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Fully Variable Valve Actuation in Large Bore Diesel Engines

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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 3rd of October 2014, at 12 noon.

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FULLY VARIABLE VALVE ACTUATION IN LARGE BORE DIESEL ENGINES

Keywords: Hydraulic, Variable, Valve, Actuation, Flexible, Control, ILC

ABSTRACT

Diesel engine combustion process optimization has become increasingly important as environmental and economic issues are setting more strict conditions on engines. Best efficiency and lowest emission are not reached at the same time, and compromise between these is required. The more flexible the control of the combustion is, the more effective operation of the diesel engine is gained with required emission levels.

Variable gas exchange valve actuation is one effective method of adjusting the combustion process, and it has already been successfully used for years in passenger cars. Variable actuation can be implemented either by a mechanical, electric or electro-hydraulic device. All constructions have pros and cons, and it depends on the application which is best suited for the case in question. The large bore diesel is a very challenging application where masses and forces are high, and required movement distances long.

An electro-hydraulic actuation gives a benefit where almost full flexibility of the valve events is reached and full potential of the variable valve actuation can be used. Electro-hydraulic valve actuation is investigated in this study via simulations and measurements. The used hydraulic circuit and actuator construction has a strong effect on the performance of the valve actuation system. A 3-way controlled actuator gives the lowest energy consumption, and the control valve characteristic has a major role in overall performance. Right dimensioning of the gas exchange valve return spring is important. An energy consumption decrease of up to 20% could be achieved if the actuator was optimized.

Because the actuation system is not mechanically linked on the engine piston position and the dynamics of the valve actuation system are challenging, a reliable and accurate control system is needed. Pure P-control is not good enough, and a state controller is too complex to use when environment variables change. An iterative learning feature can adapt automatically in different working points and it can also execute good tracking error through the whole gas exchange valve lift.

PREFACE

This study was carried out at the Department of Intelligent Hydraulics and Automation (IHA) at Tampere University of Technology during the years 2005-2014.

I would like to express my deepest gratitude to my advisor, professor and head of the IHA Kalevi Huhtala, for his guidance, comments, advice and support over the years.

I am grateful to my colleagues and all other staff of IHA (laboratory and office) for their help, support, and for providing a pleasant working atmosphere. Special thanks to Otso and Tapio for their help with controller structures and building the simulation models.

One can never underestimate the effect of the people that you share a room with, the effect of the interesting and fruitful discussions over the years.

I wish to thank all involved people at Wärtsilä Finland Oy, who gave a lot of technical details and knowledge about diesel engine technology.

Thanks are also due to John Shepherd for revising the English of the manuscript.

Most of all, I would like to thank my wife, Taru, and my daughter Fiia, their love, support and understanding during this dissertation process. I would also like to thank my mother and sister for their encouragement and support over these years.

Akaa, 2014

Mika Herranen

CONTENTS

NOMENCLATURE

LATIN ALPHABETS

A_A	Actuator piston A chamber area	[m^2]
a_{act}	Acceleration of the actuator	[m/s^2]
A_B	Actuator piston B chamber area	[m^2]
A_{pipe}	Flow area of the pipe	[m^2]
A_{spool}	Spool orifice area	[m^2]
B_{eff}	Effective bulk modulus	[Pa]
D_{pipe}	Hydraulic diameter of pipe	[m]
e_c	Deformation of the body in full contact damping	[m]
E_{close}	Energy required to close gas exchange valve	[J]
$E_{hyd\ close}$	Hydraulic energy required to close the gas exchange valve	[J]
$E_{hyd\ open}$	Hydraulic energy required to open the gas exchange valve	[J]
$E_{hyd\ total}$	Hydraulic energy from hydraulic pump	[J]
E_{open}	Energy required to open gas exchange valve	[J]
$E_{rec\ close}$	Recoverable energy in gas exchange valve closing	[J]
$E_{rec\ open}$	Recoverable energy in gas exchange valve opening	[J]
F_A	Pressure force of the actuator upper chamber	[N]
F_{act}	Force provided by actuator	[N]
F_B	Pressure force of the actuator lower chamber	[N]
F_C	Contact force	[N]
F_{Fact}	Friction force of the actuator	[N]
F_{GEV}	Sum of the gas exchange valve pressure forces	[N]
F_k	Spring force	[N]

F_{pcyl}	Cylinder pressure force	[N]
F_{pEXH}	Exhaust manifold pressure force	[N]
F_{spre}	Pre-tension force of the return spring	[N]
i	Iteration index	[-]
KA	Acceleration gain	[-]
k_c	Contact stiffness	[N/m]
Kff	Feedforward gain	[-]
KP, Kp	P-control gain	[-]
Kp_{crit}	Critical P-control gain value	[-]
k_s	Spring constant	[N/m]
KV	Velocity gain	[-]
K_v	Control valve gain	[-]
K_{vPA}	Flow coefficient of the control valve	[$m^3/s\ \sqrt{Pa}$]
l_{pipe}	Length of pipe	[m]
m_{vt}	Valvetrain total moving mass	[kg]
p_a	Pressure in upper actuator chamber A	[Pa]
$p_{A\text{-}e}$	Actuator chamber A pressure at equilibrium state	[Pa]
p_b	Pressure in lower actuator chamber B	[Pa]
p_{pump}	Pressure in pump pressure line	[Pa]
Q	Hydraulic flow	[m^3/s]
q	Learning gain	[-]
Q_-	Flow out from actuator chamber	[m^3/s]
Q_+	Flow in to actuator chamber	[m^3/s]
Q_A	Hydraulic flow of the upper actuator chamber	[m^3/s]
Q_B	Hydraulic flow of the lower actuator chamber	[m^3/s]
Q_{pipe}	Hydraulic flow through pipe	[m^3/s]

Q_{pump}	Hydraulic flow from pump	[m^3/s]
Q_{valve}	Hydraulic flow through control valve	[m^3/s]
r_c	Maximum viscous friction of contact	[N/(m/s)]
Re	Reynolds number	[-]
rr	Relative roughness of pipe	[-]
r_{vt}	Viscous friction of one actuator and GEV pair	[N/(m/s)]
t	Time	[s]
t_{close}	Gas exchange valve event 'valve closed' time	[s]
t_{open}	Gas exchange valve event 'valve opened' time	[s]
u	Control input	[-]
V_{A0}	Dead volume of the actuator upper chamber	[m^3]
v_{act}	Velocity of the actuator	[m/s]
V_{B0}	Dead volume of the actuator lower chamber	[m^3]
v_c	Velocity of contact penetration	[m/s]
v_{pipe}	Flow velocity in the pipe	[m/s]
v_{ref}	Reference velocity of the actuator	[m/s]
x_c	Contact penetration displacement	[m]
x_{ref}	Reference displacement of the actuator	[m]
x_v	Control valve spool displacement	[m]
y	Vector of output variables	[-]
y_{act}	Displacement of the actuator	[m]
y_{act_e}	Displacement of the actuator at the equilibrium state	[m]
y_{stroke}	Actuator stroke length	[m]

GREEK ALPHABETS

Δp_{valve}	Pressure difference over valve	[Pa]
Δy	Tracking error	[m]
δ_{act}	Damping ratio of the actuator	[-]
δ_v	Damping ratio of the control valve	[-]
μ	Coefficient of discharge	[-]
ρ_{oil}	Density of hydraulic oil	[kg/m^3]
ω_{act}	Hydraulic undamped natural frequency of the actuator	[rad/s]
ω_v	Hydraulic undamped natural frequency of the control valve	[rad/s]
λ	Flow friction coefficient	[-]
υ_{oil}	Kinematic viscosity of the oil	[m^2/s]

ABBREVIATIONS

CETOP	Comité Européen des Transmissions Oléohydrauliques et Pneumatiques
CO	Carbon Monoxide
CO_2	Carbon Dioxide
D1FP	CETOP 3 Control valve
D3FP	CETOP 5 Control valve
EH	Electro-Hydraulic
EHVA	Electro-Hydraulic Valve Actuation
EM	Electro-Mechanical
EXH	Exhaust gas exchange valve
GEV	Gas Exchange Valve
HC	Hydrocarbons
IHA	Department of Intelligent Hydraulics and Automation

ILC	Iterative Learning Control
IMO	International Maritime Organization
INT	Intake gas exchange valve
LVDT	Linear Variable Differential Transformer
MARPOL	The International Convention for the Prevention of Pollution from Ships
MBC	Model Based Controller
MBDC	Model Based Digi Controller
NOx	Nitrogen Oxides
PID	Proportional - integral - derivative controller
PM	Particle matter (smoke/particulates) of combustion emissions
RPM	Rotation Per Minute
SOx	Sulphur Oxides
VVT	Variable Valve Train
W20	Wärtsilä 20 size engine

1 INTRODUCTION

Environmental, 'green' values have lately been paid increasing attention to in many technological researches. Especially carbon dioxide emissions of industry, alternative energy sources and renewable fuels have been studied for years. This has had an effect on the emissions restrictions of all combustion engines. Furthermore, this has led to the increasing need of engine developers to find and research new and better technologies to fulfill these increasing demands. Better controllability of the combustion engine is vital to produce power efficiently and with lower emission levels.

There are 46,340 ocean going ships with an average main engine size of 5.6 MW and three auxiliaries of 750 kW each. These engines run 300 days per year while the auxiliaries are operated 365 days per year. In the United States, 4% of petroleum liquids go into marine transportation, which means 2,900 tons of fuel a day. The U.S. uses about 25% of the world's total petroleum consumption. Based on this statistic, marine consumption of petroleum liquids is approximately 12,000 tons/day or 4,320,000 tons/year [May 2009]. Figure 1 shows another statistic, indicating the increase of maritime transport CO_2 emissions since the mid-1980s.

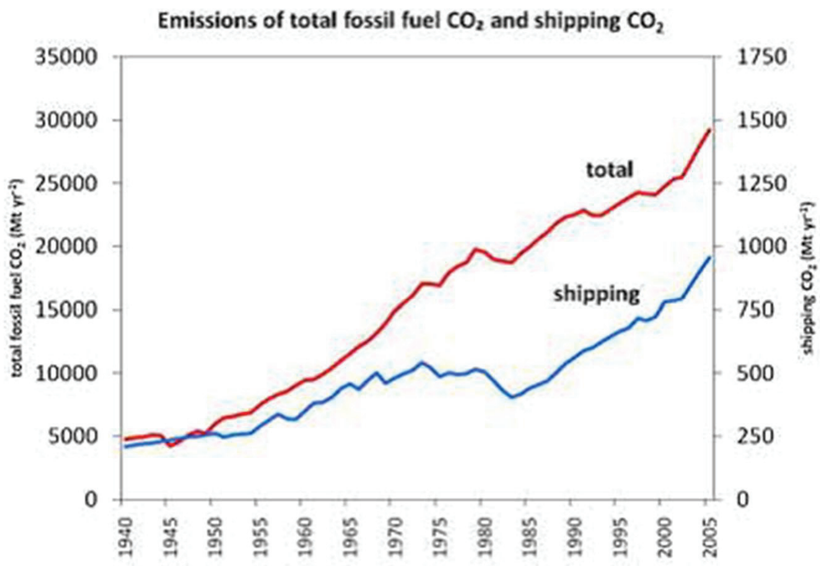


Figure 1 Historical development of CO_2 emissions from maritime transport [Lee 2009]

The major pollutants in diesel exhaust emissions are a direct result of the diesel combustion process itself. The diesel engine pollutants and their source can be summarized as follows [DeMers 1999]:

Sulphur Oxides (SOx)	- Function of fuel oil sulphur content
Carbon Dioxide (CO_2)	- Function of combustion
Carbon Monoxide (CO)	- Function of the air excess ratio and combustion temperature and air/fuel mixture
Hydrocarbons (HC)	- Very engine dependent but a function of the amount of fuel and lubrication oil left unburned during combustion
Particle matter (PM)	- Originates from unburned fuel, ash content in fuel and lubrication oil
Nitrogen Oxides (NOx)	- Function of peak combustion temperatures, oxygen content and residence time

Some of the pollutants, like SOx, are only a variable of fuel ingredients, and thus cannot be decreased by any external devices. The majority of recent research has been focused on the reduction of NOx emissions. The International Maritime Organization's (IMO) ship pollution rules are contained in the "International Convention on the Prevention of Pollution from Ships", known as MARPOL 73/78. Figure 2 shows the NOx emission limits depending on the maximum engine operating speed. Tier I was implemented in 2000 and Tier II in 2011. Tier III is to be implemented in 2016 and applies only to NOx emission control areas.

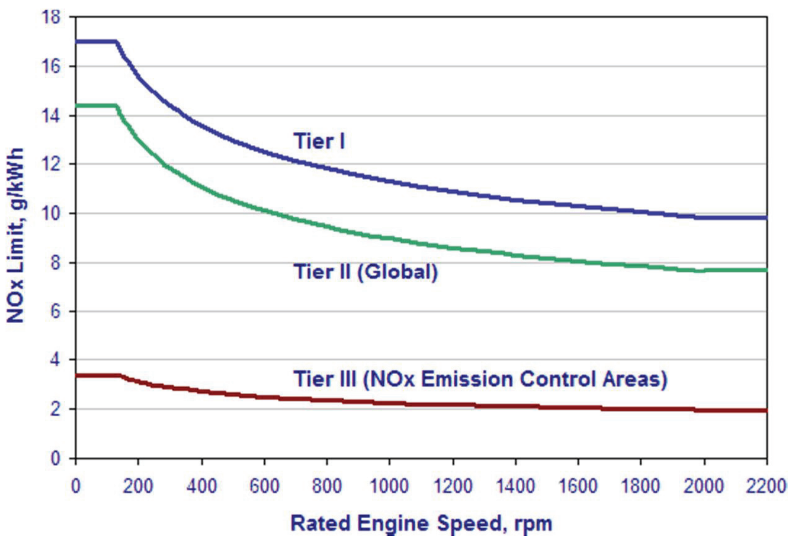


Figure 2 MARPOL Annex VI NOx Emission Limits [Anon. 2013a]

NOx reduction technologies can be divided into three basic categories: pretreatment, internal measures and after-treatment (Figure 3).

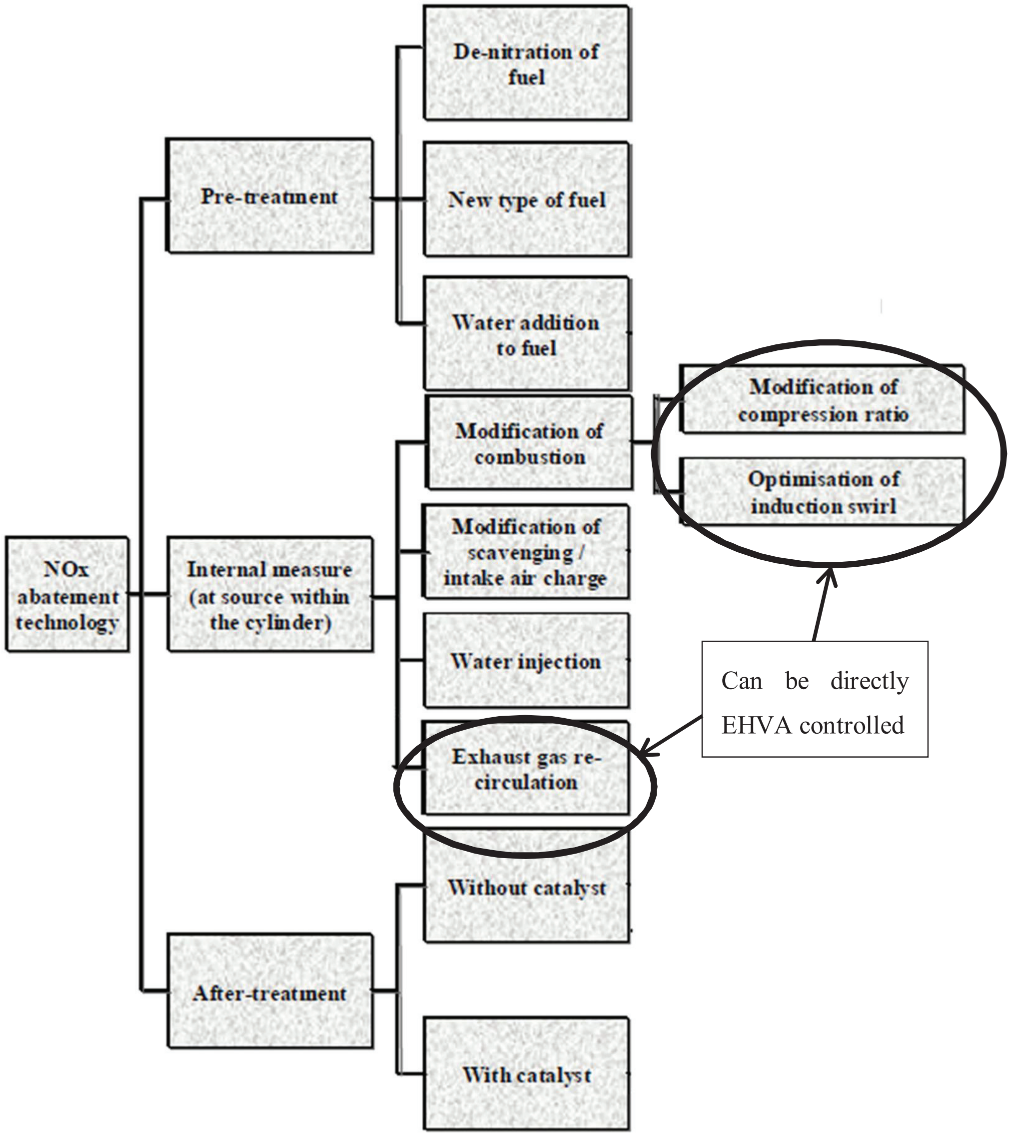


Figure 3 Different ways to improve the emissions of the combustion engine, edited [DeMers 1999]

Pretreatment and after-treatment methods are not relevant to gas exchange valve actuation. Primary (internal) methods involve changes to the combustion process within the engine and fall under four main categories, two of which can be influenced with variable valve actuation: modification of combustion and exhaust gas re-circulation (if done internally by changing GEV timing, see Figure 4).

