power in supply lines is measured. Flow meters are located both supply lines. Supply pressure transducers are located in the DVS manifolds. When common supply pressure is used, all flow is routed via lift cylinder line.

3. Output power is calculated from cylinder velocity measurements and cylinder force. The force is calculated from chamber pressures, multiplied by cylinder areas. Since the trajectory is closed path, the output work done consists only of friction based power loss.

The energy consumptions of previously explained sinusoidal cases are shown in Figure 57. Also the losses are presented.

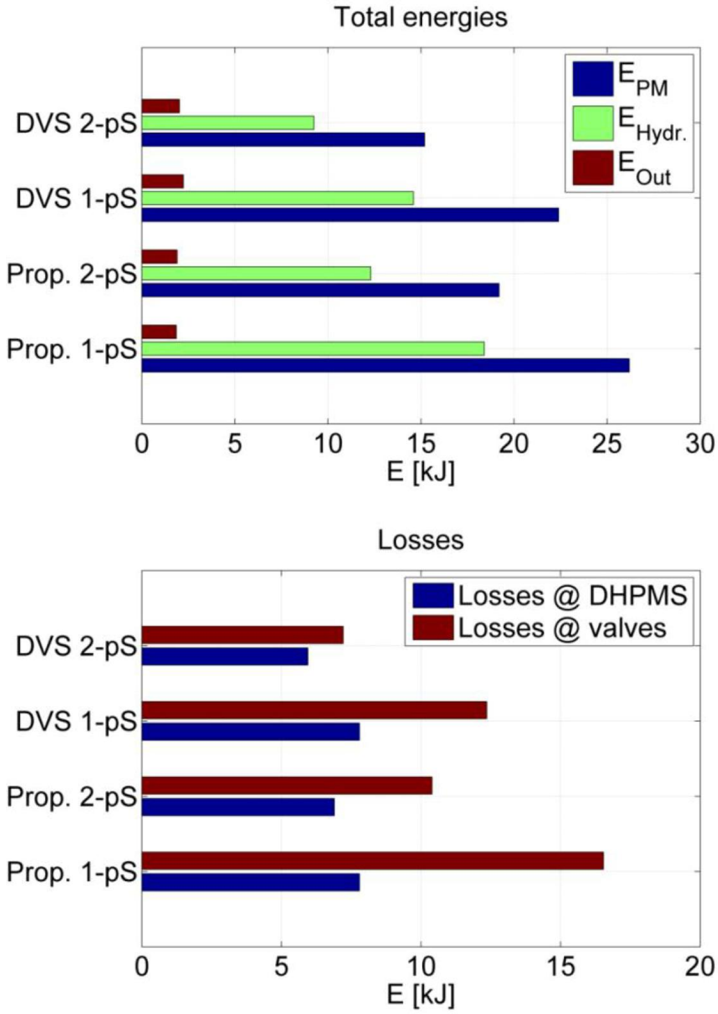

Figure 57. *Energies and losses at tested sine references. Output work consists of friction only.*

The upper subplot in Figure 57 shows mechanical input energy from the DHPMS shaft, hydraulic energy of the supply lines and output energy which is calculated from the cylinder pressures and actuator velocity. The lower subplot shows the amount of losses in the DHPMS and in DVSs.

The energy consumption is at the lowest in the case of independent supply pressures with DVS. The second is the proportional valve with independent supply pressures and later on the cases with common supply pressure in the same order of valves. The losses graph shows clearly how metering losses (including compensation losses) decrease by utilization of independent supply pressures. The difference of metering losses between DVS-2-pS and DVS-1-pS is smaller than the difference between Prop.2-pS and Prop.1-pS, which is understandable as the flow rates were smaller in the DVS cases because of good use of differential modes.

Losses at the DHPMS are not strongly affected by different cases, but in the case of independent supply pressures the losses are slightly smaller. This may be the result of smaller leakage flow as the prototype unit tends to leak, especially during high supply pressures, and the leak is smaller if there is lower pressure even in one of the supply lines.

Answers to the main research questions may already be found. For more conclusive and generalizable results about the amount of reduction of losses, the wider setup of different cases is measured. The measurements setup is as follows, and the tests are based on certain more specific research questions:

- Three load masses: 0, 100 and 200 kg
  - Does load affect, and if so, how?
- Two trajectories: A circular trajectory clockwise (CW) and counter clockwise (CCW). Trajectory in Cartesian reachspace presented in Figure 58.
  - Does a different combination and order of loadings affect? Are the results generalizable?
  - Three different speeds for a conclusive set of measurements.
- Two valve systems: DVS and proportional
  - How much loss can be reduced by independent metering and utilization of different modes?
- Two supply methods: Independent and common supply pressures
  - How much loss can be reduced by independent supply pressures?
- 3 measurements in each case.
  - Average of three measurements increases reliability.

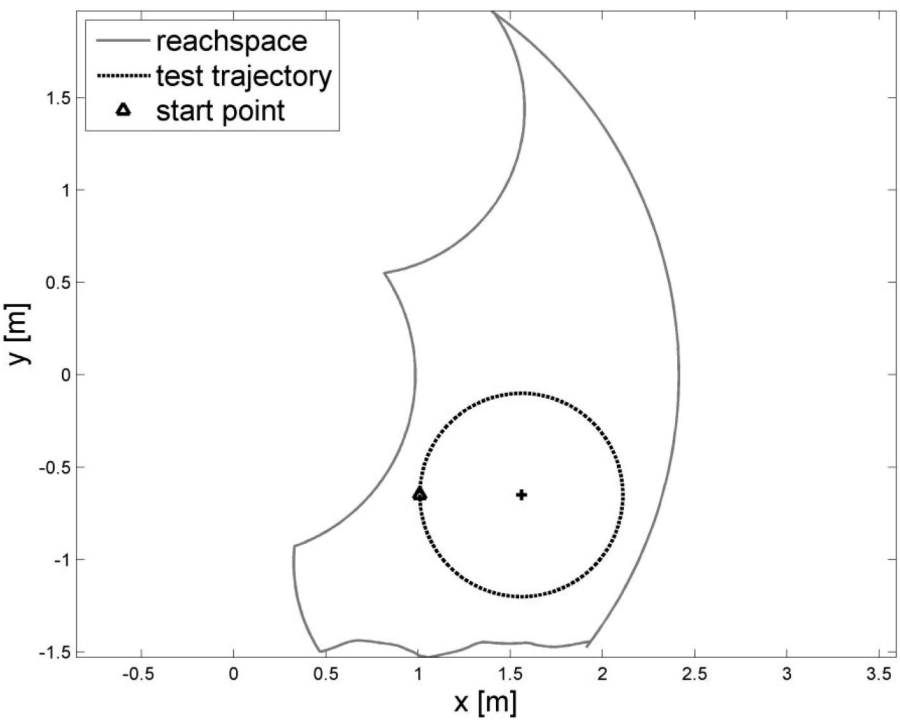

Figure 58. *Circular reference trajectory in the reachspace*

The motion control performance during the measurements was similar in every case with 0 and 100 kg loads. The CCW trajectory started smoother as the pressure levels required on start-up were lower than in the CW case. The opening ratio of proportional valve spools was not sufficient to provide cavitation free actuation with a 200 kg load. This affects the velocity control performance, as the controllability is temporarily poor during overrunning loading. This flaw cannot be corrected by changing gains in the controller as it is possible only by changing the opening ratio. The measured trajectories and various signals are presented in Appendix D. The appendix contains the CCW path driven at the slowest trajectory. The differences are clearest in that particular case.

The averages of the sets of three measurements are presented. In Table 7 energies are presented in all the measured cases. The energies presented are from the prime mover axle (E_{PM}), the sum of hydraulic power fed to the supply lines (E_{IN}) and the sum of the cylinders' work (E_{OUT}), which consists of friction only.

Table 7. *Energies of the system at the measured cases. Averages of three measurements*

	Load	0 kg				100 kg				200 kg			
Traj.	**[kJ]**	**Prop. 1 pS**	**Prop. 2 pS**	**DVS 1 pS**	**DVS 2 pS**	**Prop. 1 pS**	**Prop. 2 pS**	**DVS 1 pS**	**DVS 2 pS**	**Prop. 1 pS**	**Prop. 2 pS**	**DVS 1 pS**	**DVS 2 pS**
Fast CW	E_{PM}	21.21	16.94	14.03	12.54	22.76	17.74	16.37	13.18	25.39	18.48	19.05	14.92
	E_{IN}	15.78	12.30	9.85	8.06	17.21	12.61	10.97	8.58	19.09	13.11	13.57	10.00
	E_{OUT}	1.47	1.43	1.42	1.38	1.67	1.70	1.79	1.65	1.92	1.91	2.07	1.96
CCW	E_{PM}	22.31	17.47	15.06	12.25	25.23	18.67	18.71	12.88	26.55	19.03	22.58	14.86
	E_{IN}	16.67	12.66	10.60	7.77	19.14	13.00	12.76	8.17	19.53	12.98	16.17	9.51
	E_{OUT}	1.44	1.41	1.45	1.37	1.58	1.64	1.83	1.56	1.85	1.86	2.13	1.83
Med. CW	E_{PM}	21.24	17.69	14.82	12.84	22.99	18.36	16.68	13.95	25.95	19.32	19.15	15.92
	E_{IN}	14.79	11.79	9.46	7.48	16.30	11.98	10.46	8.06	18.14	12.62	12.56	9.81
	E_{OUT}	1.20	1.20	1.17	1.09	1.41	1.44	1.54	1.38	1.68	1.69	1.77	1.62
CCW	E_{PM}	22.36	17.98	15.41	13.20	25.51	19.06	19.30	14.00	28.16	20.75	23.27	16.09
	E_{IN}	15.62	11.97	9.96	7.66	18.14	12.28	12.23	7.90	19.39	13.01	15.47	9.27
	E_{OUT}	1.19	1.18	1.18	1.19	1.36	1.40	1.58	1.34	1.66	1.66	1.89	1.61
Slow CW	E_{PM}	22.16	18.82	15.99	14.21	24.29	19.53	18.53	15.95	27.87	20.82	21.39	17.81
	E_{IN}	14.03	11.29	9.28	7.37	15.69	11.50	10.41	8.31	17.81	12.22	12.90	9.86
	E_{OUT}	1.04	1.04	0.96	0.89	1.25	1.28	1.34	1.25	1.53	1.55	1.66	1.46
CCW	E_{PM}	23.26	18.98	16.55	14.12	26.54	20.41	20.45	15.81	29.77	22.56	25.55	18.32
	E_{IN}	15.01	11.30	9.54	7.25	17.15	11.66	11.57	7.86	18.38	12.46	15.55	9.35
	E_{OUT}	1.02	1.02	0.98	0.89	1.23	1.25	1.42	1.20	1.57	1.59	1.88	1.53

From the energies it can be seen that the new methods are more efficient. Relative results on the reduction of losses are calculated and presented in the following tables. The reference system is always a single pressure system with proportional valves, which emulates the state of the art LS-system. In Table 8 the valves are compared. Energy consumption is reduced because of independent metering and the use of different operation modes. In the second table, Table 9, common and independent supply pressures are compared. The reduction of compensation losses is presented. The third table, Table 10, presents the energy saving potential if both proposed improvements are applied.