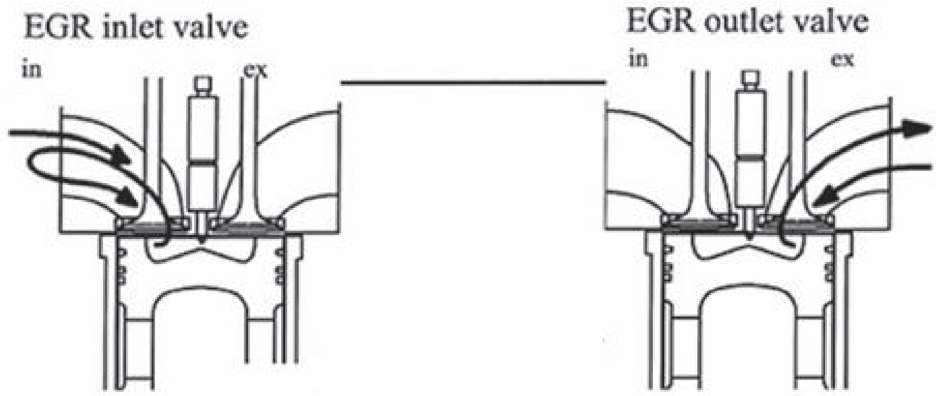


Figure 4 Controlled Internal EGR [Anon. 2013c]

In modification of combustion, a couple of listed methods apply further to the VVA: modification of the compression ratio and optimization of the induction swirl. By fully flexible valve actuation, the induction swirl can be increased and changed by uneven opening of the multiple intake valves. The compression ratio of the engine can be changed by timing the valve events.

In some cases VVA is combined with other technology areas within combustion modification methods, like two-stage turbocharging etc., and it is clear that multiple techniques are required to reach the final goal (Figure 5). However, each emission improvement method has a trade-off with decreasing NOx on other emissions such as CO, HC, PM (Figure 6).

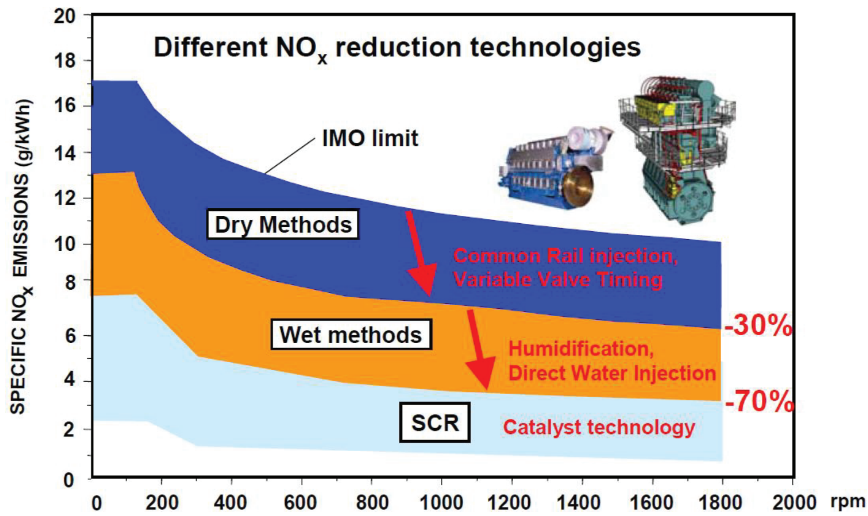


Figure 5 Sketched effect of emission technologies [Heim 2006]

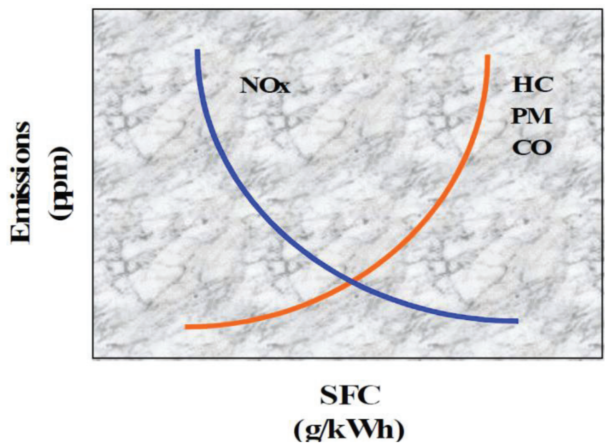


Figure 6 Trade-off of emission reductions [DeMers 1999]

2 RESEARCH PROBLEM

The combustion engine can be optimized to one, specific working point. There the air flow, fuel mixture, combustion, scavenging etc. could be designed as close to perfect as possible. Unfortunately, like said earlier, the lowest emission point and the point of the best efficiency is usually not the same. Optimal working points are changed when environment variables like engine load, rotation speed are changed. This means that the engine must adapt to the large range of variables in order to run efficiently and in an environmentally friendly way. Performance control of the engine can also be done in several ways, for example by changing the fuel mixture, amount and timing, or changing the air intake or scavenging. Changing the lift and timing/phase of the gas exchange valves is one efficient method to gain a better engine performance range. Different valve events are described in Figure 7. In the conventional, camshaft mechanism (Figure 8) all gas exchange valve events are fixed to the crank rotation and cam profile, and this linking needs to be dynamically changed in the mechanical VVA system if variable events are wanted. In camless solutions the valves are actuated with a power source other than camshaft, and thus movement is not restricted.

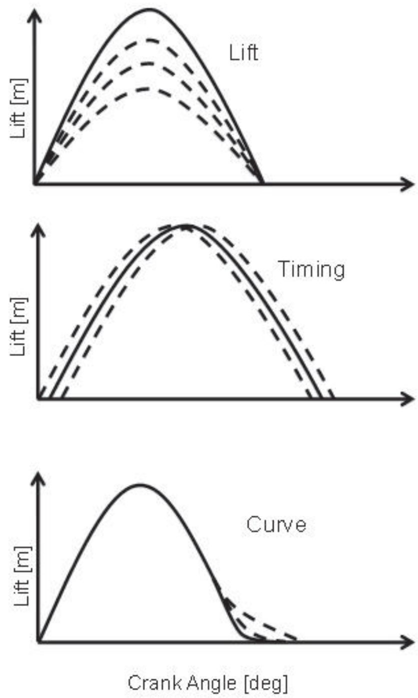


Figure 7 Different variable valve events

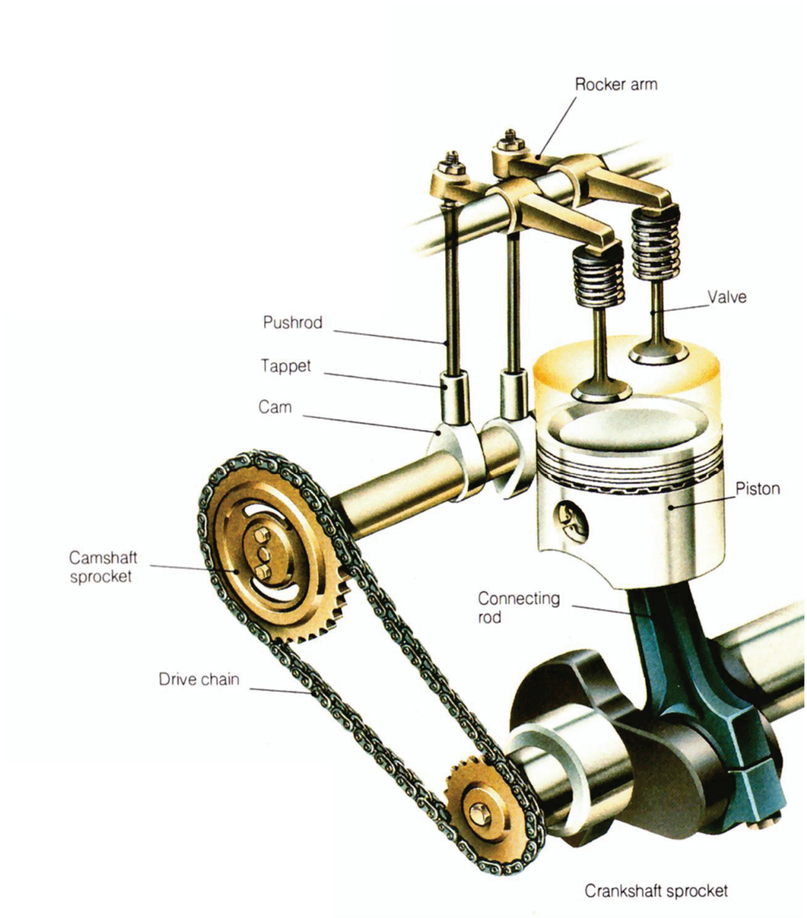


Figure 8 Conventional valve actuation mechanism

Table 1 lists the major performance characteristics of different conventional valvetrain systems, as well as Electro-Mechanical (EM) and Electro-Hydraulic (EH) camless systems. “1” represents the worst ranking and “5” represents the best.

Table 1 Performance characteristics of valvetrains [Wang 2007]

	Valvetrain Type									
	Direct Acting	**End-Pivot Rocker Arm**		**Center-Pivot Rocker Arm**					**Camless**	
Parameter	I	II (Flat)	II (Roller)	III (Flat)	III (Roller)	IV	V (Flat)	V (Roller)	EM	EH
VT Friction/Wear	4	1	5	3	5	2	3	5	3	3
VT Effective Weight	4	5	5	1	1	3	2	2	5	5
VT Stiffness	5	4	4	3	3	2	1	1	5	5
Valve Lift	2	4	4	5	5	3	1	1	5	5
Valve Acceleration	5	4	4	3	3	2	1	1	5	5
Valve Event	4	3	3	5	5	5	4	4	5	5
Valve Side Load	5	1	3	3	4	3	3	4	5	5
System Cost	3	3	3	2	2	4	5	5	1	1
Maximum Speed	5	4	4	3	3	2	1	1	1	1
Cam/Follower Wear	4	2	5	3	5	3	3	5	NA	NA
Packaging	2	3	3	4	4	4	5	5	1	1
VVA Capability	3	2	2	1	1	4	4	4	5	5

Notes:

(a) Valve event: Type I is limited only by the diameter of the follower. Types II and III are limited by cam lobe concavity. Types IV and V are limited by the diameter of the follower and valvetrain stiffness.

(b) Friction: Type I has the lowest friction potential without any roller element. The roller follower provides low friction potential.

(c) Event control: Types II and III have timing variation sensitivity due to tolerance stackup. All types have improved control when using hydraulic lash adjusters.

(d) Roller followers have a beneficial effect on the valve event and friction.

(e) There is a distinct design tradeoff between the speed capability and the valve event for all types. Type I would have the maximum design freedom due to its stiffness.

NA = Not applicable.

Implementation of variable valve timing (VVT) is challenging for the mechanical system. Either the mechanism will be very complex, or the range of the operation, for example timing, is limited. In addition some physical restrictions like surface pressures of sliding surfaces remain in mechanical constructions.

While full flexibility of gas exchange valve events is required, any fixed linkages with mechanical (rotational) movements are not wanted. This leads to a solution where the gas exchange valve event is fully controlled by electric control signals. This also gives an opportunity to use stroke by stroke control, where sequential strokes can have totally different lift and event timing. In addition, mixed 2-stroke events can be used in between 4-stroke cycles, which is not possible with a cam aided system.

With some large bore diesels, also the running frequency (RPM) causes challenges to the actuator system, because the lift of the Gas Exchange Valve (GEV) is still relatively long but the required stroke time is short (typically about 20mm lift under 15 ms). Moving masses are relatively high, too, up to a few kilograms, so inertia of the masses is significant.

The space/power ratio of the actuator is always meaningful. In a medium or large bore diesel engine, the forces of GEV actuation are relatively high, up to 15 kN. This leads to a situation where for example fully pneumatic actuator systems are not suitable for GEV actuating due to the too large size of the required actuator.

Thus the hydraulically actuated GEV is a natural choice for the actuator device. The hydraulic flow is controlled by the solenoid valve or valves. The same kind of technology is already used in common rail injection systems. One important challenge is energy consumption, which is often stated to be high in hydraulic systems when comparing to mechanical systems. The problem of GEV actuation is that the required power is highest at the opening of the exhaust valve, but decreases rapidly towards to end of the GEV lift. A mechanical system takes power from the diesel engine only when required, while hydraulic systems have a tendency to use more power if not controlled properly. On the other hand a hydraulic system has for example less bearing, power transmission losses etc. Some comparisons between different mechanical variable valve methods and camless (EH, EM) methods are listed in Table 2.

Table 2 Comparison of VVA mechanisms [Wang 2007]

Function	Fuel Economy	Emission	Power	Torque	Simplicity	Cost
Cam Phasing: Two-Step	+	++	++	++	-	- -
Continuous	+	++	++	++	- -	- -
Variable Valve L	+	0	++	++	- - -	- -
Cam Profile Switching	+	+	+++	+++	- -	- -
Variable Valve Lift with Cam Phasing	++	+++	+++	+++	- - -	-
Continuously Variable Lift with Cam Phasing	+++	+++	+++	+++	- - - -	- - - -
Lost Motion or Deactivation	++	0	++	++	- - -	- -
Electrohydraulic Valve Actuation	++	++++	++	++	- - - -	- - - -
Electromagnetic Valve Actuation	++	++++	++	++	- - - -	- - - -

Notes: + Plus or beneficial
- Minus or detrimental

2.1 *Aim of the Thesis*

The goal of this thesis is to present a novel solution for flexible electrohydraulic valve actuation (EHVA) of the large bore four-stroke diesel engine. It focuses on a hydraulic system, including a hydraulic circuit, suitable hydraulic control valve and competent control principle.

The timing and lift of the gas exchange valve event should be fully controllable in the studied system. Thus a solution for the system was sought where both gas exchange valves are opened and closed without any operational mechanical constraints, whenever it is rationally possible. GEV operation limits of the studied system according to the given opening lengths, accuracy, response times, and controllability of the system (Table 3) are examined. The opening curve of the GEV is freely definable during any part of the GEV lift. The characteristic and the difference of each studied actuation system is found. The control system's capability to improve functionality is estimated. The energy consumption of the EHVA is studied and guidelines for better economy are given.

2.2 *Restrictions*

Only fluid power solutions are investigated in this study. In addition, only systems which are freely actuated with constant pressurized fluid are considered. This means that so called 'lost motion' systems, where the gas exchange valve main motion is created with the help of a rotating element like a camshaft and hydraulics is used only to decrease the lift or timing of the gas exchange valve event (Figure 9), are excluded.

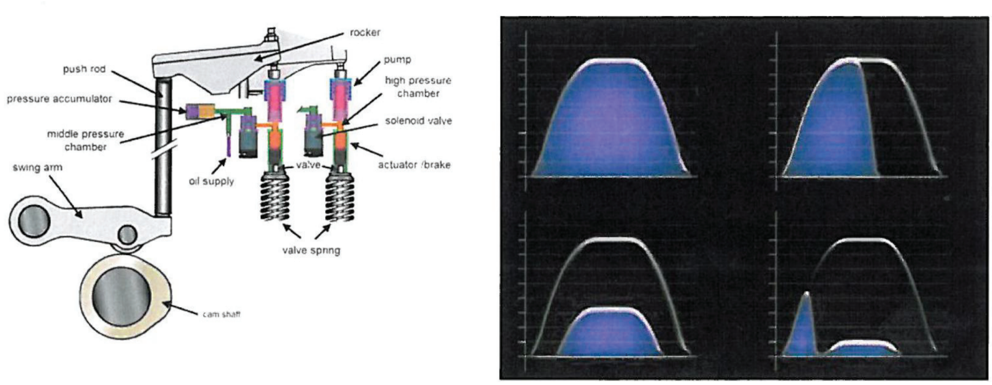


Figure 9 VVA Lost motion method [Mathey 2010]

The main restrictions of fully variable actuation are physical restrictions. The combustion piston of the diesel engine goes so close to the cylinder head that if the GEV is fully open, the paths of both moving elements intersect. Cylinder pressure is present in the moment of exhaust valve opening. The value of the pressure at the earliest wanted opening moment determines the force needed from the EHVA actuator. Once this is fixed, opening against higher combustion cylinder pressure is not possible. The maximum actuator force also determines the minimum opening time due to the maximum possible acceleration. The bandwidth of the hydraulic system and control devices (control/actuation) restricts the rate of reaction and action. Gas exchange valve seating velocity is limited due to mechanical impact forces. Contact loss between the valvetrain components is possible if decelerations are too high (due to inertia). The available space over the engine where the actuation device can be installed is limited. High ambient temperature and temperature gradients must be taken into account when choosing materials and electronic components. Also vibrations of the diesel engine disturb the actuation and cause stress to the all components. In some cases the distance required by control electronic installation is crucial, and onboard electronics is required. The optimum future combustion process demands are not yet fully known and may cause some changes to the initial data.

2.3 Research Methods

The research was carried out with simulations. The simulation models were verified with measurements, each model separately, and some results are based on measurement only. Also measurement data from a real diesel engine were available in some cases.

Because many different methods are applicable to hydraulic valve actuation of diesel engines, this study compares the different structures and control systems of EHVA, their characteristics and behavior.

2.4 Contributions of the Thesis

The contributions of the study can be summarized as follows:

An electro-hydraulic actuator device for gas exchange valve operation of the large bore diesel engine is developed.

Demands and implementations of the hydraulic system are mapped out. Proper components capable of realizing system requirements are found.

Mathematical models and a testing environment for simulating real engine are developed.

Different controller algorithms are tested and implemented, and after comparison the Iterative Learning Controller has been found to be the most effective and functional controller.

2.5 Structure of the Thesis

The thesis consists of 10 chapters and is organized as follows:

Chapter 2 introduces the needed characteristics of cam mechanisms in VVT, and compares VVA methods briefly.

Chapter 3 presents the state of the art of fully variable electro-hydraulic valvetrains.

Chapter 4 introduces technical requirements for the EHVA system, mechanical design and effect of the hydraulic parameters.

Chapter 5 provides modeling and verification of EHVA simulation models.

Chapter 6 introduces basic characteristics of the chosen EHVA system.

Chapter 7 presents different control methods used during the controller evolution process.

Chapter 8 provides results and comparison of controller performances, and presents research of the EHVA system's energy consumption.

Chapter 9 discusses EHVA tests on a real diesel engine and offers some suggestions for future research.

Chapter 10 concludes the thesis.

3.1 Development of the variable valve timing

The history of the hydraulic variable valve timing development dates back to the 1880s when it was utilized in the 'gas motor engine' [Clerk 1880]. Valve timing in internal combustion engines was started to be studied intensively after the 1920s, and several thousand related patents have been granted. Variable valve actuation has been researched around the world. Trends in the numbers of VVA related publications in different regions of the world are shown in Figure 10. It can be seen that the trend is overall increasing, but especially in North America and Japan the number of publication has doubled during the last decade. Also China is increasing activity in this area.

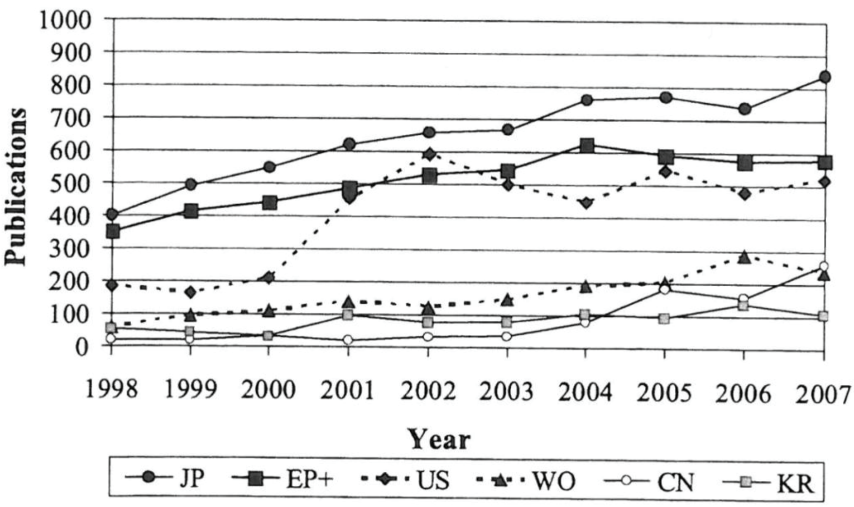


Figure 10 VVA publication activity around the world [Boye 2009]

3.2 Latest known researches, commercial systems

In this chapter, active companies and research institutes where EHVA systems have been studied in the 21st century are briefly presented.

Stanford University, CA, USA

The electro-hydraulic valve actuator was studied in Stanford University already in the beginning of the 1980s [Richman 1984], when one of the first publications of this kind of device was produced, and a test rig was manufactured. In 1989, Dresner and Barkan [Dresner 1989] published "a review and classification of variable valve timing mechanisms" where they introduced 15 different VVA concepts. In 2008, Liao et al. published "Repetitive Control of an Electro-Hydraulic Engine Valve Actuation" based on simulation in which they presented a framework covering system identification, feedback controller design and a feedforward repetitive controller [Liao 2008]. The electro-hydraulic valve actuation project includes close collaboration between researchers at Stanford University/Stanford Mechanical Engineering, the Robert Bosch Corporation (Robert Bosch Corporation Research and Technology Center) and the General Motors Corporation (General Motors Research and Development Center). As a result of this cooperation, several papers have been published (3 journal publications and 15 conference proceedings) and one dissertation, "Physics-Based Modeling and Control of Residual-Affected HCCI Engines using Variable Valve Actuation" by Gregory M. Shaver.

<u>University of South Carolina, USA (Mechanical Engineering Department)</u>

Increased efficiency, reduced emissions, and improved power over existing internal combustion engines are the three primary objectives of the research currently underway at the University of South Carolina. The research focuses on the development of a camless engine and addresses several of the design limitations of earlier camless attempts. One of their first publications dealing with camless engines is from 2001: John Brader's thesis "Development of a Piezoelectric Controlled Hydraulic Actuator for a Camless Engine". The main simulation based results have been introduced also in two journal papers [Brader 2004a, Brader 2004b].

<u>Purdue University</u>

John Lumkes is the assistant Professor of Agricultural and Biological Engineering at Purdue University. The main research areas at Purdue are controls, electrohydraulic and design of mobility systems. They also develop controller algorithms, electronics, and actuator systems for machines, camless engines, and hybrid vehicles. In several papers, they have studied different control strategies, e.g. adaptive control by simulations, for example [Hanks 2005], [Lumkes 2005].

<u>University of California, Los Angeles, USA</u>

The University of California has been studying electrohydraulic camless valvetrains since the 1990s and has established cooperation with the Ford Motor Company. In 2000, they introduced different control methods for the electrohydraulic valvetrain by means of simulations [Ashhab2000a], [Ashhab 2000b], and test rig tests were performed by Tai et al. [Tai 2000].

Indian Institute of Technology Madras

In 2006, the Institute made a patent application on Variable Valve Timing Assembly for a 4-stroke Internal Combustion Engine (970/MUM/2006) with Bajaj Auto Ltd. On the same topic, Raghav and Ramesh made simulation studies [Raghav 2007] of the new hydraulic variable valve actuation system.

Waseda University, Tokyo

Waseda University in Japan has research themes dealing with fuel consumption and exhaust gas emissions. They are using model based control for passenger car diesel engines to improve fuel consumption and exhaust gas emissions simultaneously. One of the key components in their experimental 1-cyl test set-up is electrohydraulic VVA [Murata 2007].

Zhejiang University, Hangzhou

The State Key Laboratory of Fluid Power Transmission and Control at Zhejiang University in China has published several papers dealing with VVT. According to some journal papers, they are developing an EHVVT system together with Ningbo HOYEA Machinery Co. Their VVT system is based on a three-way proportional valve, and also test rig tests have been done [Liu 2009]. Liu J.-R. has made a dissertation called "Key Technologies for Engine Variable Valve Actuator System Based on High-Speed Electro-hydraulic Valve". In recent studies they have improved the system by employing a peak and hold method [Yingjun 2010].

JiLin University, Changchun

Several publications about the electro-hydraulic valvetrain have been released by JiLin University, and the latest papers show that a test has been performed in a 1-cyl test engine and VVA mechanism and oil circuits have been optimized by CATIA modelling [Liu 2010], [Gu 2010].

Linköping University

J. Pohl is an Adjunct Professor in the Department of Management and Engineering. Pohl et al. presented during a short period several simulation based publications about the electro-hydraulic valvetrain where a fast switching on/off valve was used as a control valve [Pohl 2001a], [Pohl 2001b]. A publication investigating valvetrain power consumption [Pohl 2002] was then funded by the Volvo Car Corporation, but since then the subject has been dropped.

Technische Universität Braunschweig

Their VVA system provided different possibilities for cylinder-individual control of the valve timings and thereby new methods of exhaust gas recirculation and charge motion. Gehrke et al. have introduced development and implementation of a VVA system to a 1-cyl diesel engine [Gehrke 2008], though variable lift was not used. They have also submitted a patent application for “Gas Exchange Valve for Internal Combustion Engines”.

University of Minnesota

The Department of Mechanical Engineering at the University of Minnesota includes the Center for Diesel Research. Zongxuan Sun is one of the assistant professors and his recent research work includes novel time-varying control methodologies for rotational-angle based systems, developing key enabling technologies for clean, efficient and multi-fuel capable automotive powertrain, such as camless engine, precise pressure regulation for common-rail systems, modeling and control of efficient transmission systems, and various advanced hybrid concepts. In 2009, he published a journal paper on electrohydraulic fully flexible valve actuation. The paper presents an electrohydraulic valve actuation concept with an internal feedback mechanism. Key technical issues, such as dynamic range capability, seating velocity and closing timing repeatability, area schedule, internal feedback spool dynamics, and energy consumption, are modeled and analyzed. A test rig was also manufactured. [Sun 2009], [Heinzen 2011].

Tampere University of Technology / IHA

Valve actuation research began in Tampere in 2002, when Aaltonen et al. presented an EHVA system for medium speed diesel engines. The system was first a servo valve system [Aaltonen 2002], and was later changed to a proportional valve controlled system [Herranen 2007]. This research carried out with Helsinki University of Technology produced one of the very first (if not the first) running large bore single cylinder test engines with an EHVA system. Later the test engine has been in active research use, and many different phenomena have been investigated with the help of the EHVA system.

The aim of the developed EHVA system was to be able to follow the changing valve lift curves as precisely as possible [Herranen 2009, Herranen 2010b], and thus give the engine researchers an investigation tool for valve lift events and profiles. The proposed EHVA system has also been tested in a full scale 4-cylinder diesel engine in the Wärtsilä Finland laboratory. Also the energy consumption of the EHVA system has been investigated [Herranen 2010a, Herranen 2010c].

Lotus Engineering

Lotus has studied variable valvetrains [e.g. Allen 2002] during the last decade, and now they are offering the commercially available *Lotus' Active Valve Train (AVT) research system.* It is an electronically controlled, hydraulically operated system that provides control of individual valve lift profiles by a digital signal processor based controller. This enables faster research into advanced combustion performance and fuel economy. The AVT system is used on single cylinder engines in the research department of vehicle manufacturers and universities around the world.

International Truck and Engine

William de Ojeda and Jorge Fernandez introduced in 2003 a hydraulic needle valve actuator which could be used in camless engines [Ojeda 2003]. They defined that the hydraulic bandwidth of the actuator is fast enough to accommodate a valvetrain profile and the power consumption is comparable to its mechanical counterpart. The system was tested in a test rig and on a 6-cyl engine.

General Motors Corporation

The General Motors Corporation has intensively studied variable valve actuations during the last years. Sun and Cleary studied the dynamics and control of an electro-hydraulic fully flexible valve actuation system in 2003 [Sun 2003]. In 2007, Sun presented the development of a laboratory electro-hydraulic fully flexible valve actuation system for a single cylinder diesel [Sun 2007].

Recently, Chen has actively studied different electrohydraulic fully flexible valve actuation systems [Chen 2010]. An iterative learning control [Wu 2012] has later been utilized, and results show that the actuator tracking error is minor and valve motion is repeatable.

Sturman Industries

Sturman Industries has long experience of fast-acting on/off valves. For example, a digital magnetic latching valve is a low-voltage and highly accurate valve which was used in Apollo space flights. Later, their valves have been used also in variable valvetrains and in diesel injector technology. In 2004, Turner, Babbitt, Balton, Raimao & Giordano published a paper dealing with the design and control of a two-stage electro-hydraulic valve actuation system

[Turner 2004]. This system allowed fine control of seating velocities and the ability to respond to viscosity changes in working fluids. Sturman's hydraulic valve actuation system has been implemented in several test engines from passenger cars to heavy duty trucks, and some demonstration vehicles. Also a module for research purposes is available.

Robert Bosch GmbH

Robert Bosch GmbH is the world's largest supplier of automobile components, and has business relationships with virtually every automobile company in the world. However, they have published only a few papers related to electro-hydraulic variable valve actuation. Dirk Denger (AVL List GmbH) and Karsten Mischker (Robert Bosch) evaluated an electro-hydraulic valvetrain system (EHVS) in 2005 [Denger 2005]. The system was tested also on a 4-cyl test engine. The first results showed reduction of fuel consumption as well as emissions. The Robert Bosch Corporation has established a lot of cooperation with universities. Together with Stanford University they have studied for example the control of HCCI engines with VVT by simulations [Shaver 2005]. The focus has been on discovering the potential of VVT.

LGD Technology, LLC

LGD Technology develops and commercializes Variable Valve Actuation (VVA) technologies, especially camless technologies, for improved fuel economy, reduced emissions, and enhanced engine performance. One of the studied approaches is VVT design with two-spring pendulum and electrohydraulic latching [Lou 2007]. According to simulations, the system has lower electrical demand and better lift controllability.

Magneti Marelli Powertrain / University of Perugia

Magneti Marelli S.p.A. is an Italian company which deals with the development and manufacturing of systems, modules and high-technology components for the automotive industry. They are working on a VVT in collaboration with the University of Perugia. In 2007, Batistini et al. introduced a new VVA system, which has a low power demand and less complex design. Results of the experimental tests are published in [Postrioti 20008] and a parametric optimization study in [Battistoni 2008]. Recently, they have re-designed a hydraulic valve actuator, and tested it with a 1-cyl test engine. Its performance in terms of valve lift profile, repeatability and soft landing capabilities are reported in [Postrioti 2009].

MBtech Powertrain Group

MBtech powertrain solutions has studied a fully variable valve actuation system, but only a few publications can be found from the literature. Peter Dittrich, head of MBtech powertrain solutions, has presented the paper “Thermodynamic Potentials of a Fully Variable Valve Actuation System for Passenger-Car Diesel Engines” [Dittrich 2010]. The main goal has been to find a VVA strategy that could match the standard combustion process with respect to emissions and fuel consumption with the potential to replace all components relating to external EGR. The issue has been studied in the Lotus AVT system and a 1-cyl test engine. Results demonstrate that fully variable valve control (VVA) enables reducing nitrogen oxides, fuel consumption and costs in compression-ignition engines.

<u>Linz Center of Mechatronics GmbH</u>

Linz Center of Mechatronics concentrates on wide scale of mechatronics research technologies, and they have done research about the energy efficiency of the electrohydraulically actuated valvetrains. Their idea is based on hydraulic spring and mass oscillation system, with energy saving system and fast hydraulic piloting valve which is also development of their own [Plöckinger 2003]. Simulations and test rig measurements are carried out, and 60% energy consumption reduction is found out [Plöckinger 2004].

<u>Wärtsilä Corporation, MAN Diesel&Turbo</u>

Large bore two-stroke class engines have two commercial hydraulic actuated exhaust valve systems; RT-Flex manufactured by Wärtsilä Corporation is shown in Figure 11 and MAN Diesel&Turbo (ME Intelligent engine) in Figure 12. In low speed two-stroke engines, rotation speeds are usually under 200 RPM, but on the other hand due to only one exhaust valve, moving masses are high (up to 3000kg).

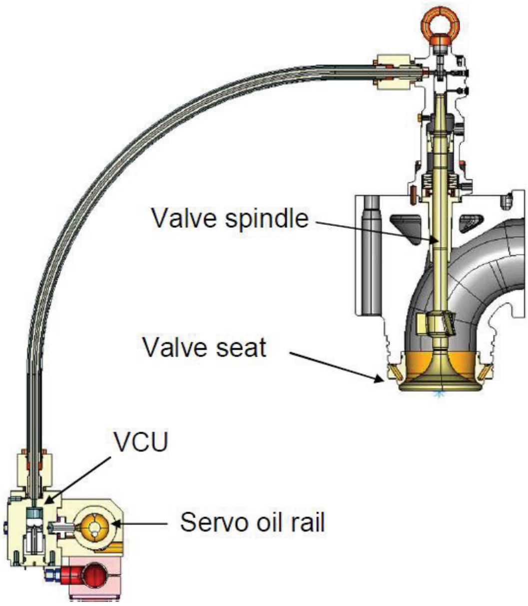


Figure 11 RT-Flex [Boletis 2010]

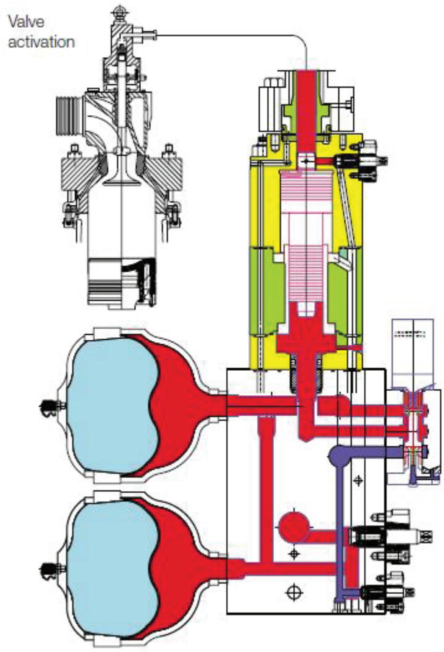


Figure 12 ME Intelligent engine [Anon 2013b]

3.3 Conclusions

Research into the Electro-Hydraulic Valvetrain has increased in the 21st century. Development of valve and sensor technologies has improved the performance of hydraulic systems. Research carried out by both academic and company based research groups around the world indicates encouragingly that electrohydraulic systems do have the potential to fulfill all the requirements of fully flexible valvetrains. Very accurate curve tracking or promising total energy consumption capability have been achieved in laboratory tests. Many research groups have built at least a 1-cyl test engine for experimental tests, which increases the validity of the results. Commercially available research VVA systems are produced by Sturman Industries and Lotus Engineering.

Different mechanisms and solutions have been developed and investigated in earlier research. Variable valve systems have often been designed to produce conventional or pure sinusoidal valve lift motion, instead of optimized valve lift. This is due to a lack of knowledge on what the optimum valve lift curve would be, because this information has not been available from any sources. Recently, the research methodology has slightly changed. In some of the latest publications, a trend can be seen where variable valve movement is taken for granted, and research is concentrated purely in other areas like NOx production, HCCI mixture control, or total controllability of the engine. This could lead to better usage of variable valve actuation, as a wider working range and performance of these VVA systems are known and other combustion technologies are developed. Eddie Sturman, co-founder of Sturman Industries, has said that *"The problem in the acceptance of his concept is that engine designers think conventionally and don't exploit this potential. They basically mimic the usual sinusoidal motions from the conventional camshaft electronically, thereby failing to realize the significant gains in economy and emissions cleanliness — not to mention multifuel compatibility — these new freedoms from electronic valve controls make possible."* [Sturgess 2010].