Table 8. *Reduction of energy consumption by applying independent metering*

Independent metering		0 kg	100 kg	200 kg	ave:
Fast	**1 pS, CW**	33.8 %	28.1 %	25.0 %	26.2 %
	2 pS, CW	26.0 %	25.7 %	19.3 %	
	1 pS, CCW	32.5 %	25.8 %	14.9 %	26.0 %
	2 pS, CCW	29.9 %	31.0 %	21.9 %	
Medium	**1 pS, CW**	30.2 %	27.4 %	26.2 %	25.1 %
	2 pS, CW	27.4 %	24.0 %	17.6 %	
	1 pS, CCW	31.1 %	24.3 %	17.4 %	24.7 %
	2 pS, CCW	26.6 %	26.5 %	22.4 %	
Slow	**1 pS, CW**	27.9 %	23.7 %	23.2 %	22.1 %
	2 pS, CW	24.5 %	18.3 %	14.5 %	
	1 pS, CCW	28.8 %	22.9 %	14.2 %	22.1 %
	2 pS, CCW	25.6 %	22.5 %	18.8 %	
	ave:	28.7 %	25.0 %	19.6 %	**24.4 %**

Table 9. *Reduction of energy consumption by applying independent supply pressures*

Sep. supply pres.		0 kg	100 kg	200 kg	ave:
Fast	Digi CW	10.6 %	19.5 %	21.7 %	23.4 %
	Prop. CW	20.2 %	22.1 %	27.2 %	
	Digi CCW	18.6 %	31.2 %	34.2 %	26.7 %
	Prop. CCW	21.7 %	26.0 %	28.3 %	
Medium	Digi CW	13.4 %	16.4 %	16.9 %	21.1 %
	Prop. CW	16.7 %	20.1 %	25.6 %	
	Digi CCW	14.3 %	27.4 %	30.9 %	24.0 %
	Prop. CCW	19.6 %	25.3 %	26.3 %	
Slow	Digi CW	11.1 %	13.9 %	16.7 %	17.0 %
	Prop. CW	15.1 %	19.6 %	25.3 %	
	Digi CCW	14.7 %	22.7 %	28.3 %	21.9 %
	Prop. CCW	18.4 %	23.1 %	24.2 %	
	ave:	16.2 %	22.3 %	25.5 %	**21.31 %**

Table 10. *Reduction of energy consumption by applying both methods*

Apply both		0 kg	100 kg	200 kg	ave:
Fast	CW	40.89 %	42.07 %	41.23 %	43.7 %
	CCW	45.09 %	48.96 %	44.04 %	
Medium	CW	39.56 %	39.33 %	38.64 %	41.1 %
	CCW	40.98 %	45.10 %	42.87 %	
Slow	CW	35.88 %	34.34 %	36.10 %	37.4 %
	CCW	39.29 %	40.41 %	38.46 %	
	ave:	40.5 %	42.0 %	40.6 %	**40.73 %**

There are differences in energy saving percentages in the case of different load masses. Energy saving potential with the DVS is smaller with higher loads. This is so because the proportional valve is working optimally, that is there is no cavitation nor unnecessary high back pressures, with a 100 kg load. At a load of 200 kg, the controllability with proportional valve is weak because of cavitation. At 0 kg unnecessary throttling occurs constantly, which causes losses. Even though there are some errors in velocity tracking between the cases, the cylinders make the same strokes anyway. That is, the paths are similar in all cases, so they are comparable. Independent metering reduces losses by optimizing the opening ratio constantly. Also the DVS is utilizing differential modes, which affect the losses.

7.4. MEASUREMENT RELIABILITY

The transducers which are used contain both analogue and incremental sensors. Torque and pressures are measured by strain-gauge based transducers. Prime mover angular velocity and flow rate are measured by rotary encoders. The cylinder position is measured by rotary encoders with pull-wires. The transducers are listed in Table 1.

Incremental encoders have weakness when measuring velocity, as the measurement uncertainty tends to be infinity at zero velocity and zero at infinite velocity. Also the data processing and filtering method affect the uncertainty, so classic style measurement uncertainty is not definable.

The comparative results are firstly integrals, which filter out zero biased disturbances, and secondly are the averages of three measurements which reduce random error. Thirdly, the results are compared to each other which all are measured with the same instruments and in the same operation conditions. Therefore, it can be claimed that uncertainties from the instrumentation are negligible. The simulated values correspond well with the measured values, which also stand for sufficient measurement accuracy; in the simulations there are no measurement uncertainties.

8. DISCUSSION

Digitalization is a major trend. Generally the term means that features are shifted from hardware into software and thus new possibilities emerge. Transition from analogue and manual includes discretization and/or virtualization. In this thesis, pioneering work about mapping of possibilities of digitalization into machine automation, particularly work hydraulic circuit, is conducted.

Understanding of basic physics gives engineers clear idea why and where energy losses occur in hydraulic circuits. Therefore the sources of inefficiency tackled in this thesis are not anything new. However, the solution for improvements through digitalization of not only the software or control code but also the whole hardware is revolutionary; it has not been attempted before.

The base theory assumes that independent supply pressures enable bigger energy savings on the highest loads. This is natural as the variation of load pressure magnitude is greater on higher loads (cf. torque at a pendulum joint vs. the load and the angle). This is also shown by the measurements. The trajectory contains both positive and negative loads and the load affects the amplitude. In the case of the tilt actuator, the movement is pendulum-like. It is worth noticing that the simulated results presented in (Karvonen et al. 2011), which predicted the 22% energy saving potential by the separation of the supply pressures, matches well with the measured values.

Applying both independent metering and independent supply pressures, slightly over 40 % of energy can be saved. In the simulations presented in (Karvonen et al. 2014), 46 % energy savings were achieved. The difference most likely originates from the fact that the cylinder chambers were occasionally at the cavitation stage in the measurements. In the simulation model, the fixed opening ratio was freely adjustable and selected so that it prevented cavitation. In the measurements, this was not easily achievable and there were a finite number of spools available in reasonable time. During cavitation, the oil is not taken from the supply line, but from the return line through anti-cavitation valves. The energy calculations consider only energy flow via the supply lines and therefore the cavitating cylinder is more efficient as the flow taken from the supply line is smaller.

The difference between the energy saving potential in fast, medium and slow trajectories is noticeable. The difference is most likely based on the leakage of the DHPMS. The leakage is bigger on higher loads, which is well visible in Table 6, which contains energies. There are more losses in the DHPMS in longer measurement times and higher loads.

What if the ideas were applied to a machine containing more than two actuators? How would the results change? With improved valve technology, the reduction of energy consumption is over 20 % in both common and independent supply pressures. From these pieces of information, the following hypotheses can be made:

- Independent metering reduces metering losses and thus, energy consumption.

    - Valve losses are related to the actuator, not the system, although they affect each other if unnecessary throttling is the reason for the level of the LS-pressure. Thus, as long as the load is the reason for the supply pressure, not throttling, the result should be generalizable.
    - The bigger the variation of load during the work cycle, the bigger the possible energy savings by independent metering.
- With independent supply pressures, there is a major potential for energy savings.
    - The amount of reduction depends greatly on the combination of loads on all of the actuators.
    - It is likely that more actuators would lead to bigger energy savings. Compensation losses occur in all valves except? the one controlling the highest load.
    - A higher number of actuators result in a higher amount of compensator throttling. Therefore, it is assumed that the bigger the number of actuators, the bigger the potential for energy savings with these methods.
- By applying both of the presented improvements, it seems that a virtually constant amount of energy, over 40 %, can be saved.
    - Although the potential savings differ depending on the amount or combination of loads, the total savings remain at a surprisingly constant level.
- Simulations show that utilization of pressurized tank line in to the studied system is beneficial: Recuperating modes are more often possible and even greater reduction of losses is likely achievable.

Of the utilized methods, independent metering is part of the newest serially produced systems. Also multi-pump systems are technically feasible, as are transformers. The presented system shows a digital hydraulic version of the holistically improved system and the future will show how well it will be adapted to solutions. Especially, the DHPMS requires a lot of design improvements, but taking into consideration what has been achieved in the area of digital valves during the last decade, finding feasible technological ways to create the next generation of DHPMS can be considered likely. Realization of the pressurized tank line has been studied and clear motivation for further development of that technology is shown.

The research conducted for this thesis gives insight that digitalization of hydraulic circuit is potential way for improved efficiency. But these benefits do not come without a prize. Compared to the state of the art system, the tested system requires highly complex control system and sophisticated signals processing in the software. This requires highly developed controller units, transducers and power electronics for feeding power to the digital control elements. For instance, a traditional pump with check valves or port plate do not require any controls, the digital pump requires actively controlled valves for each piston together with shaft angle measurement and very accurate & pressure dependent port valve timing. Analogue systems with machined functionality may not be as modular and online optimizable but hydraulically sensed signals and hydro-mechanical controls does not require transducers or electrical actuators.

Utilization of DVS with serial orifices, as is done in this thesis, requires major amount of measurement and tuning in order to achieve good resolution and accurate model. Applying miniaturization into the digital valve technology and using only similar size bits instead of e.g. binary coded series has advantages. DVS based on equally coded series requires simpler control code, is easier to implement, is more reliable, more fault tolerant, more reliable, smaller in physical size, requires less control power and is easier to manufacture in large scale. However, promising technology; a DVS package containing valves, block and valve drivers is yet found only from laboratory but going to be tested in situ. This technology is almost ready for large scale manufacturing.

A DHPMS requires considerable amount of engineering until it may reach market proof stage. Achilles heel of the machine are the port valves. They must be extremely precise and reliable but also be small with good flow rate and consume only a little energy. Otherwise, the pump structure is quite simple in case of inline pumps but only radial piston devices are possible. Axial piston DHPMS is trickier but with rotary swashplate design it could also be possible base structure. However, the research has shown benefits which should be motivation for a design process. Working control codes, which were considered almost impossible by some, have been achieved. Maybe same will happen also with hardware.

Results of this thesis are to be treated as pioneering work, giving insight into possibilities of digitalization but also reveal demands and requirements. Simulations and measurements match well which proofs that simulations are trustworthy method. This should encourage engineers to test the applied methods in their own virtual prototypes. When knowledge about the possibilities of digitalization in machine automation becomes more generally known and clear benefits of new solutions are found via simulation studies, urge for real solutions emerge and pilots in certain special applications shall be seen. Though, drastic amount of R&D are required on DHPMS and its miniaturization before market proof products can emerge. Concept has been proven, and next steps are to be taken.

9. CONCLUSIONS

In this thesis, two new technologies were united; a digital hydraulic power management system – a DHPMS and a digital valve system – a DVS. The research was performed by simulations and measurements. The combination was tested on a 2-dergree-of-freedom backhoe boom, standing on a stiff platform.

Holistic level system and control design was conducted by joining together the DVS controller with the DHPMS controller with proper augmentations. The multi-port feature of the DHPMS was applied to a two-actuator system, enabling independent supply pressures. Hence, necessity for matching losses was removed. The first of the research questions was, is the DHPMS capable for this kind of task. Shortly, it can be answered that yes, it is. The control system designed for the DHPMS provides the swift and accurate control of pressures in both of the supply lines, having different pressure levels, different flow rates and good performance is maintained also during recuperation. Naturally, recuperation is possible only in systems where the control valve and its control system enable that, which is the case with a DVS.

The second research question considered possibility to unleash the combinatory benefits of DHPMS and DVS. The online mode switching capability, which is a practical requirement for the effective utilization of recuperation, requires a highly dynamic four quadrant supply unit. The DHPMS has high dynamics, as the pump output depends on the sequence of the port valves. Therefore, a full amplitude response of a digital pump is always achieved within one revolution of the driving shaft, or in half revolution, if pistons are arranged as they are in a boxer-type machine.

The DVS controller is model based one, thus it provides estimates for various signals. Flow rate estimates were found out very useful for the DHPMS controller, especially in mode switching instants when step wise changes in reference signals are present and delays in control would cause jerky behavior.

The third research question asked about amount of possible energy savings. The conducted research shows, that together, independent supply pressures and independent metering resulted a 40% energy savings. On average 24 % energy saving was achieved because of the independent metering and the effective use of different modes. 21 % energy savings were obtained by removing matching losses by independent supply pressures. Loading case and trajectory affects in which fraction metering and matching losses are reduced, but net effect was virtually constant, 40%.

The ideal hydraulic power management system independently provides required pressure and flow for multiple actuators and is capable of handling energy flows to and from the consumers. The digital hydraulic power management system – a DHPMS is a practical realization of this "machine-of-dreams", and in order to unleash its full potential, digital valve system – a DVS is required. The potential of the studied topic is proved. It is expectable that new innovations in the field shall be seen.

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APPENDIX A – NON-LINEAR LOAD FORCE FILTER

```
function [F_out,F_high_,F_low_,rate__] = fcn(F,rate,increase_rate)
%Nonlinear Filter for Load Force filtering.
%Mikko Huova, Matti Karvonen (& Markku Luomaranta) 2013.
%Utilizable as embedded matlab function in Matlab/Simulink

%Declare persistent variables
persistent F_high
persistent F_low
persistent rate_
persistent dir_
%Initialization at first run
if isempty(F_high)
    F_high= 1;
    F_low= -1;
    rate_= rate;
    dir_= 1;
end
%Filtering code
dir=dir_;

if F>F_high
    dir=1;
elseif F<F_low
    dir=-1;
end
if dir==dir_
    rate_=rate_+increase_rate;
else
    rate_=rate;
end
if F>F_high
    F_high=F; %F_high directly follows F
else
    F_high=F_high-((F_high-F)*rate_); %F_high calculated by first
order filtering
end

if F<F_low
    F_low=F; %F_low directly follows F
else
    F_low=F_low+((F-F_low)*rate_);%F_low calculated by first order
filtering
end
F_out=(F_high+F_low)/2;
F_high_=F_high;
F_low_=F_low;
dir_=dir;
rate__=rate_;
end
```

APPENDIX B – VALVE PARAMETERS

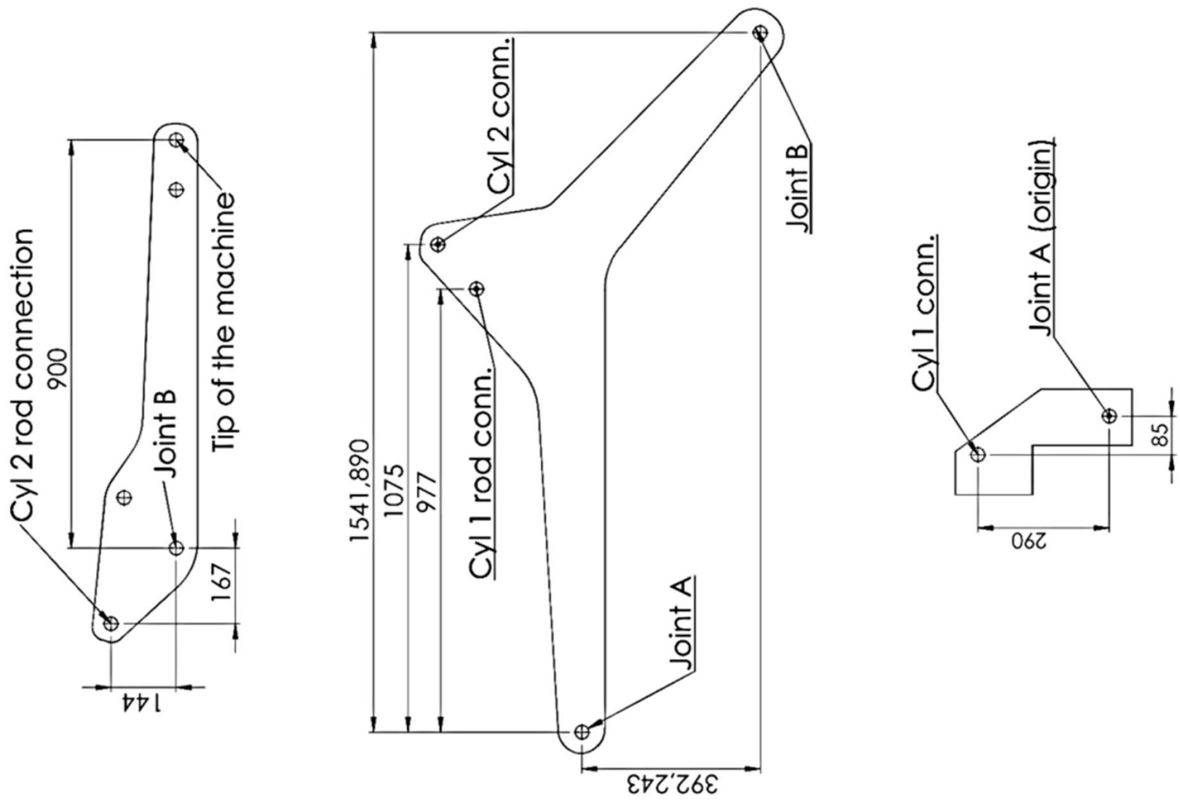
```
%% Lift
DFCU1_PA.Kv1 = [1.9582e-009, 4.2576e-009, 1.5314e-008*1.0, 2.7998e-
008, 9.3774e-008, 1.0131e-007];
DFCU1_PA.Kv2 = [1.8662e-009, 4.2350e-009, 2.4700e-008*1.1, 4.5561e-
008, 8.4288e-008, 1.0947e-007];
DFCU1_PA.x1 = [0.6200 0.6000 0.5700 0.5700 0.53 0.53 ];
DFCU1_PA.x2 = [0.6200 0.6000 0.5400 0.5400 0.54 0.53 ];
DFCU1_PA.b1 = [0.3 0.25 0.3 0.36 0.30 0.30];
DFCU1_PA.b2 = [0.07 0.25 0.29 0.25 0.15 0.05];

DFCU1_AT.Kv1 = [2.8549e-009, 3.0381e-009, 1.5408e-008, 3.8217e-008,
7.5471e-008, 7.8567e-008];
DFCU1_AT.Kv2 = DFCU1_AT.Kv1;
DFCU1_AT.x1 = [0.5900 0.6200 0.5700 0.5500 0.5500 0.5500];
DFCU1_AT.x2 = DFCU1_AT.x1;
DFCU1_AT.b1 = [0.10 0.24 0.28 0.28 0.25 0.2];
DFCU1_AT.b2 = DFCU1_AT.b1;
DFCU1_ATs.Kv1 = [4.8639e-009, 1.7815e-008, 6.4503e-008, 1.1761e-
007, 2.3719e-007, 2.1062e-007];
DFCU1_ATs.Kv2 = DFCU1_ATs.Kv1;
DFCU1_ATs.x1 = [0.5600 0.5100 0.4800 0.4800 0.4800 0.4900];
DFCU1_ATs.x2 = DFCU1_ATs.x1;

DFCU1_PB.Kv1 = [9.0658e-010, 4.3163e-009, 1.2856e-008*1.05,
3.8260e-008, 1.0898e-007, 1.1618e-007];
DFCU1_PB.Kv2 = [2.1818e-009, 4.8542e-009*1.1, 2.3909e-008*1.05,
4.6461e-008, 8.4707e-008, 9.1550e-008];
DFCU1_PB.x1 = [0.6700 0.6000 0.5800 0.5500 0.5200 0.5200];
DFCU1_PB.x2 = [0.6100 0.5900 0.5400 0.5400 0.5400 0.5400];
DFCU1_PB.b1 = [0.30 0.23 0.29 0.37 0.30 0.30];
DFCU1_PB.b2 = [0.09 0.20 0.29 0.29 0.20 0.16];

DFCU1_BT.Kv1 = [2.4307e-009, 4.7910e-009, 1.8098e-008, 3.3811e-008,
8.7415e-008, 9.0070e-008]*1.05;
DFCU1_BT.Kv2 = DFCU1_BT.Kv1;
DFCU1_BT.x1 = [0.6000 0.5900 0.5600 0.5600 0.5400 0.5400];
DFCU1_BT.x2 = DFCU1_BT.x1;
DFCU1_BT.b1 = [0.08 0.20 0.27 0.28 0.20 0.16];
DFCU1_BT.b2 = DFCU1_BT.b1;
DFCU1_BTs.Kv1 = [4.8238e-009, 2.0522e-008, 5.5695e-008, 1.2163e-
007, 1.9832e-007, 2.0000e-007];
DFCU1_BTs.Kv2 = DFCU1_BTs.Kv1;
DFCU1_BTs.x1 = [0.5600 0.5000 0.4900 0.4800 0.4900 0.4900];
DFCU1_BTs.x2 = DFCU1_BTs.x1;

%% Tilt
DFCU2_PA.Kv1 = [1.4535e-009, 5.7615e-009, 2.0689e-008*1.1, 3.7886e-
008*1.075, 9.6194e-008*1.075, 1.2104e-007];
```

```
DFCU2_PA.Kv2 = [2.1639e-009, 5.7783e-009, 2.4809e-008, 5.2960e-008,
7.4384e-008, 7.8096e-008];
DFCU2_PA.x1 = [0.6400 0.5800 0.5500 0.5500 0.5300 0.5200];
DFCU2_PA.x2 = [0.6100 0.5800 0.5400 0.5300 0.5500 0.5500];
DFCU2_PA.b1 = [0.3000 0.2500 0.3000 0.3800 0.2900 0.3000];
DFCU2_PA.b2 = [0.0800 0.2500 0.3000 0.3000 0.2400 0.1400];

DFCU2_AT.Kv1 = [2.4890e-009, 4.8933e-009, 2.4487e-008*1.0, 5.3089e-
008, 8.5398e-008*1.05, 9.0760e-008]*1.125;
DFCU2_AT.Kv2 = DFCU2_AT.Kv1;
DFCU2_AT.x1 = [0.6000, 0.5900, 0.5400, 0.5300, 0.5400, 0.5400];
DFCU2_AT.x2 = DFCU2_AT.x1;
DFCU2_AT.b1 = [0.08 0.22 0.29 0.35 0.14 0.16];
DFCU2_AT.b2 = DFCU2_AT.b1;
DFCU2_ATs.Kv1 = [5.3214e-009, 2.4400e-008, 6.5448e-008, 1.3800e-
007, 1.9279e-007, 2.0252e-007];
DFCU2_ATs.Kv2 = DFCU2_ATs.Kv1;
DFCU2_ATs.x1 = [0.5550 0.4900 0.4800 0.4700, 0.4900, 0.4900];
DFCU2_ATs.x2 = DFCU2_ATs.x1;

DFCU2_PB.Kv1 = [1.8984e-009, 5.7245e-009*1.1, 1.7654e-008*1.1,
4.5363e-008*1.05, 1.1328e-007, 1.4125e-007];
DFCU2_PB.Kv2 = [2.4433e-009, 4.9669e-009, 2.0777e-008*0.95,
4.5301e-008*0.9, 8.7403e-008*0.85, 9.3360e-008];
DFCU2_PB.x1 = [0.6200 0.5800 0.5600 0.5400 0.5200 0.5100];
DFCU2_PB.x2 = [0.6000 0.5900 0.5500 0.5400 0.5400 0.5400];
DFCU2_PB.b1 = [0.2900 0.2200 0.3000 0.3700 0.3200 0.3100];
DFCU2_PB.b2 = [0.0600 0.2400 0.2900 0.2900 0.2200 0.2100];

DFCU2_BT.Kv1 = [1.6073e-009, 5.8018e-009, 2.0866e-008*1.0, 4.8967e-
008*1.075, 7.3352e-008*1.1, 9.2027e-008];
DFCU2_BT.Kv2 = DFCU2_BT.Kv1.*[1,1,1.1,0.9,1,1];
DFCU2_BT.x1 = [0.6300 0.5800 0.5500 0.5350 0.5500 0.5400];
DFCU2_BT.x2 = DFCU2_BT.x1;
DFCU2_BT.b1 = [0.0500 0.2200 0.3000 0.2900 0.1600 0.1700];
DFCU2_BT.b2 = DFCU2_BT.b1;
DFCU2_BTs.Kv1 = [2.6844e-009, 2.0960e-008, 5.9226e-008, 1.4011e-
007, 2.6168e-007, 2.8111e-007]*1.1;
DFCU2_BTs.Kv2 = DFCU2_BTs.Kv1;
DFCU2_BTs.x1 = [0.6000 0.5000 0.4850 0.4700 0.4700 0.4700];
DFCU2_BTs.x2 = DFCU2_BTs.x1;
```