All found VVA system publications were focused on small and medium engine bore engines (Ø < 120mm bore), and GEV strokes were basically 6 - 10 mm (max 12mm), moving masses under 1kg, and actuating forces a few hundred Newtons. Overall, no fully flexible valve actuation systems were found in the large bore (Ø >180mm) engine area, except in [Aaltonen 2002] and the 2-stroke engine solutions of Wärtsilä and MAN B&W. All other large bore related systems were either 'lost motion' or 'hydraulic pushrod' type systems where the hydraulic pressure required by the GEV stroke has performed by camshaft, which will limit the operating range of the VVA system. On the other hand, two-stroke applications also have different requirements/boundary conditions due to moving masses, frequencies and valve lifts. Due to the lack of 4-stroke large bore system research publications, there was a clear opportunity within the framework of this study to perform a full research and development life cycle from concept level to full scale application on a running diesel engine.

4 HYDRAULIC DESIGN OF EHVA

There is a need to find out a suitable development platform for the EHVA system. Because the studied VVT system development is focused on marine diesel engines, the natural choice is the Wärtsilä W20 engine. The demand values of the valve actuation force and flow performance as well as cycle time of the W20 engine are the most suitable for the available hydraulic components. On the other hand, a W20 laboratory test engine is available for later full scale tests. The main technical specifications of the Wärtsilä W20 engine are: nominal rotation speed of 900 RPM, cylinder bore 200mm, power per cylinder up to 180 kW.

The development of EHVA can be divided in two separate parts, which are hydraulic and control system design. In the hydraulic system one has to pay attention to the hydraulic circuit and also to individual hydraulic components. The hydraulic control volume of the actuator is a critical component. Also the volumes of the hydraulic circuit affect the properties of the system. In the control system one has to pay attention to the controller and to sensors and other electrical parts.

4.1 Main demands

There are several different demands which the EHVA system should fulfil. The changing environment variables are challenging to the studied system. One demand is a proper work cycle. The load profile of the gas exchange valve (especially exhaust valve) is very challenging. At the beginning of the valve lift, the loading force is at its maximum. When exhaust gases are discharged from the cylinder, the force rapidly decreases. Then, depending on what kind of return spring is used, the force is increased again at the end of the valve lift, see Figure 13. The maximum force could be as high as 14 kN, and the force can change within a few milliseconds, which makes it challenging for the controller. In addition, in some cases the force could be negative, i.e., the force is trying to open the valve when the valve should remain closed. This is needed to ensure either the hydraulic or mechanical (spring) forces, which in turn affect the force balance of the actuator. The gas exchange valve lift event must be performed within a certain time in order to perform sufficient air flow in and out. This defines the opening and closing period's initial requirements, which are based on the timing of a conventional cam shaft. The velocity of the GEV must be controlled especially in the seating of the GEV, when it should not exceed its maximum value. Due to fast movements and relatively high moving masses, also inertia demands extra efforts from the control system. The long maintenance interval of the large bore ship diesel engine makes a demand on the components. In the maintenance interval, up to 20,000h, the endurance of the components should be equal.

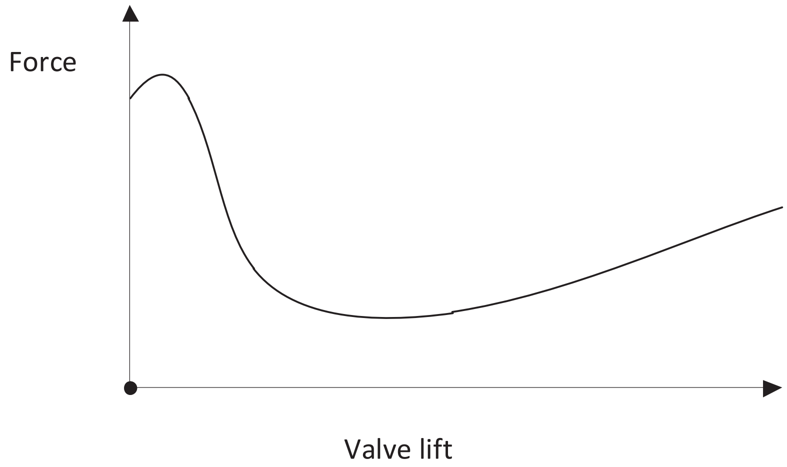


Figure 13 Sketched actuator force curve as a function of GEV lift (qualitative sketch)

Table 3 presents the boundary values of the development of the studied system

Table 3 EHVA boundary design values

Maximum opening time duration of the GEV	15ms
Maximum closing time duration of the GEV	15ms
Maximum GEV lifts	17mm
Moving masses of one actuator	3 kg
Static force of the actuator	-4 - 14 kN
Tracking accuracy of the GEV lift	±0.5mm
Pressure difference over the GEV in opening	2.8 MPa
Repeat accuracy (1000 events)	±1,5 CA
GEV seating velocity	< 0.5 m/s
Lifetime	20,000 hours

4.1.1 Tracking accuracy

In mechanical actuated systems, variation between the lifts of the gas exchange valve is relatively small. In the EHVA system, due to fully flexible actuation, one has to pay a lot of attention to keeping the multiple valve lifts inside a certain error range. Also, when the gas exchange lift curve can be defined more freely and can be changed during the engine run, it is desirable that the EHVA system can perform the valve lift as designed. The demanded error range of the gas exchange valve lift is defined as ± 0.5 mm.

4.1.2 Controllability

The main function of the VVT system is to react to changing environment parameters. This means that when the load or speed of the diesel engine changes, also the event of the gas exchange valve need to be changed. Normally, changes are relatively slow, and the system has plenty of time to adjust to the new conditions. However, in some cases, like generator drop off, response to these sudden, unpredicted changes is essential.

4.1.3 Repeatability

More important than the tracking capability is the repeatability of gas exchange valve lifts. Good repeatability ensures that combustion processes are as similar as possible. This ability enables to keep the running engine in balance between the cylinders. The diesel combustion process is basically a very stable process. This means that variations in the combustion process between adjacent strokes are not remarkable. The gas exchange valve opening allows some variation in valve timing and lift curve variation, providing that the fuel injection amount and timing are properly controlled. In an Otto cycle, like in gas engines, the process is much more sensitive [Rocha-Martinez 2002]. This leads to much higher requirements on the opening and closing timing of the gas exchange valves, especially the intake valve. Timing of the GEV is defined as certain, usually about 10% of maximum GEV lift. Experience has shown that a ±1.5 crank angle variation in timing does not have a significant effect on diesel engine performance.

4.1.4 Reaction in malfunctions, reliability

The maintenance period may be up to 20,000 hours in ship diesel engines, so the duration of the moving components is challenging. This will lead to up to 6×10^8 strokes of an individual component.

All feedback controlled systems need sensing devices of the controlled output. This may lead to a less reliable system. Because of the free valve events, collision with the combustion piston must be avoided. This situation awareness of the process requires sensor data. In some cases open loop control is possible with certain arrangements. A backup sensor could be installed to use in case of a malfunction of the main sensor. In case of emergency, the system should react as fast as possible and go into safe mode. The safe mode should be defined according to the dangerousness of the malfunction, because it is recommended to maintain as much functionality as possible in an emergency case (the system can be driven at limp mode). Also, the system should be able to react to wear and reduced performance of different components.

4.1.5 Usability

The test engine is often studied under extreme conditions to reach new information on the use of gas exchange valves and how these conditions affect emissions and engine performance. The usability of the control system is essential especially with a laboratory test engine. Like explained before, the control system should be able to perform gas exchange valve lifts according to certain pre-defined ranges, and this will require changes to the control parameters. Some of the parameters could be automatically adjustable (adaptive), or test personnel are needed to tune manually some of the parameters themselves. Knowledge about the effect of the parameters would then be required. Therefore, the number of manually tuned parameters should be as low as possible.

4.1.6 Energy consumption

The overall energy consumption of the valvetrain is essential. This energy is called a 'parasite loss', which is a disadvantage but cannot be totally eliminated. The conventional electro-hydraulic system also has another drawback when compared to the mechanical system. The hydraulic pump will get approximately constant energy from the diesel engine even if the energy needed by the valvetrain changes remarkably during the gas exchange valve stroke. Moreover, the constant energy must be sized according to the maximum temporary force during the gas exchange valve lift.

Energy recovery is difficult to apply because of fast movements and short deceleration times. Also small actuator chamber volumes and relatively large dead volumes will decrease the efficient recovery. High flow peaks (up to 100 L/min) require good flow performance of the hydraulic control valve, which usually leads to poorer switching time performance.

4.2 EHVA mechanical and hydraulic systems

The intake and exhaust valves both have individual actuators in the W20 diesel engine, where one actuates two gas exchange valves together via the yoke. At this point, the construction of the valvetrain has been kept as close to the original as possible in order to keep changes at a minimum. This helped to avoid any new problems with durability and reliability. Also the possibility to easily retrofit the EHVA to the existing engines has been kept in mind.

First, the hydraulic circuit has been chosen for the concept study of the EHVA. The used actuator is a single stage type. The actuator is either one side controlled, when only the upper actuator chamber flow is controlled (3-way control), or double side controlled, when both chambers are controlled (4-way control), Figure 16. The closing of the gas exchange valve can be made with any combination of fluid pressure force and the return spring force. The control is done with one or more hydraulic directional control valves, or with on/off valves. The fluid power source is a variable displacement pump, which will take the input power from the diesel engine in the final application.

The hydraulic force of the actuator should overtake the shutting forces of the GEVs and accelerate the GEVs to the opening speed. When the exhaust valve starts to open, the forces against the movement are cylinder pressure and force of the return spring (Figure 14).

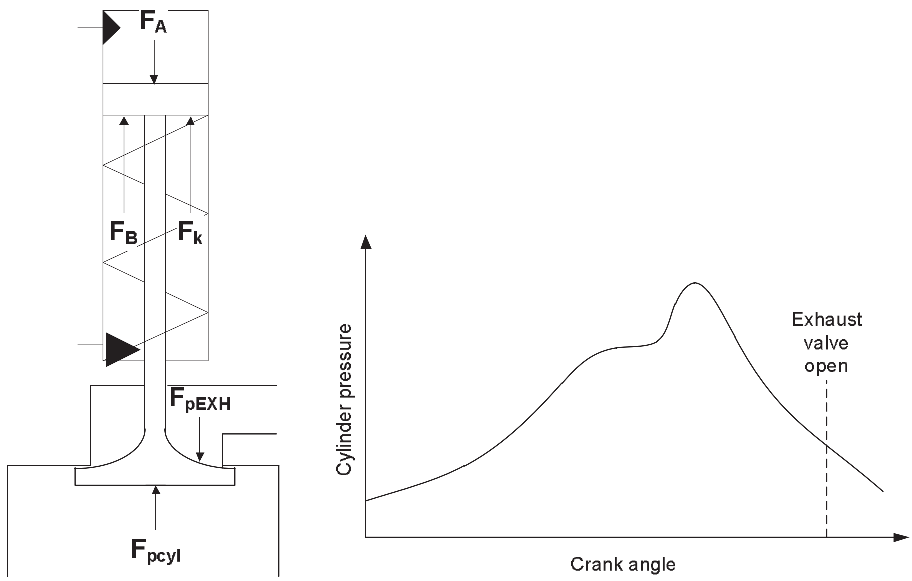
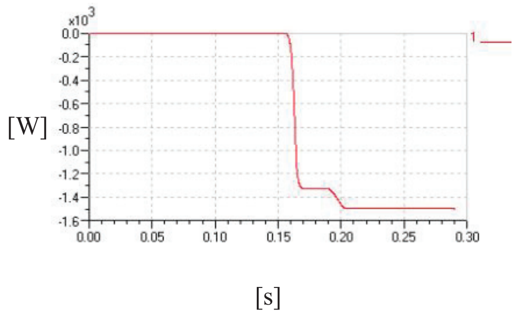
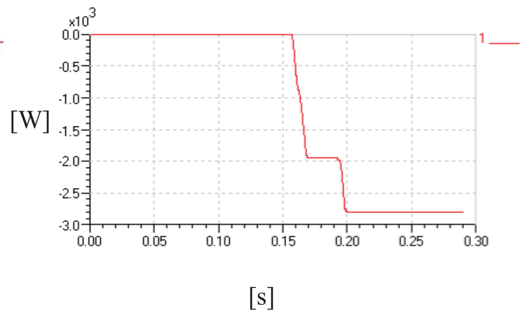


Figure 14 Actuator forces and schematic cylinder pressure curve

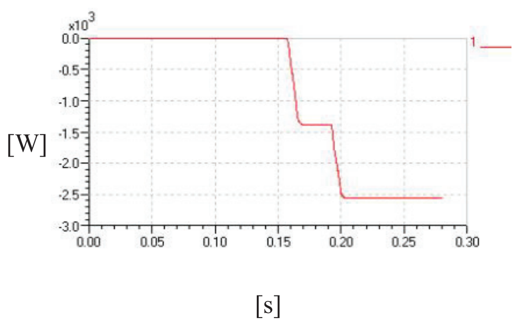
Four different alternatives of hydraulic circuits have been taken under investigation, and simplified concept simulations have been performed. Control valve characteristics were similar in all simulated cases. The studied hydraulic systems were a 4-way control valve without a return spring where both actuator chambers were controlled by one spool valve, a 3-way control valve with a return spring where the upper chamber is controlled by a spool valve, and two different 3-way controlled systems where the control areas of the actuator are changed during the valve lift by different mechanical constructions or components (according to US patents 5595148 "Hydraulic valve control device" and 5531192 "Hydraulically actuated valve system"). All systems also include some kind of hydraulic damper solution or end cushion in order to keep the seating velocities under control, though the end cushions are not necessarily needed if the actuator position is properly feedback controlled. The first simulations have been made by open loop on/off control. These concept simulations showed clear differences between hydraulic circuits, shown in Figure 15, where cumulative power consumption of one stroke is calculated. Values of the simulated power consumptions are comparable only between the investigated systems in the current simulation model.



3-way



4-way



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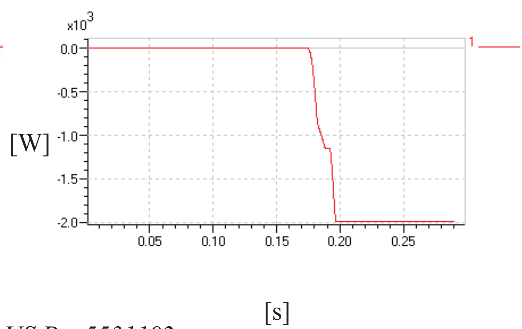

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Figure 15 Simulated energy consumptions of different hydraulic concepts

According to the concept simulations, when energy consumption, simple construction and controllability have been taken into account, simple 3-way system has been chosen (Figure 16) [Herranen 2007]. Used control valve can be either on/off or servo/proportional valve. The same type of hydraulic circuit has also given good results in another, similar type of application [Haikio 2006].

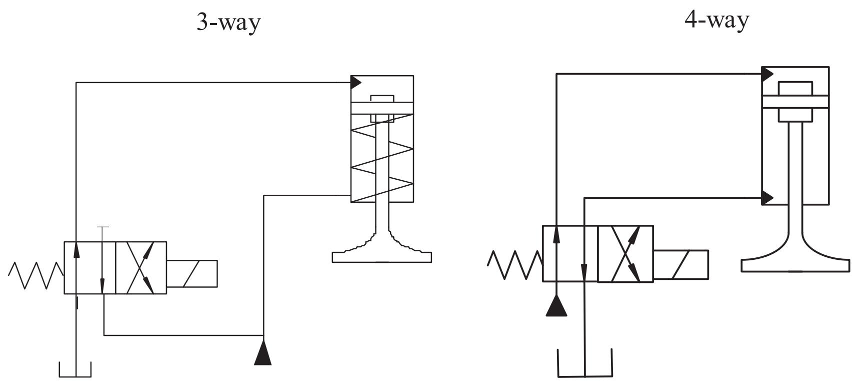


Figure 16 3-way and 4-way controlled actuator

After the principle of the hydraulic circuit has been chosen, the next step is the component enquiry. The required pressurized area of the actuator is defined by means of effecting forces. The area leads to the flow demand which the hydraulic system needs to deliver to the actuator. Supply pressure of 30 MPa has been chosen, and this leads to a 30mm/28.5mm = 1.05 diameter ratio of the actuator, but flow of the lower chamber is not directed through the control valve. Simulated flow rates over the control valves are given in Figure 17.

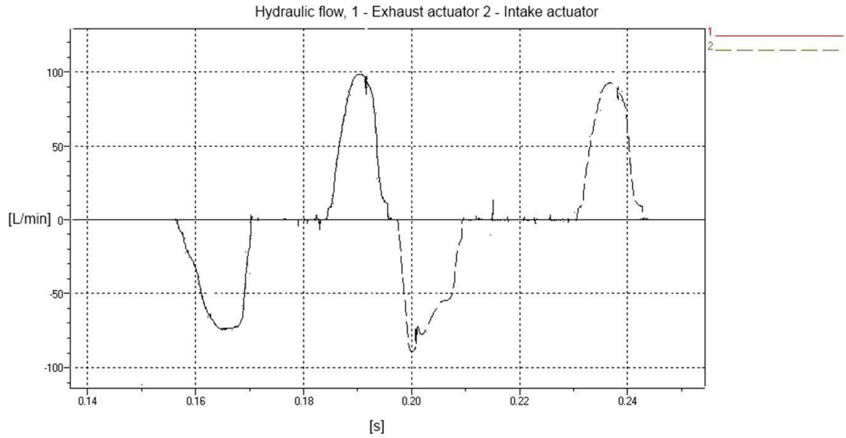


Figure 17 Flow rates through the control valves during the stroke

The flow performance of the control valve needs to be high enough, so that the actuator can open and close the GEV within the required time. Valve flow performance and effect on the actuator opening is shown in Figure 18, and closing in Figure 19.