APPENDIX C – SIMULATION PARAMETERS

```
%     Cylinder parameters
Cyl.Dd = [63e-3,36e-3];
Cyl.DeadVols = [0.05e-3,0.05e-3];
Cyl.HoseVols = [0.05e-3,0.05e-3];
Cyl.Oil_B = 1000e6;
Cyl.Hose_B = [300e6 300e6];
Cyl_1.stroke = 1000e-3;
Cyl_1.x_0 = 600e-3; %Cylinder min length
Cyl_2.stroke = 700e-3;
Cyl_2.x_0 = 400e-3; %Cylinder min length
%Static friction force
Fric.Fs = 800;
%Coulombian force
Fric.Fc = 750;
%Viscous coefficient
Fric.b = 300;
%Velocity of minimum friction
Fric.vmin = 0.02;
%Tanh-steepness coefficient;
Fric.K = 4000;
%%    Digital valve Dynamic parameters
%Delays & movement time
Valve.delay_open = 6e-3;
Valve.delay_close = 10e-3;
Valve. movtime_open = 4e-3;
Valve. movtim_close = 5e-3;
%%    DHPMS Model parameters
%Dampening volumes
DPMT.Vol_1 = 5e-3;
DPMT.Vol_2 = 5e-3;
%Volumes of supply lines
DPMT.V_hose1 = 4*pi/4*(3/4*25.4e-3)^2;
DPMT.V_hose2 = 4*pi/4*(3/4*25.4e-3)^2;
%Effective bulk modulus
const.B_eff = 1000e6;
%Tank pressure
const.p_T = 1e6;
%Rotational speed of the prime mover
Pump.n = 1000; % [rpm]
% Number of pistons
piston.N = 6;
% Phase shift of pistons
piston.ph_rad = 2*pi / piston.N * (0:piston.N-1);% Phase shift of pistons
piston.S = 16e-3; % Stroke
piston.d = 20e-3; % Diameter
piston.A = (pi*piston.d^2)/4; %Area
piston.V_disp = piston.S*piston.A;
piston.V_0 = 20e-6; % Dead volume
```

```
%DHPMS valve parameters:
valve.nominal_pressure = 3e6;
valve.nominal_flow = 55/6e4;
valve.delay = 0.001;
valve.move_time = 0.0005;

%%    Mechanism:
%Body masses: 80 kg & 40 kg.
%Inertias calculated by equation of solid rod having diameter of
100 mm.
%In the following figure, units are in [mm]
```



APPENDIX D – MEASURED TRAJECTORIES

The figures in following pages present the trajectories of single measurements of the slowest trajectory. The differences between DVS and proportional valve controlled systems are visible. Also the differences between single supply pressure (1 pS) and independent supply pressures (*2 pS*) are shown. The measurement cases are mentioned in the captions. Reader may further explanation of signals is presented in text of chapters.

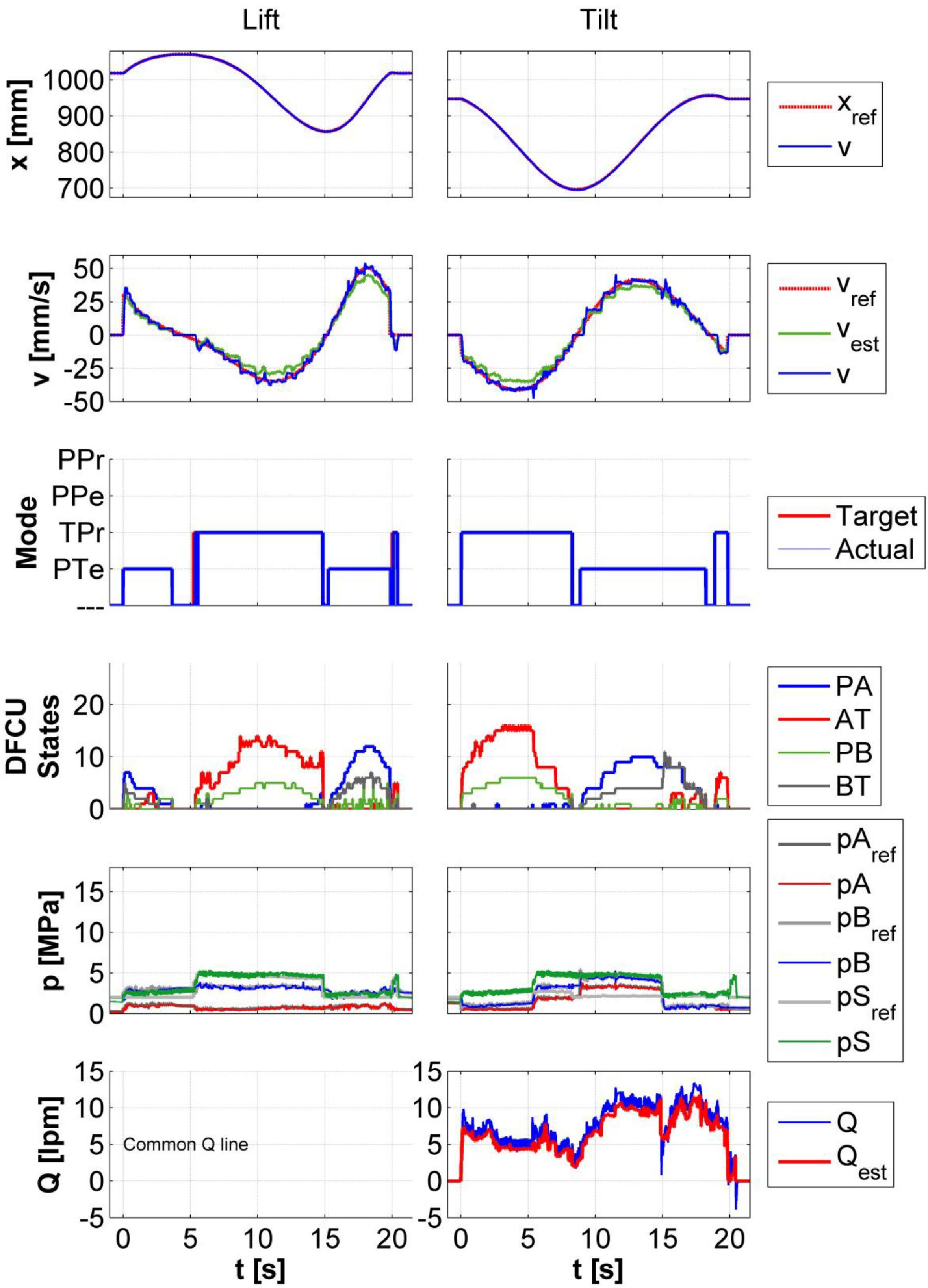


Figure 59. *DVS 0 kg, 1 pS*

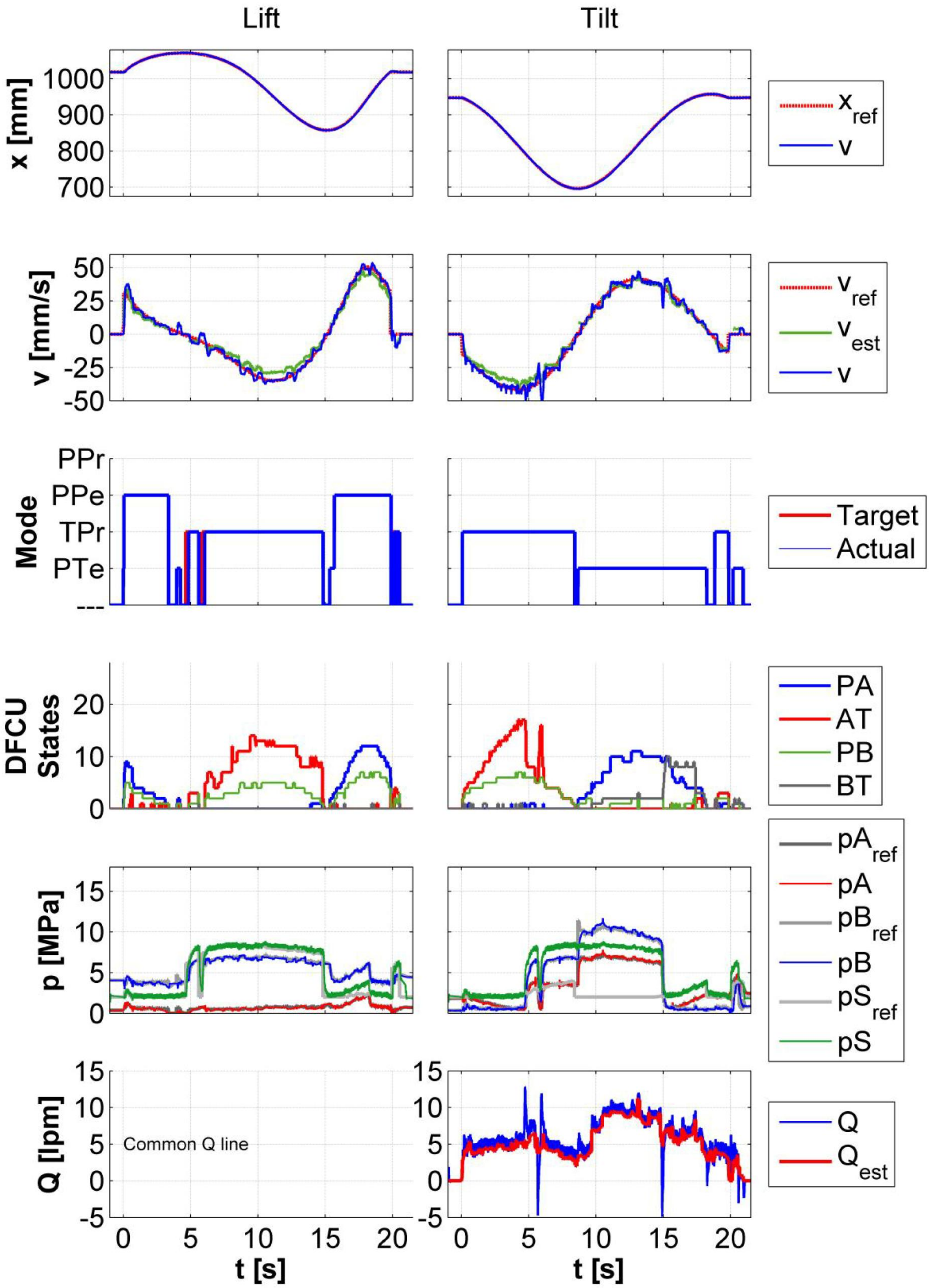


Figure 60. *DVS 100 kg, 1 pS*

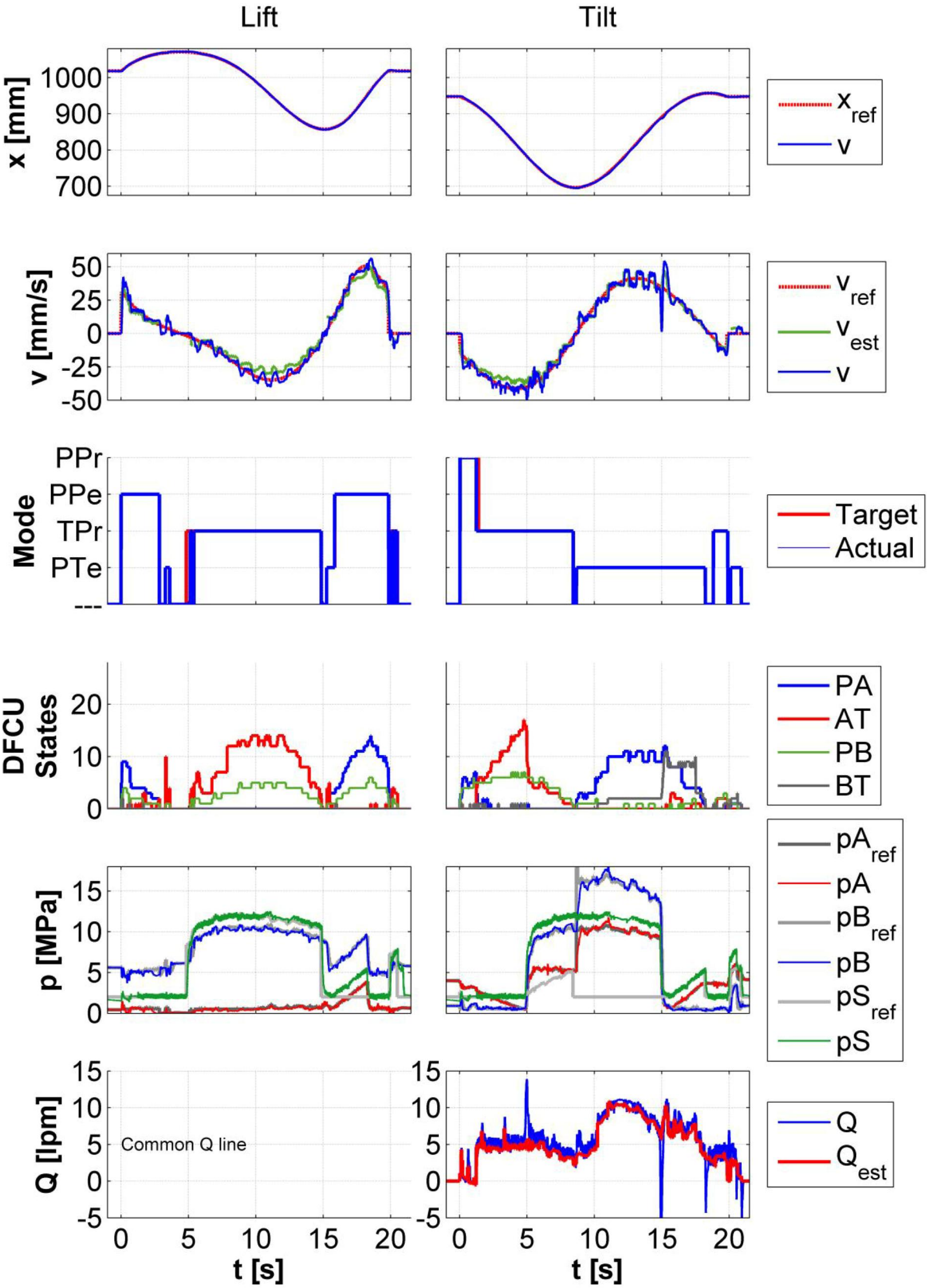


Figure 61. *DVS 200 kg, 1 pS*

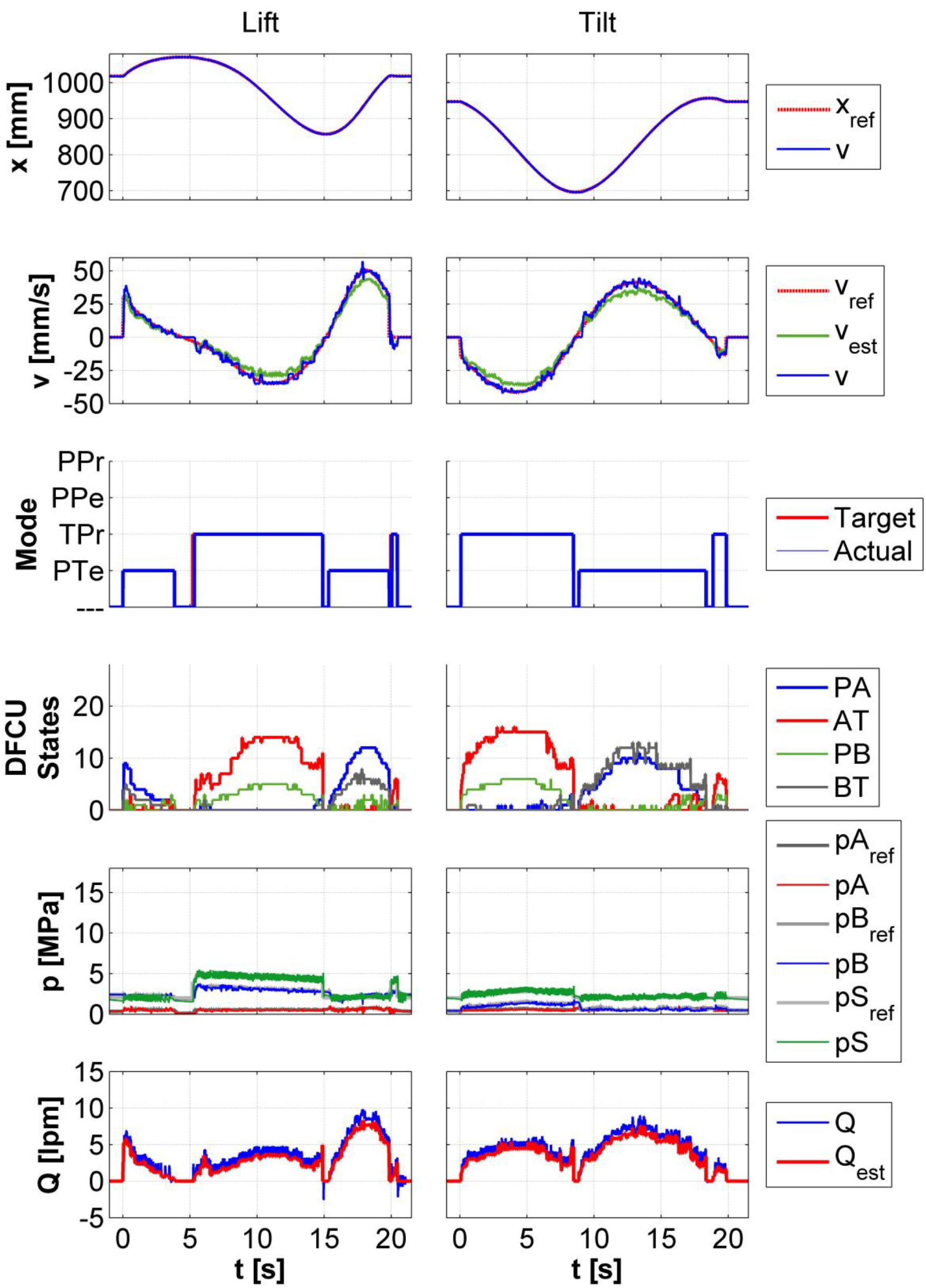