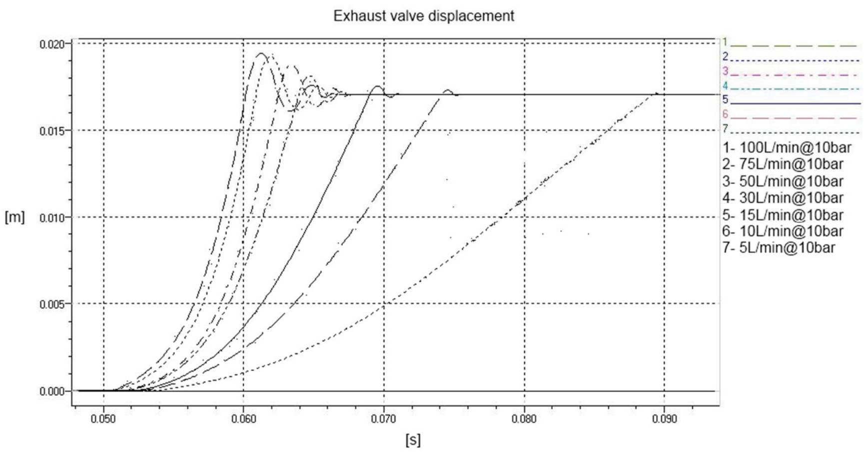


Figure 18 Effect of the control valve flow performance, GEV opening

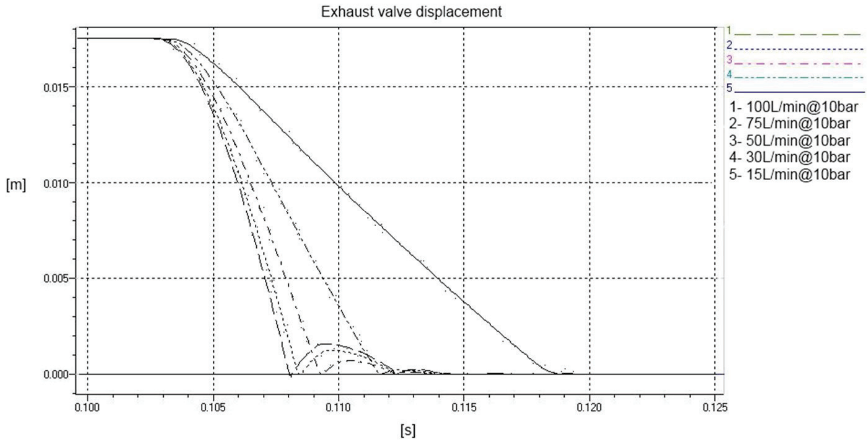


Figure 19 Effect of the control valve flow performance, GEV closing

The limit of the required flow performance seems to be more than 30 L/min at a 1 MPa pressure drop over the valve. Thus a 50L/min at 1 MPa valve is chosen for further simulations. The effect of the control valve's natural frequency (defined as -3 dB change in amplitude ratio or -90 degree phase lag of the valve spool second order response) is investigated and the results are shown in Figure 20. It can be seen that a valve faster than 100 Hz does not give any improvement in the on/off opening, though in a closed loop feedback controlled system, frequency of the valve should still be greater than the system's natural undamped frequency.

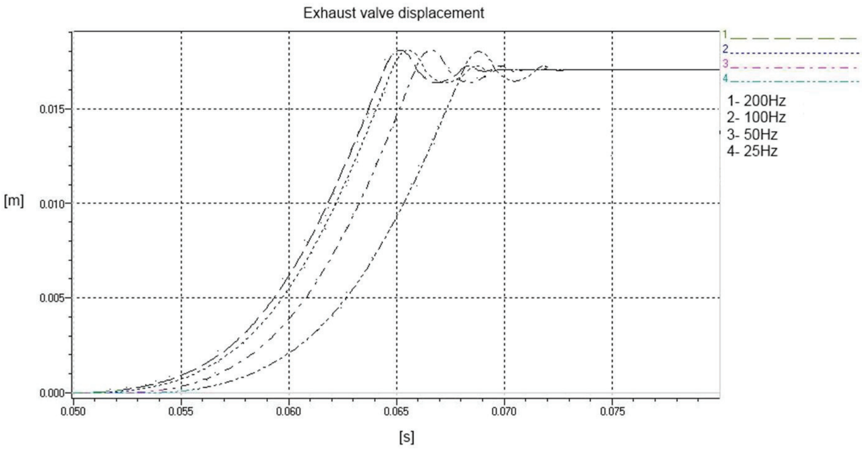


Figure 20 Effect of the control valve natural frequency, GEV opening

When choosing the control valve, servo valves have been rejected due to zero point drifting problems as a function of temperature or supply pressure, energy consumption caused by high pressure difference and constant leak flow, lack of robustness because of small orifices and a sensitive working principle, and the price caused by expensive materials, complex design and demanding manufacturing tolerances. A hydraulic valve based on voice-coil technology offers sufficient performance values: 100 L/min at a 3.5 MPa pressure difference over the control edge (which equals required flow performance in the above simulation), -3db bandwidth of 200Hz and a 90 degree phase lag. The current D3FP type of valve is CETOP 5 size, but it is also available in a smaller but faster size D1FP (CETOP 3 valve). Because the control valve is an important part of the EHVA, and the entire simulation models have been verified with test bench measurements in Chapter 5.

Suitability of a smaller valve for EHVA has been tested in simulations (Figure 21), which show that a shorter valve lift can barely be performed within the required time, but valve controllability is reduced due to saturation of the flow capacity.

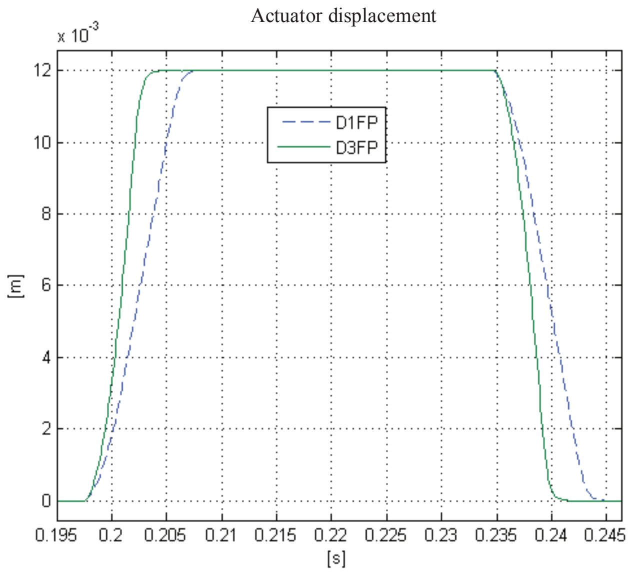


Figure 21 Effect of the chosen control valve size, GEV full stroke

Certain hydraulic circuit parameters affect the dynamic behaviour of the system. Hydraulic dead volumes have a significant effect on the vibrations and responses of the system. One goal of the this work is to minimize the dead volume effect on the characteristics of the system by locating the control valve as close as possible to the actuator, and decreasing the diameter of the flow lines as much as possible.

The effect of the dead volume can be seen clearly when the actuator is stopped as fast as possible in the middle of the lift. The increase of the system vibration is drastic when the dead volume goes over 150 cm^3 (Figure 22) in the pressure line between the control valve and the actuator. The volume effect has been tested also by adding dead volume instead of increasing the pipe diameter, and both methods give similar results. In a P-controlled stroke, the dead volume between the control valves and actuator has a strong influence on the beginning of the stroke. Opening is delayed due to time taken by the pressure rising, and this poses challenges on the controller due to the dynamics of the system. The settling time of the position also heavily increases with the dead volume (Figure 23).

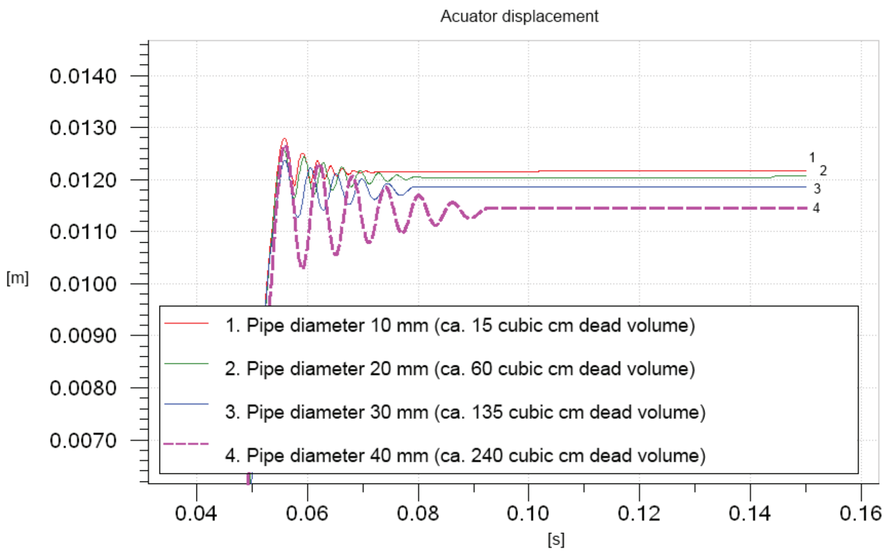


Figure 22 Effect of the dead volume defined by pipe diameters

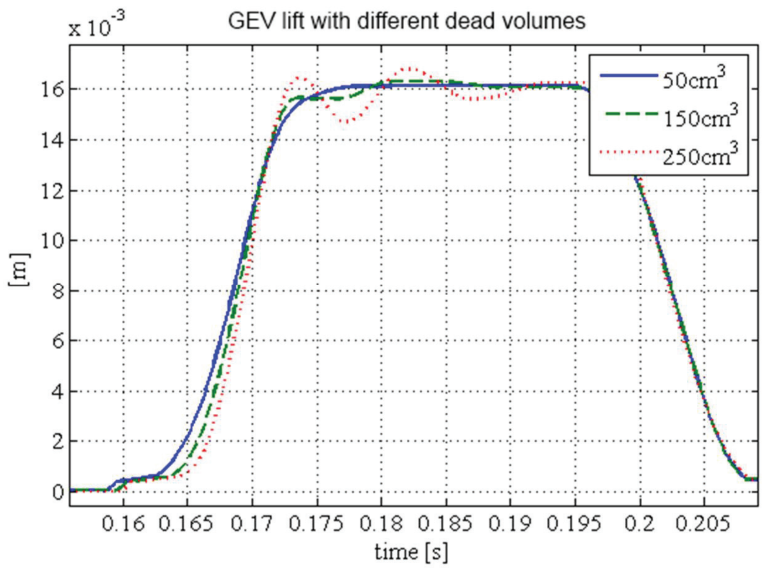


Figure 23 Dead volume effect on the P-controlled GEV lift

Flow line design has been supported by simulation results, too. Flow resistance of the line is tested and as small as possible drilling is used. According to the simulations, it is clear that Ø7 mm or larger pipe/drilling has no more effect on the movement of the GEV (Figure 24) with control valve full opening.

Actuator displacement

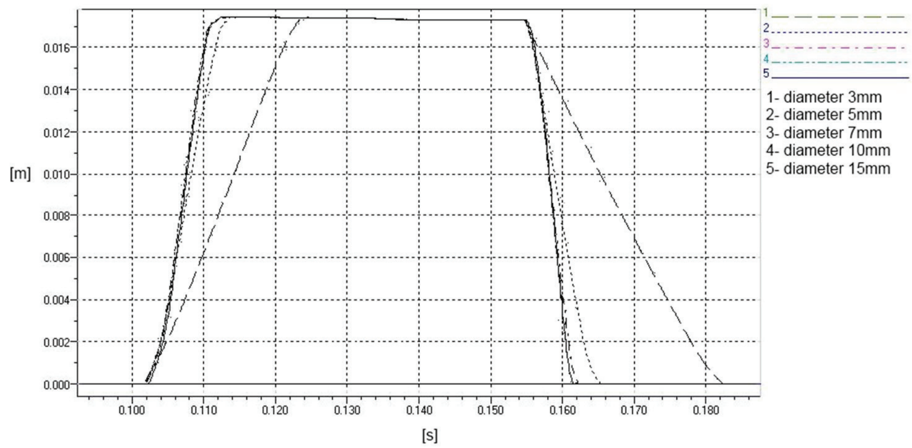


Figure 24 Effect of the connection line diameters 3-15mm

Once the control valve and the actuator size are locked to 50L/min at a 10bar pressure difference, the hydraulic supply pressure has its own effect on the actuator movement. Initial design supply pressure was 30 MPa, and the effect on the different pressure is shown in Figure 25.

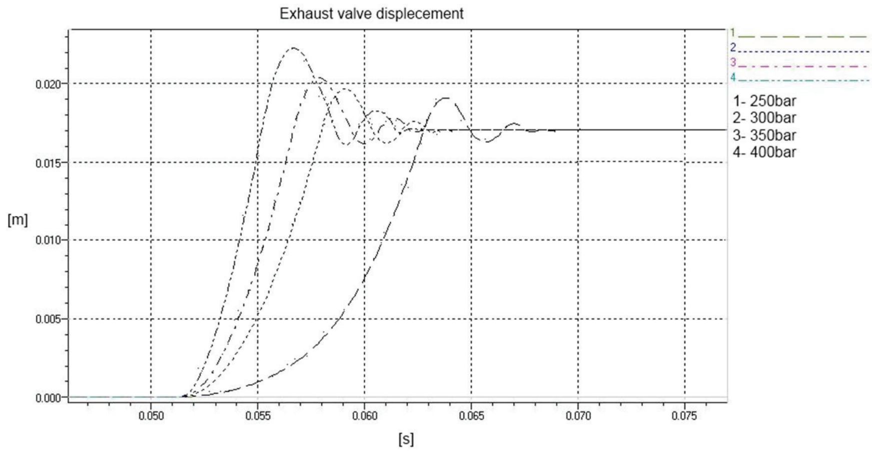


Figure 25 Effect of the supply pressure, GEV opening

Simulations have shown that the maximum needed volumetric flow is up to 100 L/min. It is not reasonable to use a hydraulic pump capable of such high constant flows, but instead a pump which has the capacity to produce the mean flow during the run. Thus, resources for the high flow peaks must be created, and this can be done by adding a rail/accumulator before the control valves. The volume of the rail or accumulator will then affect the pressure during the flow peak, and at some point it will be show in the actuator movement due to a pressure drop (Figure 26). The rail is used if the maintenance free component is required. The hydraulic accumulator is less reliable, but the physical size of the accumulator is smaller if the rail and accumulator have same effective function. Pressure drops with different size accumulators before the control valves are shown in Figure 27. According to the simulations, about a 0.3L accumulator has decent capacity to maintain pressure high enough in the working pressure line.

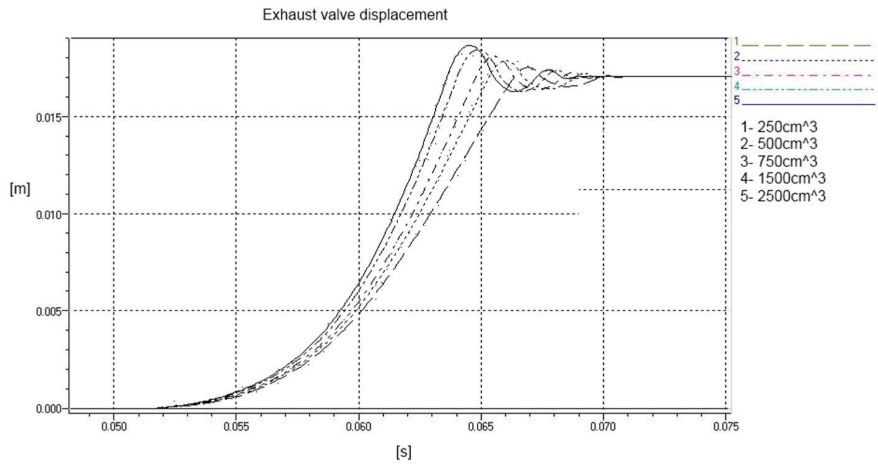


Figure 26 Effect of the rail volume, GEV opening

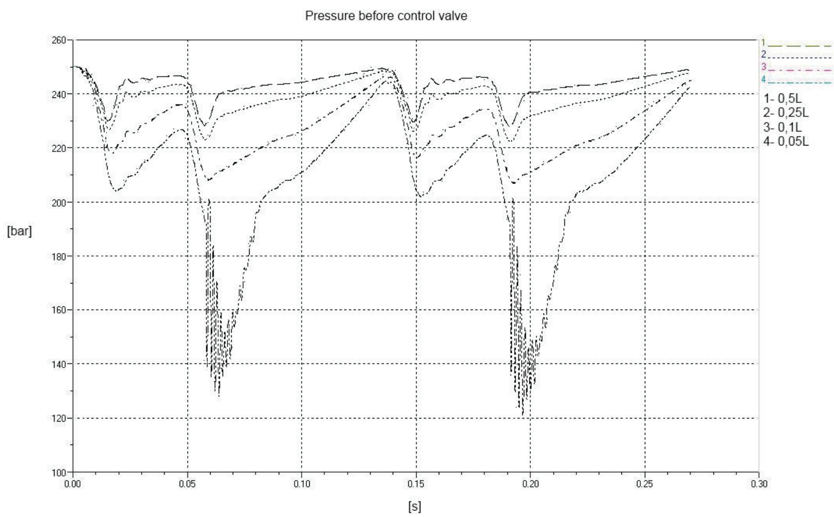


Figure 27 Effect of the different accumulator sizes, pressure before control valves

Figure 28 shows the schematic hydraulic circuit of the chosen EHVA system. The 4-way control valve (though used in 3-way) controls the upper chamber of the actuator. When the chamber is pressurized, the pressure force rises high enough, and the hydraulic oil flows to the chamber. The actuator pushes the yoke, which opens the gas exchange valves. When the GEVs need to close, the upper chamber is connected to the tank line. The return springs and constant pressure of the lower chamber affect the actuator, and push the actuator and the GEVs to the close position.