Figure 62. *DVS 0 kg, 2 pS*

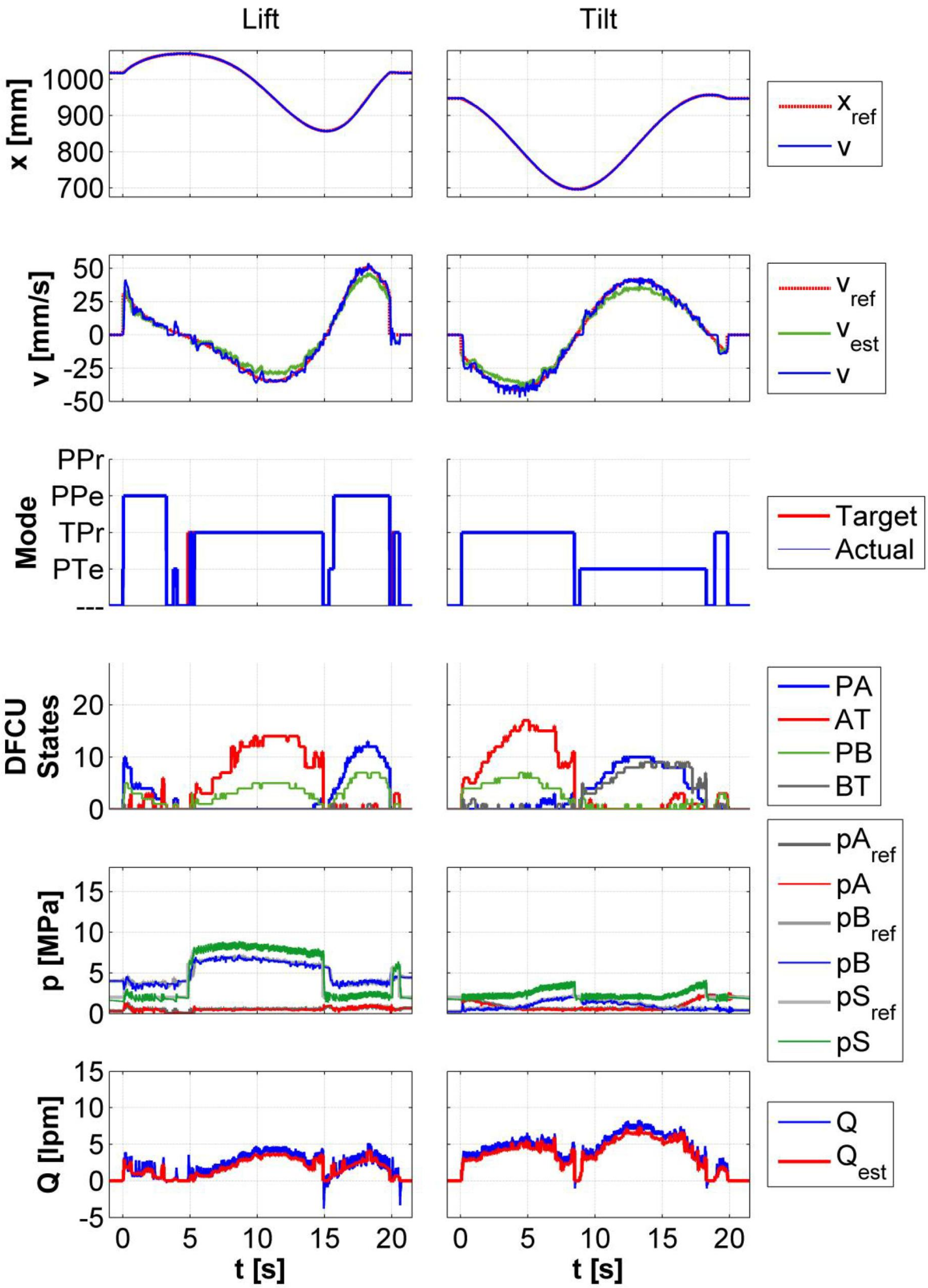


Figure 63. *DVS 100 kg, 2 pS*

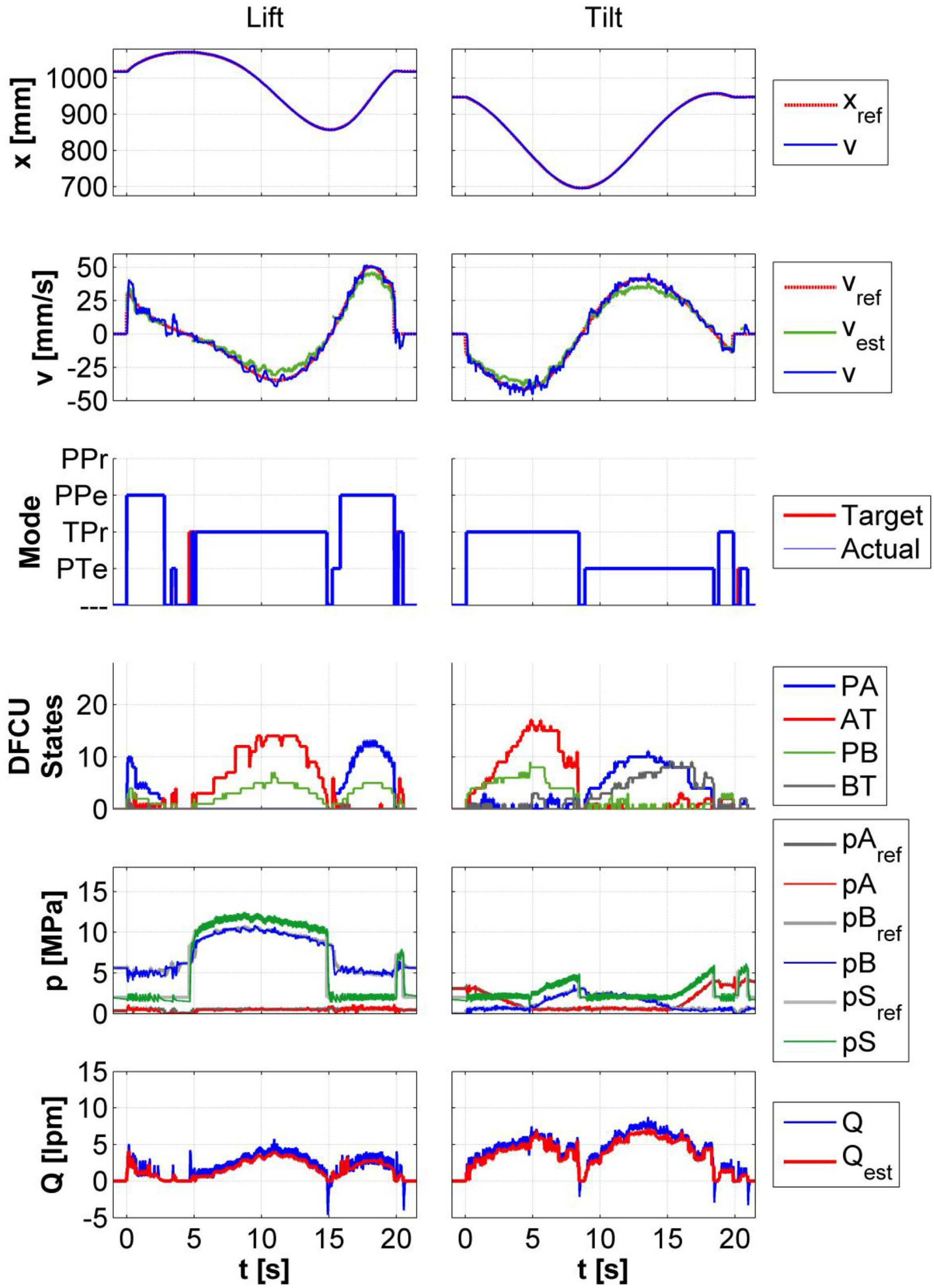


Figure 64. *DVS 200 kg, 2 pS*

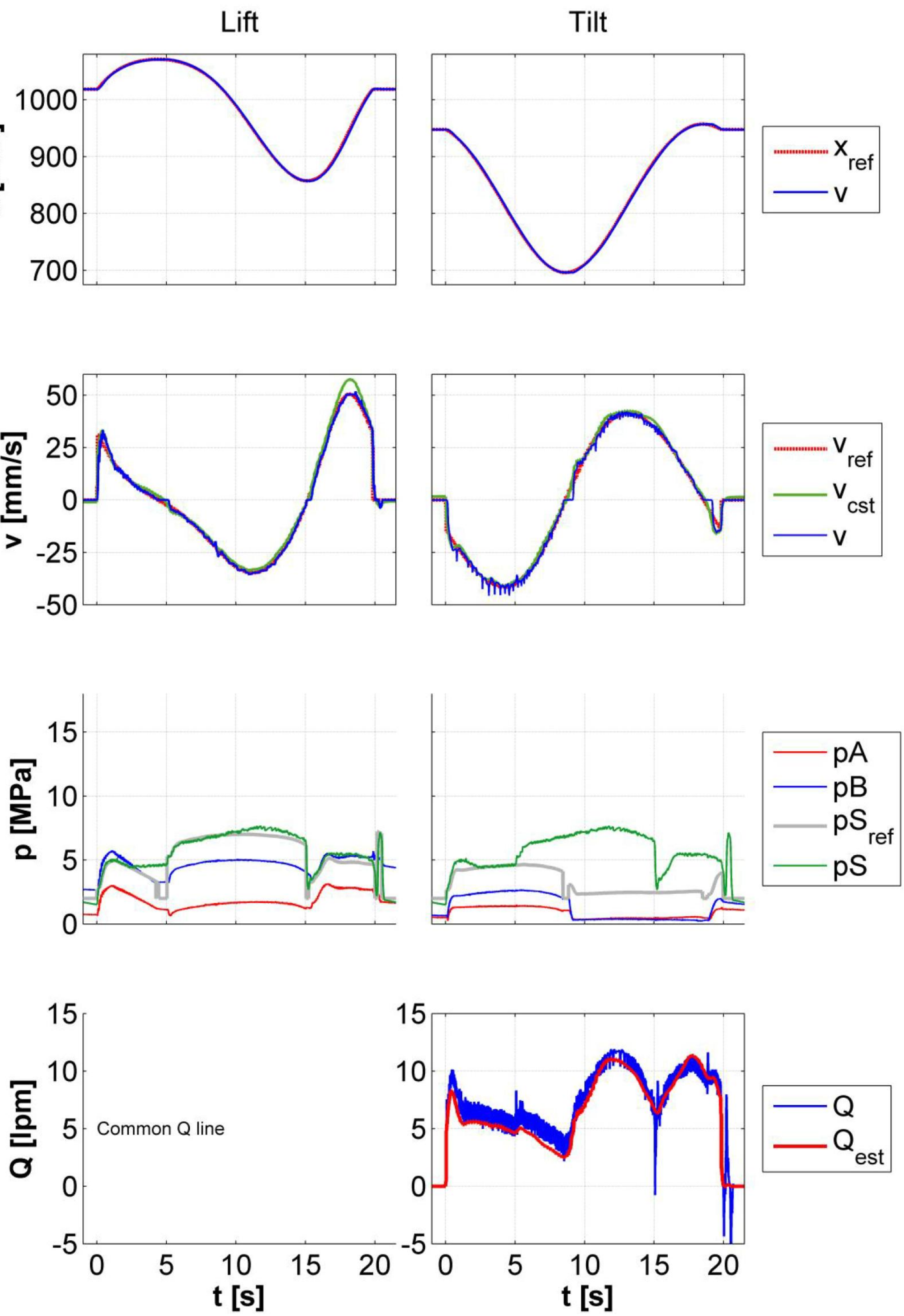


Figure 65. *Proportional 0 kg, 1 pS*

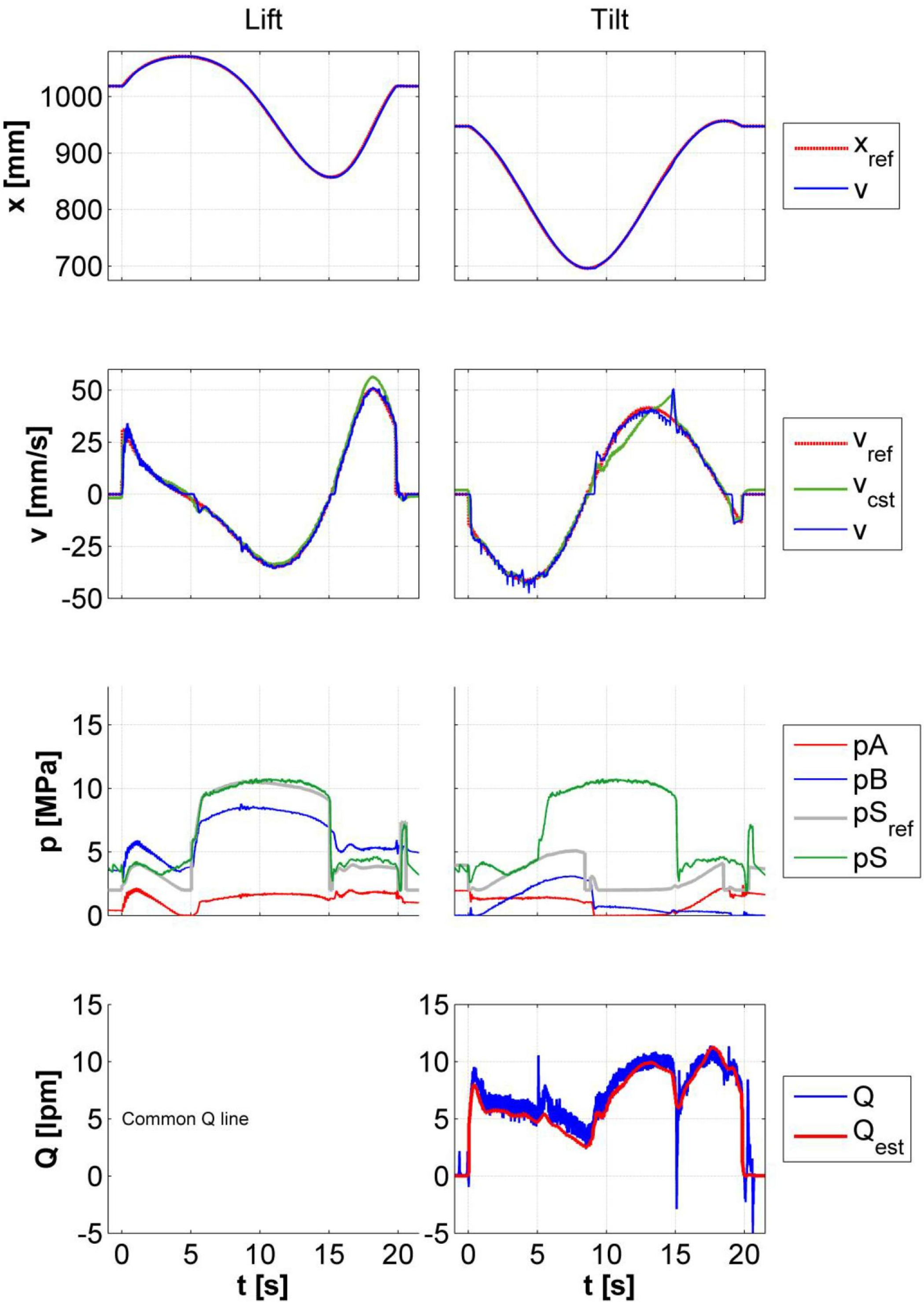


Figure 66. *Proportional 100 kg, 1 pS*

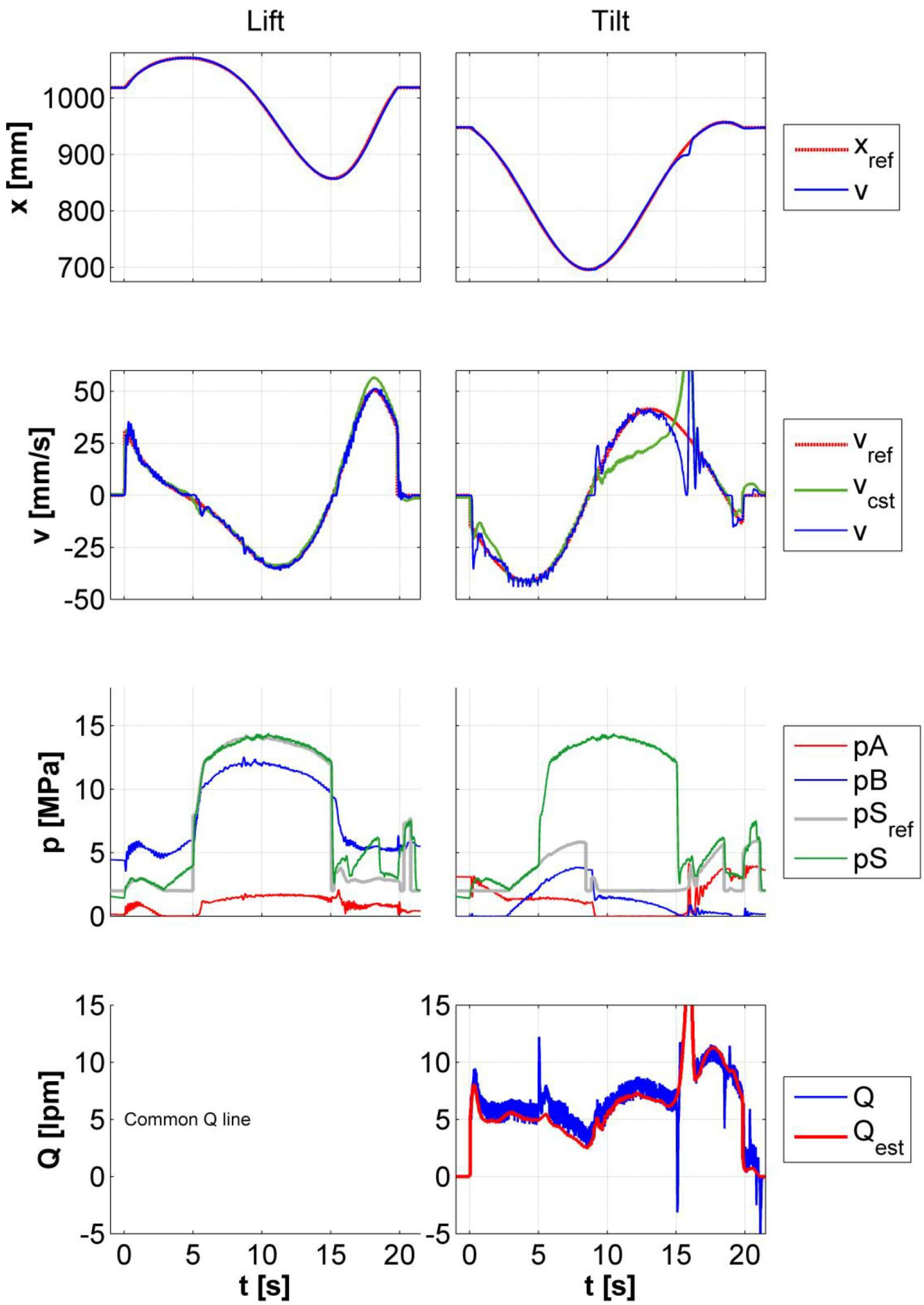


Figure 67. *Proportional 200 kg, 1 pS*

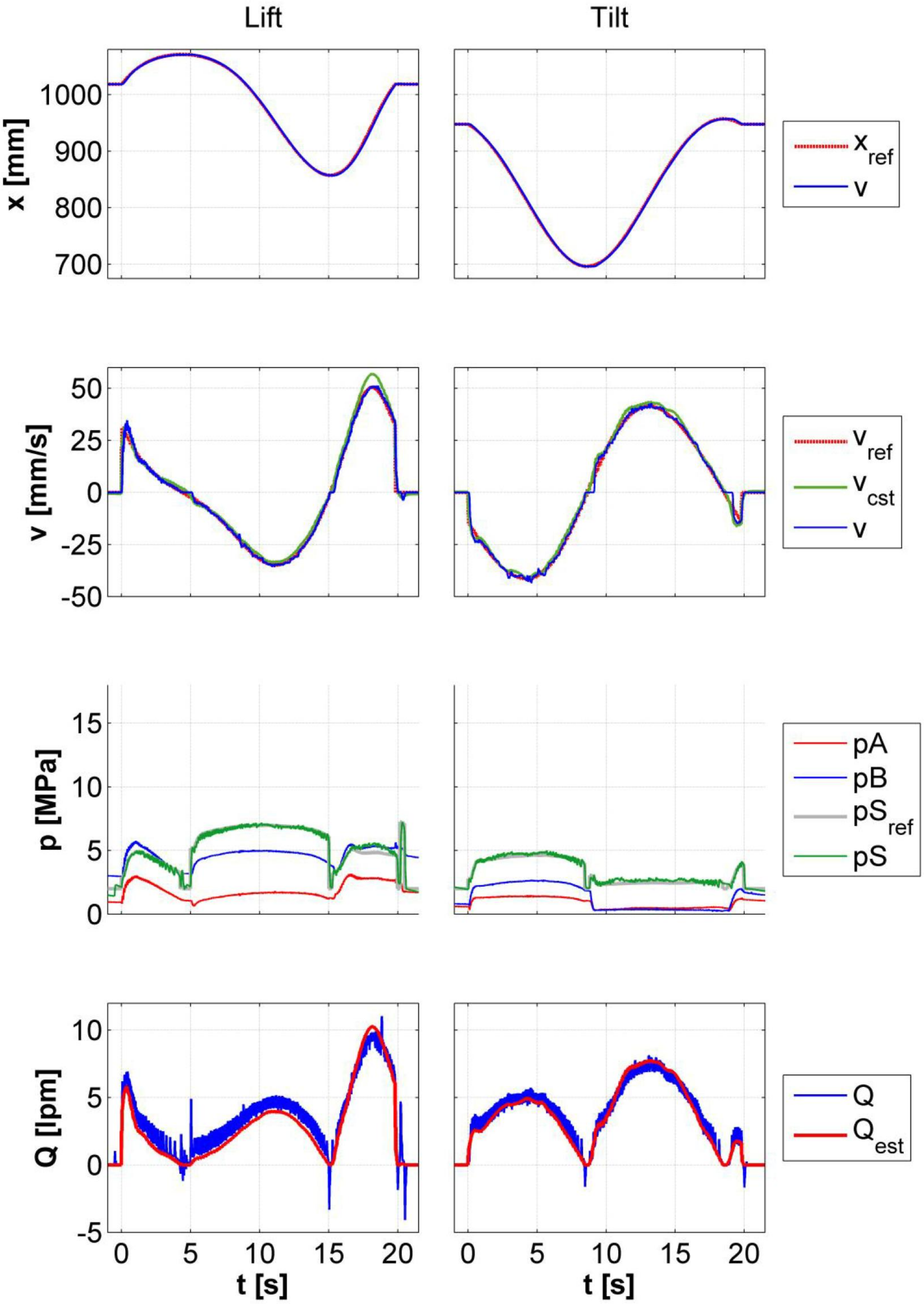