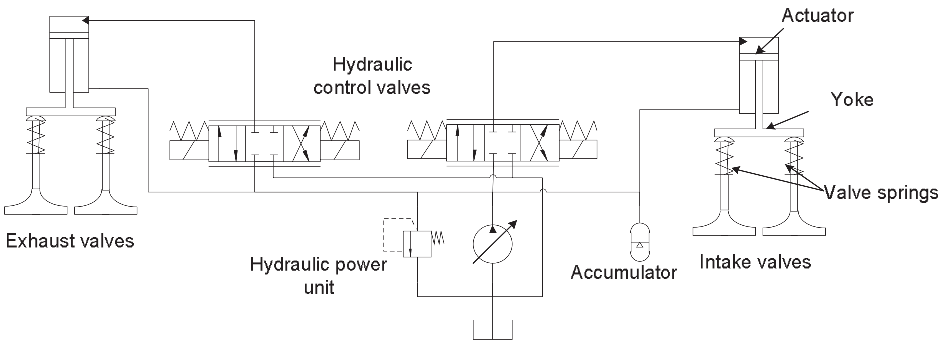


Figure 28 Principle of the EHVA

The contact between the actuator and yoke (and yoke/valve stem) is like tappet contact, and it cannot ‘draw’ the yoke or valve step upward to the close position. The return spring is needed in the system because the GEV should remain in the closed position when the pressure difference over the GEV is negative. This means that pressure in the exhaust or intake manifold is higher and tries to open the valve. In the 3-way controlled system also the effect of the hydraulic force has been investigated (Figure 29). The return spring alone can close the GEV in the required time, but the actuator's hydraulic force makes the closing time shorter. The return spring is thus a safety device too, and it can close the GEV if power of the control valve is lost. Because of this, the control valve power off position is chosen so that the opening of the control chamber is connected to the tank.

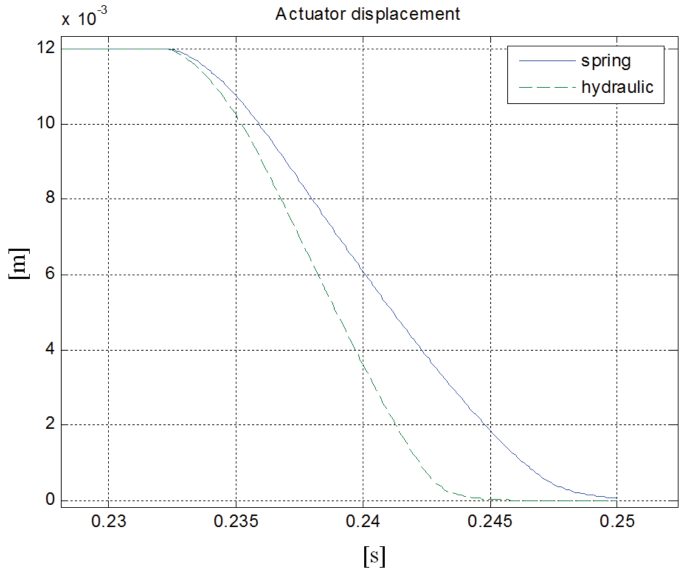


Figure 29 Simulated closing of the valve with and without lower chamber hydraulic pressure

The effect of the oil viscosity is tested because in an engine environment the used oil is engine oil, and its viscosity changes more than that of high grade hydraulic oil. Figure 30 shows the simulated GEV lift with 46cSt and 138cSt viscosities. It can be seen that the only major difference is found in the seating velocity, where the closing is decelerated more with higher viscosity oil.

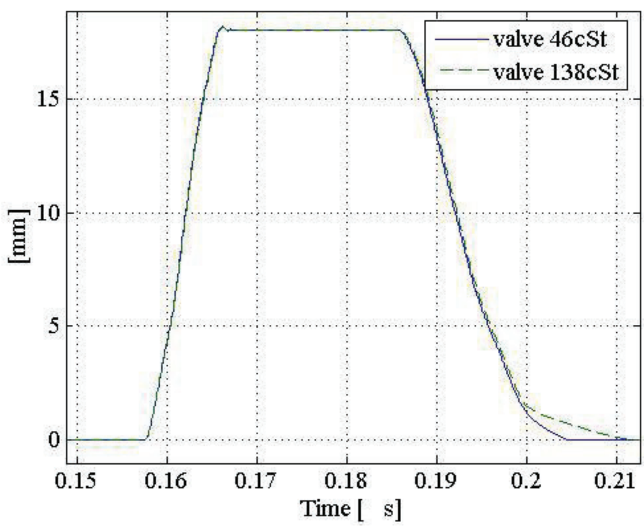


Figure 30 Oil viscosity's effect on the GEV displacement

Finally, the unwanted situation where the combustion piston hits the GEV is simulated. The results depend on the position of the control valve spool and whether the GEV is moving toward or away from the piston. If the control valve and GEV are open, the pressurized oil can partly flow through the valve to the pressure line, decreasing the pressure peak (Figure 31). The duration of the collision is only a few milliseconds, and overlap of the piston and GEV is about 5mm.

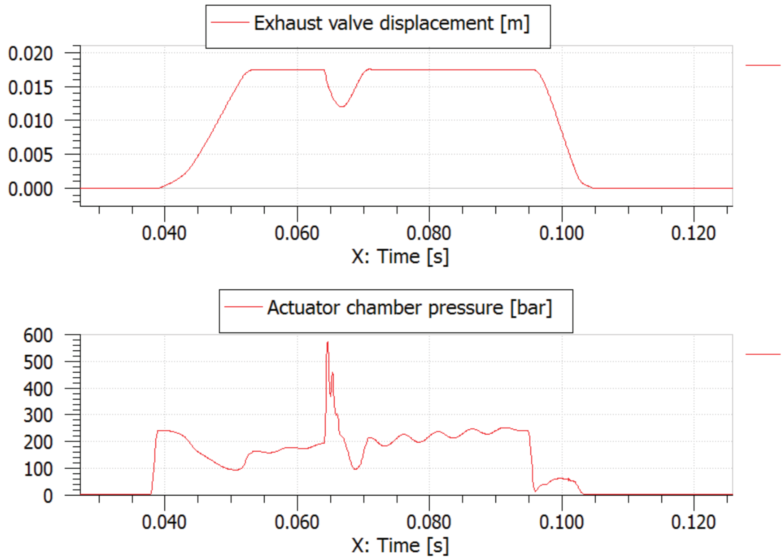


Figure 31 GEV displacement and pressure in the actuator chamber during a combustion piston collision

4.3 EHVA control/electrical system

The controller of the actuator and its development are discussed in more detail in later sections of this study. Like stated earlier, measurements of the different diesel engine and EHVA components are required for the controlled and reliable actuation of GEVs. The crank angle of the diesel engine is the primary variable, and the GEV event is defined as a function of CA. The measurement of CA is already used because of the diesel engine injection system, so measurement is available, and the controller only changes the lift profile from the CA domain to the time domain for the hydraulic valve controls. The displacement of the combustion piston, which is used for safety definitions, can be calculated from CA. In the simulations and test rig measurements, CA data have been generated virtually. The additional compulsory measurements required by EHVA are the displacements of the EHVA actuators. Hydraulic pressure measurements as well as the control valve spool displacements are normally used for diagnostic purposes if the used controller type does not require them.

GEV position measurements also have some challenges. The measured distance of the GEV is up to 17 mm, the accuracy should be better than 0.1mm and the frequency response is around 1 kHz. The measurement from the GEV stem or head has very limited space and the valve head vibrates heavily causing inaccurate measurements if not compensated by two opposite sensors. The engine vibrations and the ambient temperature have to be tolerated. The measurements of the valve lift have been decided to be brought away from GEV assembly itself, measuring the displacement of the actuator instead. After this the GEV position could not be ensured during the heavy deceleration in the GEV opening, because contact between the yoke and actuator is not fixed. A non-contact sensor is desirable, and LVDT-type sensors have been chosen. The sensing element is attached to the actuator or a sensor probe is entered inside the actuator. This means that hydraulic pressure puts strain on the displacement sensor, too.

The used data acquisition and control system was the dSPACE® DS1003 digital signal processing board and Simulink® configuration. Controller turnaround time is aimed to be kept smaller than 0.2 ms if possible.

5 MATHEMATICAL MODEL OF EHVA

5.1 Mathematical model of the hydraulic actuator system

A schematic picture of the actuator is shown in Figure 32.

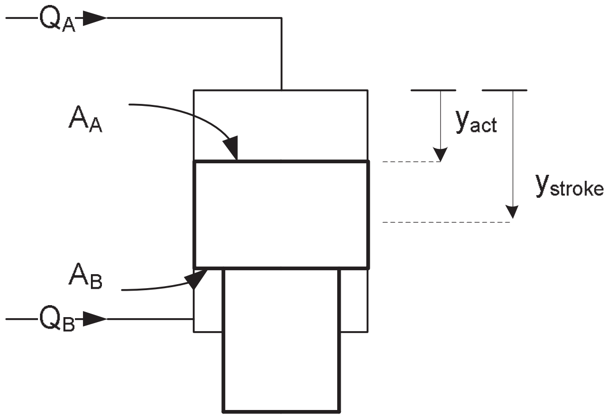


Figure 32 Schematic picture of the actuator

The force equation of the actuator is the following:

$$m_{vt}a_{act} = F_A - F_B - k_s y_{act} - F_{spre} - F_{Fact} - F_{GEV} \qquad \text{(Eq.1)}$$

Friction F_{Fact} is defined by three different parameters. The stiction force is the maximum friction force when the mass is at rest and the Coulomb friction force is the friction force (assumed constant) when the mass is in motion. Viscous friction is proportional to actuator relative speed and adds damping to the system.

The pressure equation of the actuator chamber A is

$$\frac{dp_A}{dt} = \frac{B_{eff}}{V_{A0}+A_A y_{act}}\left(\sum Q_A - A_A\frac{dy_{act}}{dt}\right) \qquad \text{(Eq.2)}$$

For the B chamber, the equation is

$$\frac{dp_B}{dt} = \frac{B_{eff}}{V_{B0}+A_B(y_{stroke}-y_{act})}\left(\sum Q_B + A_B\frac{dy_{act}}{dt}\right) \qquad \text{(Eq.3)}$$

Leaks of the system are not modelled. The hydraulic flow to the actuator chamber through the hydraulic control valve can be described as

$$Q_A = \mu A_{spool}\sqrt{\frac{2\Delta p_{valve}}{\rho_{oil}}} \qquad \text{(Eq.4)}$$

The flow area of the valve is dependent on spool movement, and dynamics of the valve spool can be described as second order lag equation

$$K_v u = \frac{1}{\omega_v^2}\frac{dx_v}{dt^2} + \frac{2\delta_v}{\omega_v}\frac{dx_v}{dt} + x_v \qquad \text{(Eq.5)}$$

The hydraulic undamped natural frequency of the actuator system is

$$\omega_{act} = \sqrt{\frac{B_{eff}A_A^2}{V_{A0}m_{vt}}} \qquad \text{(Eq.6)}$$

The undamped natural frequency of the 3-way system is lower than that of the 4-way controlled system because only one volume contributes an oil spring and only one line is controlled. This leads to a worse dynamic response of the system, and the system static and dynamic errors are more sensitive to the loads [Merrit 1967]. Thus the relative dynamics of the control valve is better.

The contact between the actuator and yoke, and between the yoke and the GEV stems, are modelled with AMEsim elastic contact submodel. According the AMESim, when the contact occurs, a contact force in is applied to both bodies. This consists of a spring force and a damping force. To give continuity in this force, the damping coefficient is modified so that it is zero at first contact and approaches its full value asymptotically, achieving 95% of its full value at a penetration in mm specified by the user. The force during the contact is calculated as

$$F_C = k_c x_c + r_c(1 - exp^{\left(-3\frac{x_c}{e_c}\right)})v_c \qquad \text{(Eq.7)}$$

Flow resistance calculation of the AMESim pipe models is based on Reynolds number which changes according the viscosity of the oil, flow velocity and diameter of the pipe (Eq.8) and relative roughness rr (ratio of the average height of asperities to the tube diameter). Friction coefficient λ is determined by a cubic spline interpolation based on a table of empirical values from harp of Nikuradse. Then the pressure drop in the pipe line can be calculated from Darcy-Weisbach equation, Eq.9.

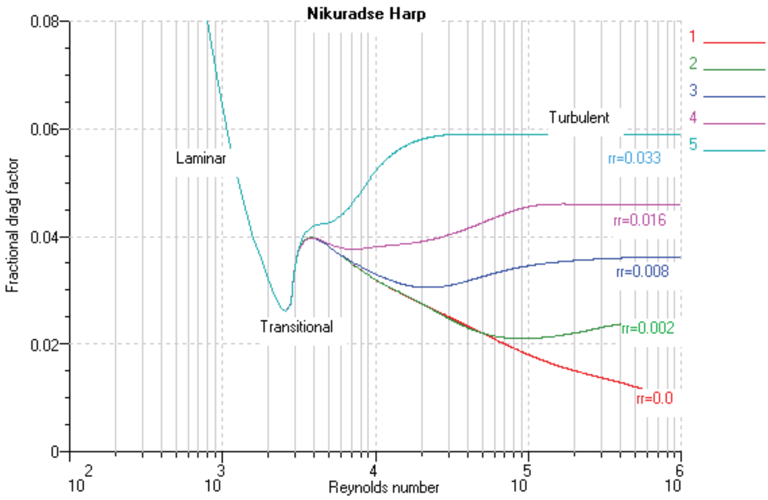


Figure 33 Nikuradse harp

$$Re = \frac{D_{pipe} v_{pipe}}{v_{oil}} \tag{Eq.8}$$

$$\Delta p_{pipe} = \lambda \frac{l_{pipe}}{D_{pipe}} \frac{\rho_{oil} Q_{pipe}^2}{2A_{pipe}^2} \tag{Eq.9}$$

A simplified simulation model is shown in Figure 34. Parameter list of the main components is presented in Appendix I.

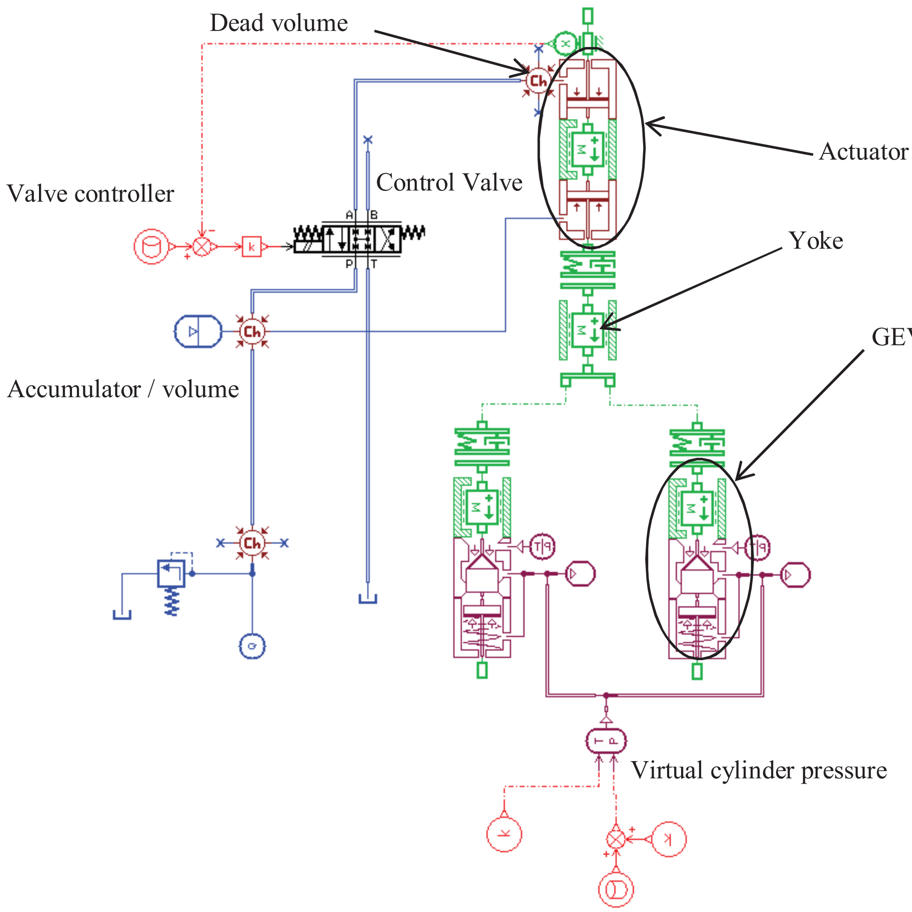


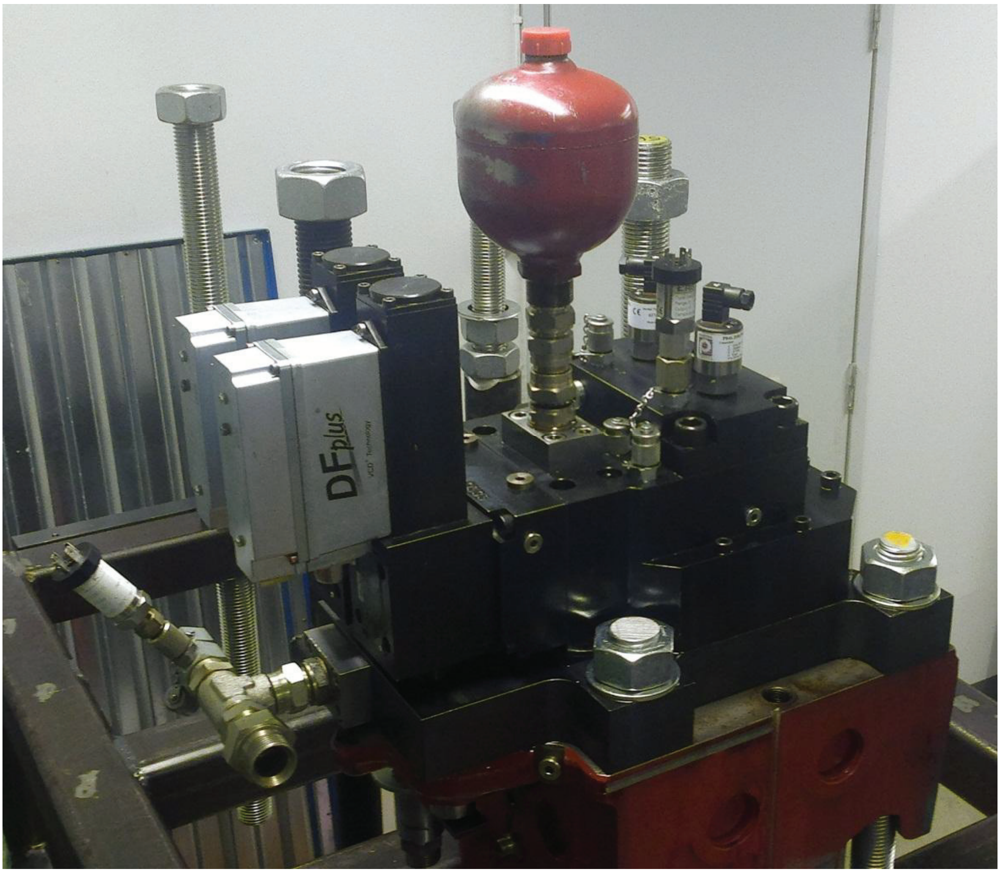
Figure 34 Simplified AMESim simulation model of one pair of GEVs [Herranen 2009]

5.2 Verification of the simulation models

The simulations have been carried out with different simulation programs, AMESim, and Matlab/Simulink, or their combination. This is due to the different strength of each simulation program, for example AMESim is suitable for modeling of overall hydraulic systems while controllers are easier to build in a Simulink environment. Co-simulation of Simulink can be used with AMESim too. Also the control/data acquisition system used can later use Simulink models directly.