Figure 68. *Proportional 0 kg, 2 pS*

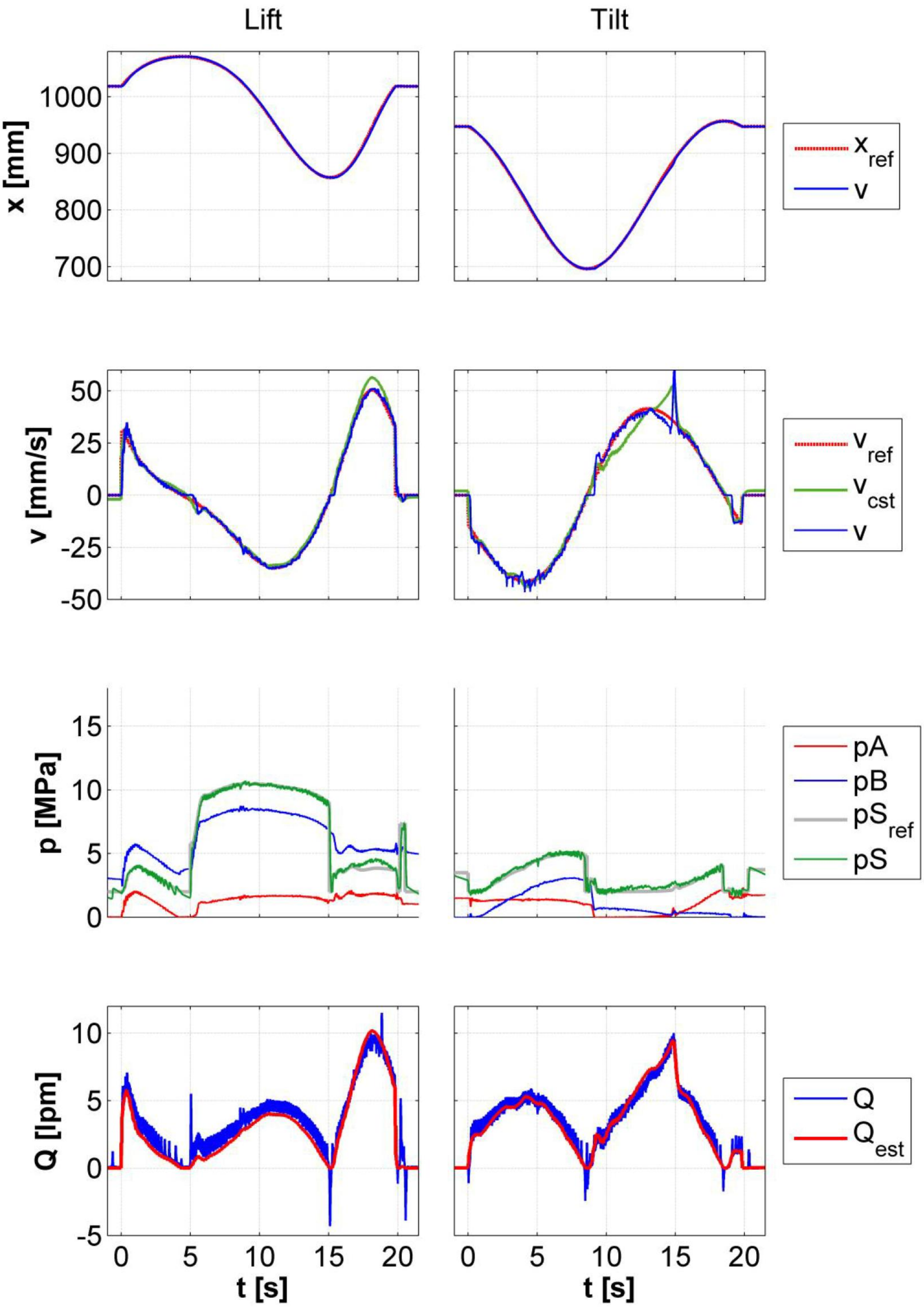


Figure 69. *Proportional 100 kg, 2 pS*

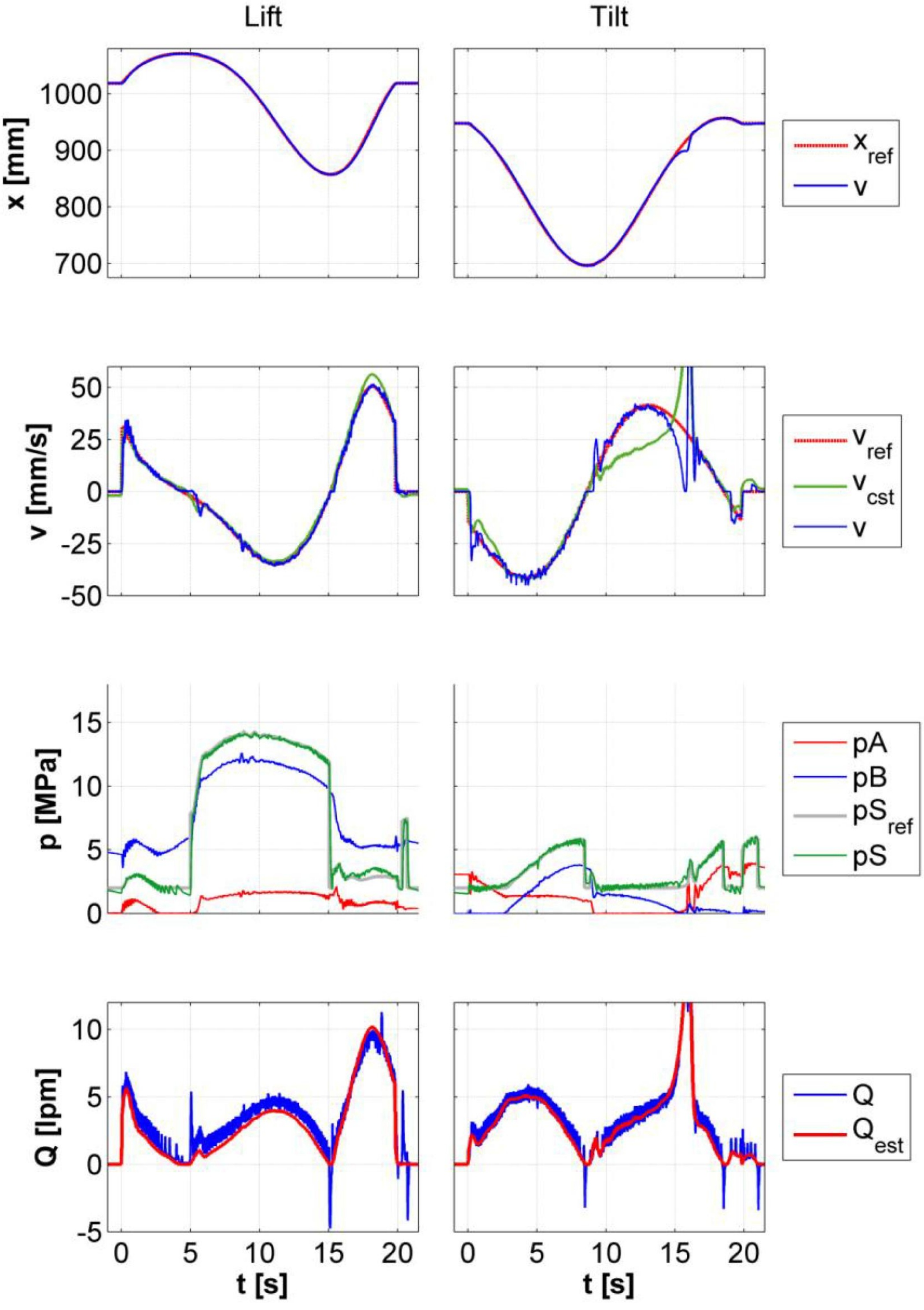


Figure 70. *Proportional 200 kg, 2 pS*

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