In the simulation phase of the study, the focus is mainly on the tracking ability of the EHVA system. Later, also some pressure peak and vibration studies are performed, but the main interest is usually in the actuator movement performance.

The simulation models have been verified by measurements. The measurements have been done with a laboratory test rig (Picture 1), where the valvetrain of the W20 engine has been installed on the cylinder head, and GEVs have been actuated by the currently investigated EHVA system. Verification is done mainly to the electrohydraulic model. The displacements of the actuators are verified, and the hydraulic pressures compared. After the mechanical properties of the system have been proved, different controllers are easy to compare. Also different hydraulic circuits could be tested with adequate accuracy.



Picture 1 EHVA test rig

First, the basic system with open loop control has been verified. The pressure in the actuator chamber during the motion matches quite well (Figure 35), and this leads also to good agreement between the displacement curves of the actuators (Figure 36). Actuator system natural frequency (Eq.6) is shown as a vibration in chamber pressure when the GEV is stationary open. Pressure vibrations may cause additional mechanical stress to the system, disturb the control and cause unstability. The reason why the simulation pressure does not vibrate in this particular result, is that actuator is pushed here against end stop, and in addition large dead volume is modelled between pump and control which is giving more stable feed pressure.

The actuator and gas exchange valve are investigated separately in order to find possible contact loss between the parts. The contact loss is not desirable; it causes noise and additional stress between surfaces in contact. This can be clearly seen in the case where EHVA has been tested with parameters where the actuator is stopped very rapidly. The sudden stop causes contact loss between the yoke and actuator, and this can be seen in both measurements and simulations, Figure 36. In order to keep the contact between moving parts, inertial forces of the actuator should not exceed the force provided by the return spring.

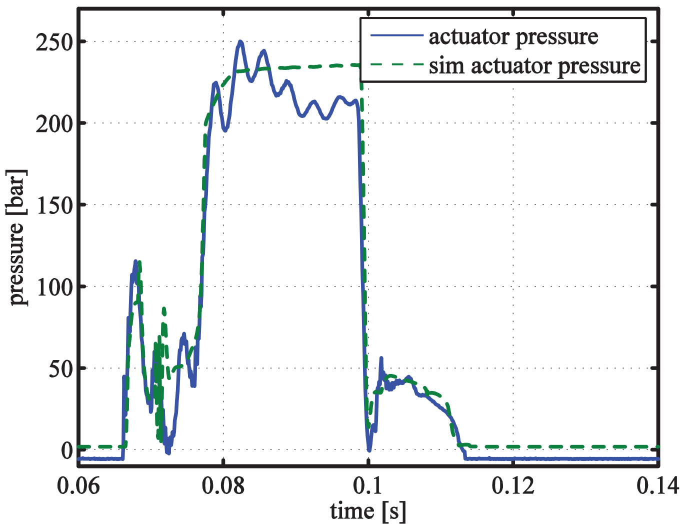


Figure 35 Simulated and measured actuator pressures [Herranen2007]

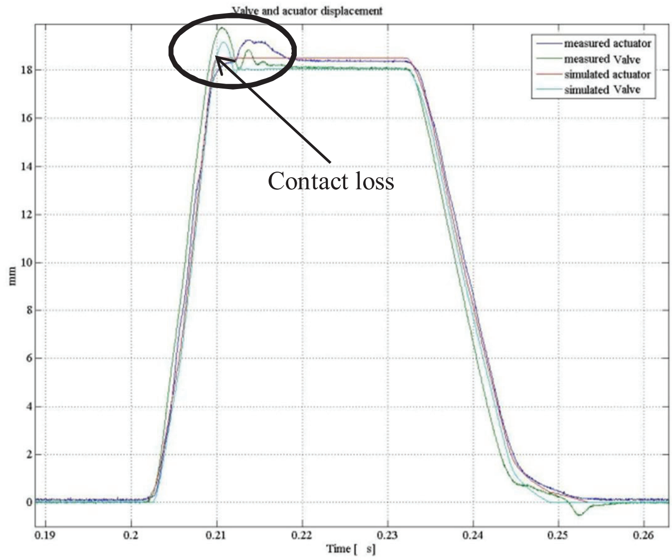


Figure 36 Contact loss verification [Herranen 2007]

Next, P-control simulations are verified. Figure 37 shows the measured and simulated step-wise responses. These simulations show also that the model is verified and the tuning of the controller parameters can be done. [Herranen 2009].

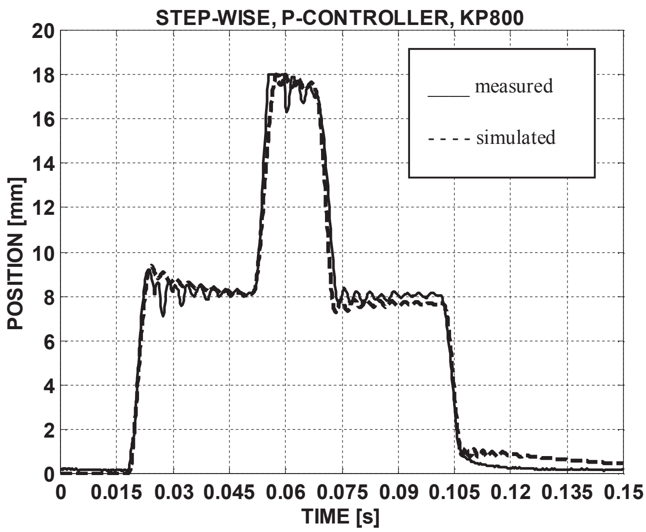


Figure 37 Simulated and measured step-wise responses of the actuator [Herranen 2009]

6 BASIC CHARACTERISTICS OF EHVA

The main design requirements were: GEV opening and closing times <15ms, max. pressure difference over the GEV 2.8 MPa, a return spring pre-tension of 1 kN + 60 N/mm spring constant. A nominal tracking accuracy should be kept inside ±0.5 mm during the whole GEV lift. This led to an actuator with an upper diameter ratio of 30/28.5mm.

Open loop tests showed that opening and closing could be done in the required time range. Now, this has been tested with a P-controller too. Figure 38 shows the step response of the EHVA GEV lift. Also closed loop controlled lift events are inside the time range.

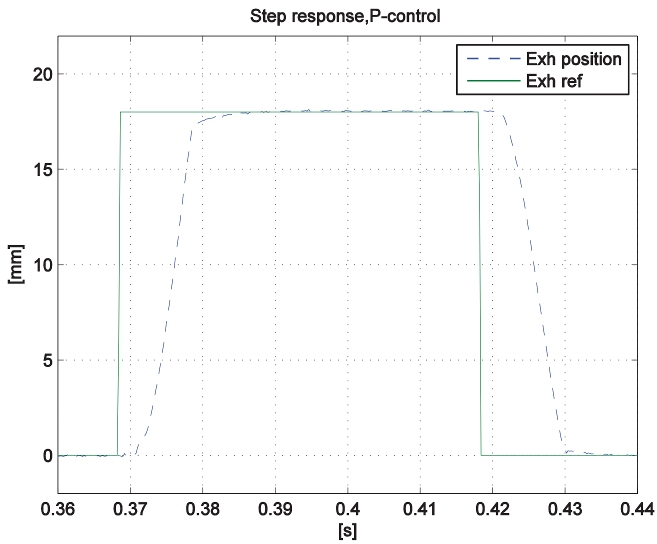


Figure 38 P-controlled step response measurement of the GEV [Herranen 2009]

Opening tests have been done against atmospheric pressure, because cylinder pressure has not been able to be produced in the test rig. In simulations, measured cylinder pressure data is fed to the GEV model. So pressure in cylinder volume is always changing similarly as a function of CA, and is not changing according the simulated gas discharge from the volume. Simulation tests showed that the highest designed cylinder pressure (pressure difference over the GEV) will delay the opening by 5ms at maximum if the control valve opening was kept constant but not fully open. In feedback controlled opening (Figure 39), the load effect is decreased.

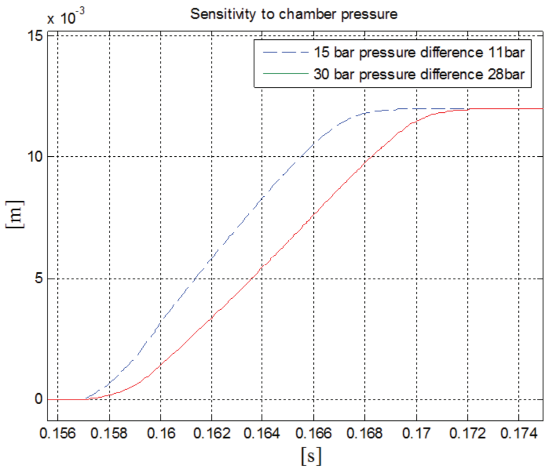


Figure 39 Simulated effect of the cylinder pressure

Because the delay is the largest single source of error [Herranen 2009], delay compensation is needed. Because the reference curve was the function of the crank angle, it is relatively easy to advance the reference fed to the controller just by shifting the CA value of the lift profile data. In Figure 40, the desired target curve and reference curve to the controller are on the top of each other. This will lead to huge tracking error, up to several millimeters. In Figure 41, the reference has been advanced, which results in the actuator displacement match being much better on the target.

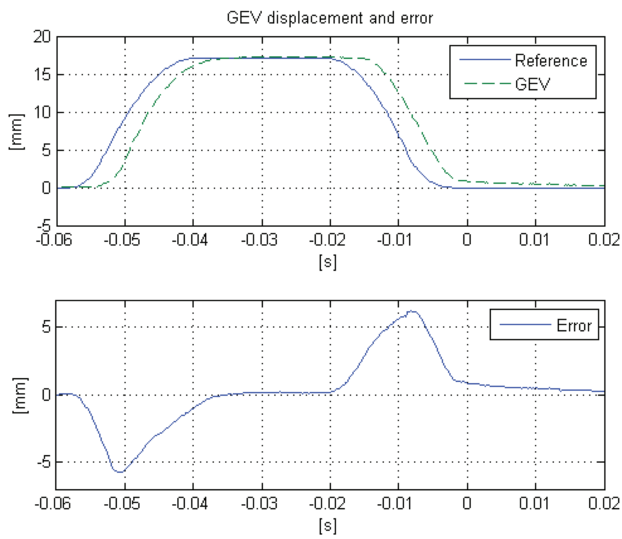


Figure 40 P-controlled measured tracking error of GEV, without delay compensation

Figure 41 shows that P-controlled displacement still cannot fulfill the requirements, especially in the closing part of the valve lift. Because the hydraulic area of the actuator is strongly asymmetric, also the gain of the P-controller has been changed when the closing part of the GEV is applied. Tracking error with asymmetric P-gain is shown in Figure 42. Now the error is already close to the required range, but it is tuned only to the initial working conditions. When the load or speed of the engine changes, the P-controller has trouble to perform accurate enough tracking. This has been found especially when running EHVA on a real 4-cyl engine under full load (Figure 43). Also the control of the GEV seating velocity is not easy with P-control.

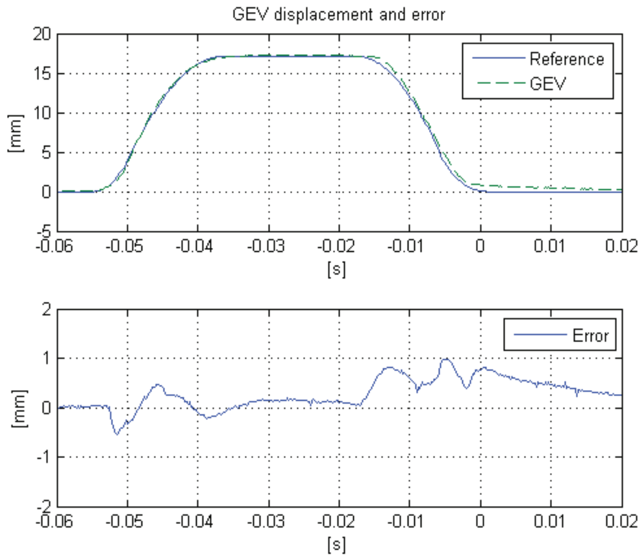


Figure 41 P-controlled measured tracking error of GEV, symmetric gain

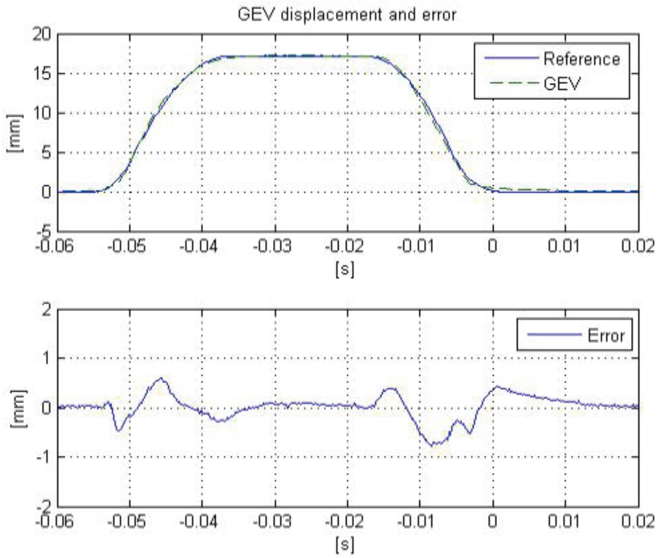


Figure 42 P-controlled measured tracking error of GEV, asymmetric gain

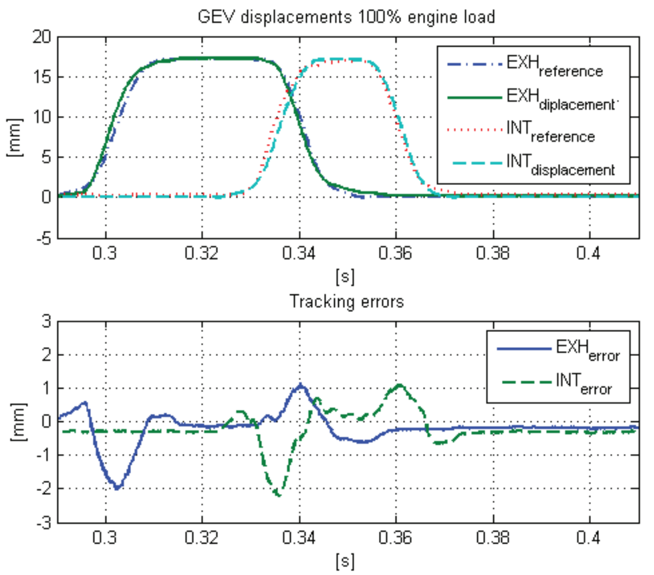


Figure 43 P-controlled GEV lifts in running engine, 100% load

Repeatability of the GEV stroke is measured in position when the GEV is for example 1.5 mm open. Test rig measurements showed that a timing window of 3CA degrees is reached when measuring the timing of 1000 strokes (Figure 44).

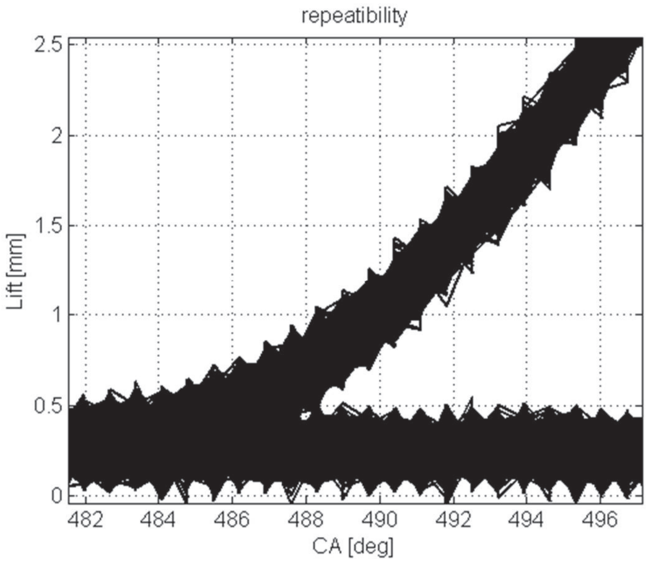


Figure 44 Measured opening repeatability of exhaust valve, 1000 strokes.

Although the basic requirements of the EHVA could be reached in the test bench with careful tuning, a lack of proper controllability in a real, varying environment necessitated a better controller of the EHVA.

7 EHVA CONTROL SYSTEMS DEVELOPMENT

Studies have indicated that despite the simulations, the required tracking ability needs a more complex controller than the P-controller [Herranen 2009], [Virvalo 2008]. The most commonly used controller in the industry is a PID controller. The PID is nevertheless rejected because of several reasons. A disadvantage of the integral term is that it increases the oscillatory or rolling behavior of the controller output. Also, the three tuning parameters of the PID controller interact in their influence and it is sometimes difficult to determine what action to take if controller performance is not as desired. Another disadvantage relates to the uncertainty in the derivative computation for processes that have noise in the measured process variable. [Cooper 2005]. When relatively fast responses with short settling times are required, there will be overshoots and limit cycling when PID controllers are tuned to work. That is why a non-linear controller has better performance [Virvalo 2001].

A State-space model has been built up in order to find out basic dynamics of the studied system. But because non-linear simulation model is also built up, it is used in further simulations and controller performance tests are mainly applied by experimental measurements.

The state controller was the first tested type of controller which could fulfil the requirements of the tracking error in varying environments. However, usability of the state controller is poor because of the number of parameters. This leads to a quest for a more automatic or adaptive type of controller. Adapting the state controller parameters is considered, but rejected due to the number of parameters, the interactive nature of them, and known problems of the online measure based derivative calculation.

A more automatic controller method is sought, and because of the repetitive nature of the combustion process, the Iterative Learning Control (ILC) has been found to be a very suitable one. ILC is a good solution for a relatively slowly changing environment, like the diesel engine working point at its best normally is.

While the control system with lower calculation load is a demand, a model-based controller (MBC) has been found to be suitable for development purposes of the EHVA control system. MBC with adaptive parameters can be relatively fast acting, and it also has the possibility of open loop control [Linjama 2003b].

7.1 Linear State-space model

State-space model of the system is created to study the main dynamics of the system [Jelali 2004] and to find a starting point for the controller development. Actuator position and velocity and chamber pressure are selected as state variables (Eq. 10). Valve spool position is the input of the model and in this case each state is output (Eq.11). Second order dynamics of the valve spool is modelled as separate state-space model. Corresponding block diagram is shown in Figure 45.

$$\begin{bmatrix} \dot{y}_{act} \\ \dot{v}_{act} \\ \dot{p}_A \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ \frac{-k_s}{m_{vt}} & \frac{-r_{vt}}{m_{vt}} & \frac{A_A}{m_{vt}} \\ 0 & \frac{-A_A B_{eff}}{V_{A0}+A_A y_{act\_e}} & 0 \end{bmatrix} \begin{bmatrix} y_{act} \\ v_{act} \\ p_A \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{B_{eff} K_{vPA} \sqrt{p_P - p_{A\_e}}}{V_{A0}+A_A y_{act\_e}} \end{bmatrix} x_v \qquad \text{(Eq. 10)}$$

$$y = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} y_{act} \\ v_{act} \\ p_A \end{bmatrix} \qquad \text{(Eq.11)}$$

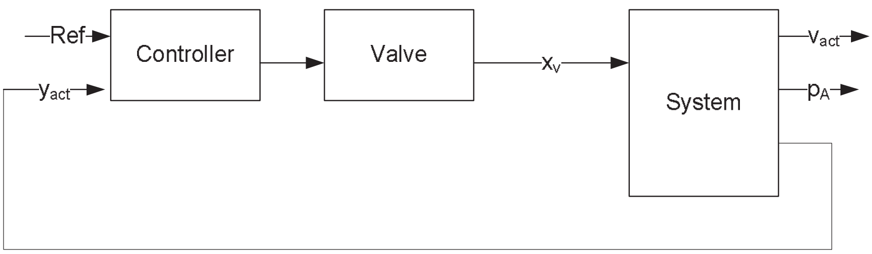


Figure 45 Block diagram of the State-space model

Parameters of the model are:

K_{vPA} = 1.67e-3 m^3/s / √3.5 MPa;

B_{eff} = 1200 MPa;

V_{A0} = 4.7e-5 m^3;

A_A = 7.1e-4 m^2;

A_B =9.1e-5 m^2;

p_P = 25 MPa;

m_{vt} = 2.7 kg;

k_s = 1.2e5 N/m;

r_{vt} =330 N/(m/s);

F_{spre} = 2000 N;

ω_v = 200*2π;

δ_v = 0.5;

The system is linearized in equilibrium, where the piston position and velocity are zero. The chamber pressure p_{A-e} is set such that the sum of forces acting on the actuator is zero. Viscous friction parameter is doubled in comparison to non-linear simulations to compensate for the lack of damping from pQ-relationship of the control valve.

The open loop bode (Figure 46) shows that the system has gain margin of 43.3 dB. Furthermore, the actuator has a natural frequency of 345 Hz, which is significantly higher than the corner frequency of the control valve (200 Hz). Thus the dynamics of the control valve has significant effect on the overall dynamics of the system. The dc-gain is infinite due to the integrator in the system dynamics. Therefore the steady-state error is zero even with simple P-controller. Control valve proper spool offset is however affecting to error.

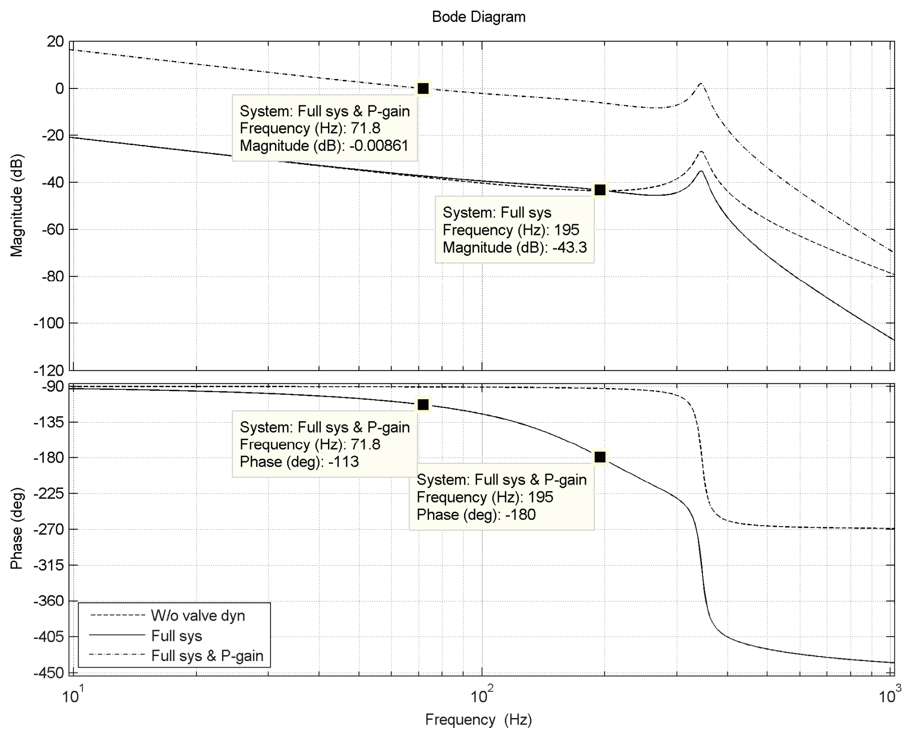


Figure 46: Bode diagram of the actuator (--), the complete system of actuator and valve (-) & the complete system with P-gain (-.-)

P-gain is adjusted to have 6 dB gain margin. The system response multiplied by the P-gain shows phase margin of 67 degrees.

Figure 47 presents the closed-loop bode diagram. The -3 dB point is beyond 200 Hz, but the phase of the system response is reversed already slightly below 200 Hz. The bandwidth of the system is practically limited by the phase delay. Under 100 Hz the gain of the system is very close to 0 dB and the phase delay is less than 75 degrees. Fundamental frequency of the target trajectory curve (at 900 rpm) is 25 Hz. The system is easily capable of full amplitude at that frequency, but the phase delay is already considerable 21.7 degrees. This equals to 2.4 milliseconds, which is not acceptable. Example trajectory is presented in Figure 48. Both linear and non-linear simulation models are giving similar results, only actuator velocity and pressure has some discrepancy. This may be due to more accurate friction modeling and pressure dynamics of the non-linear system, which also can trigger the vibrations in system natural frequency.

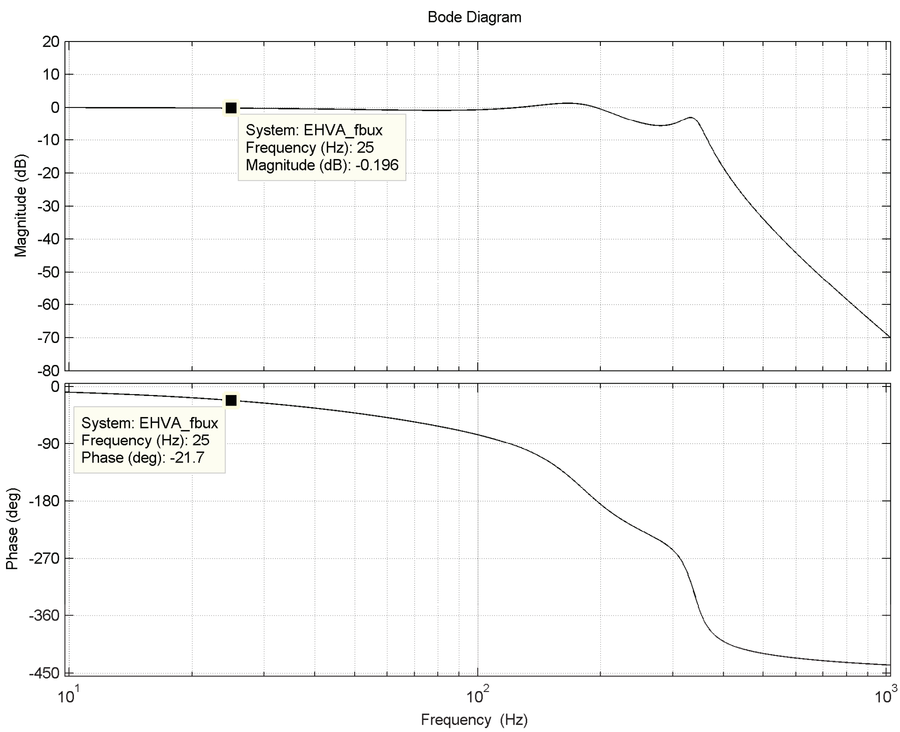


Figure 47: Closed-loop bode diagram.

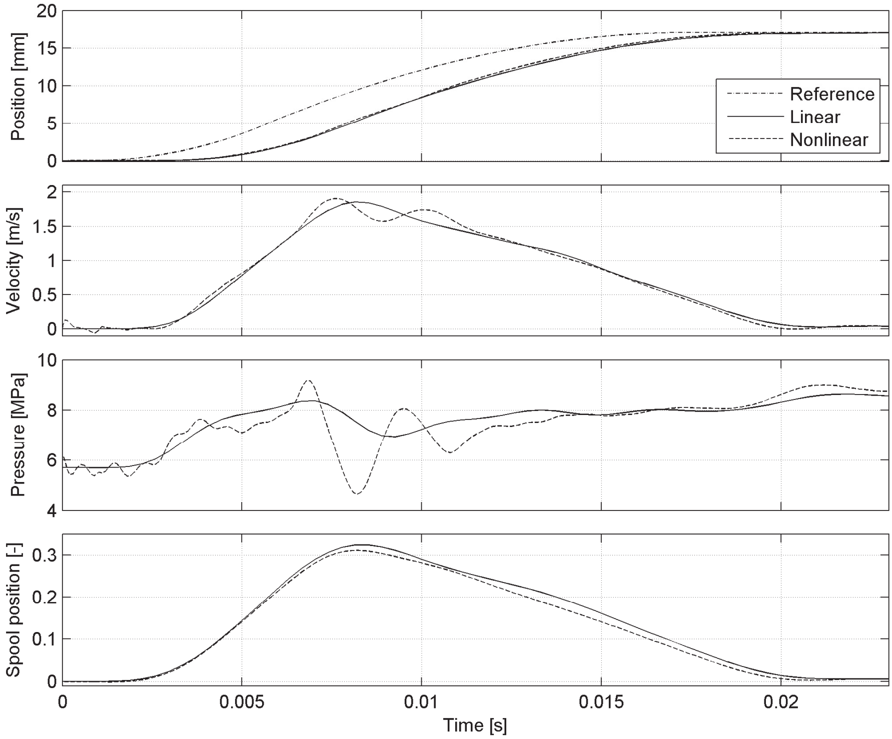


Figure 48: Closed-loop trajectory tracking. P-controller. KP = Kp_{crit} -6 dB

While P-control is giving encouraging results both in linear and non-linear model simulations, experiments of the test rig and especially engine measurements (Figure 43) shows that tracking ability of the controller is insufficient. Difference between simulation model and real system may be unknown non-linearity in control valve. Parameters of the control valve simulation model needed to be compromised between step and frequency responses. This is probably due to control valve internal spool controller. One uncertainty is cylinder gas force effect. Modeling of the pressure behavior is based on one measurement data values only. In addition geometry of the GEV head and thus gas forces effecting to GEV are simplified. Control valve spool offset has effect to the control, and if it is not properly set it may cause problems. Also hydraulic pressure dynamic behavior in multi-cylinder system may disturb the system. Finally, although the system delay can be compensated by shifting the reference, delay still exists and has influence to external disturbance rejection capability.

7.2 State control

In state control, state variables like velocity and acceleration contain all information necessary to predict the future behaviour of the system. Now the use of higher closed loop gains is possible, and this improves the dynamic response and steady state error of the system. Gain values depends on used profile, used average values of the dSPACE system (output divided by ten) are for example KP=650, KV=0.08, KA=0.002. Return gain factor is about 1.2 .

The schematic block diagram of the used state controller is shown in Figure 49. In addition, the controller needs feedforward loops, direction dependent gains and also position dependent gains in order to decrease the tracking error inside the required range through the valve lift [Herranen 2009].

The valve trajectory curve fed to the controller is generated by the look-up table, where the valve lift is as a function of the crank angle. Valve lift array has 360 elements so valve lift is defined at 2 CA increments. Look-up table is interpolating data points between given data points, and crank angle sensor/input has 10-bit resolution so with 0.2ms time step the controller is having a fresh data at each step if the RPM is greater than 293min^{-1}. In the feedforward loops, the reference velocity and the reference acceleration signal are calculated offline from the valve lift profiles. Feedforward thus cannot cause the control system to oscillate, and system response and stability are improved. The feedforward control could also eliminate the influence of the measuring inaccuracies and disturbances, which are common in state controlled systems. [Jelali 2004], [Cooper 2005]. In addition, the lift of the gas exchange valve is divided in sections (for example 0-4mm, 4-15mm, 15-17mm). Each of three partitions has individual feedforward gains, for example Ref KV=[1, 1.2, 1.2], Ref KA=[0.001, 0.002, 0.002]. Thus the tracking error could be controlled more accurately during the whole lift.

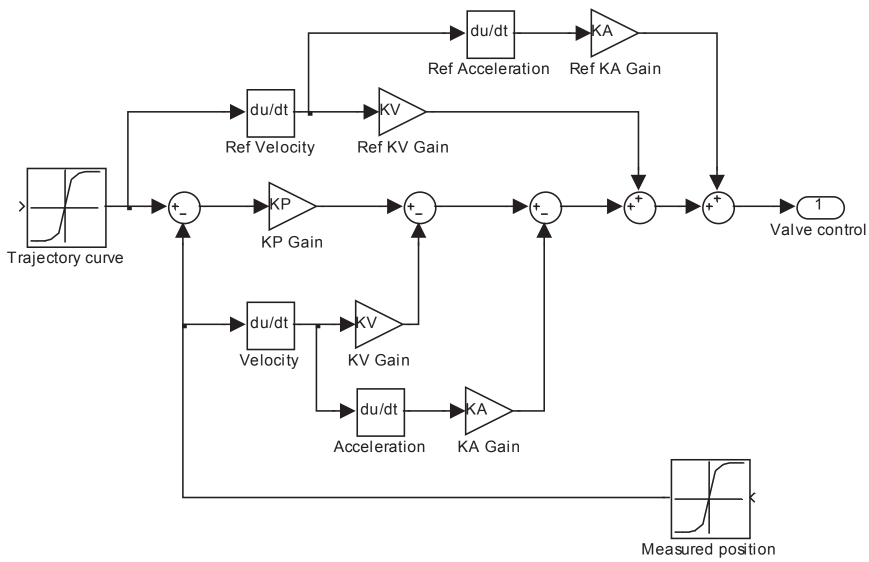


Figure 49 Schematic block diagram of the State Controller [Herranen2009]

7.3 Iterative Learning Control

The very first ILC system designed for robotic application is published by [Arimoto 1984]. ILC controllers have been presented in electro-hydraulic GEV systems [Heinzen 2011], [Wu 2012], and closely similar controllers like repetitive controllers (RC) are also used in gas exchange valve actuation systems [Sun 2009], [Liao 2008].

In the ILC tracking control method, the control system learns from previous repeated control trials whether they succeeded or failed. An observed error in a previous repetition is used to adjust the reference in the current run for the reference to produce the desired trajectory (target reference). This is realized through memory-based learning. The simplest ILC scheme is that in Equation 12.

$$u_{i+1}(t) = u_i(t) + q\Delta y_i(t) \qquad \text{(Eq.12)}$$

The tracking error is thus measured at every data point and saved in the memory as a curve resembling the valve lift curve. In the next repetitive round, the saved error of the previous round is added to the previously used reference valve lift curve, and the modified curve is also saved. When the process continues, the valve lift curve gradually transforms to a shape resulting in the minimum system tracking error. In this study, only the P-controller has been used because of its simplicity, whereas other controller types are also useful with ILC, if the P-controller is not effective enough [Herranen 2009]. Value of the learning gain q has been experimentally tuned. With high gain strong and fast changes are caused in reference curve, and temporary disturbances have also stronger effect. The learning process may be stopped because system detects degrade in tracking error. It has been defined that reasonable response level of the GEV lift learning is few seconds in diesel engine operation and normally gain value q = 0.02 provides that. KP gain can be between 550-750, when return stroke gain has factor of 0.8-1.4 .

Characteristically, ILC is such that at some point the tracking error stops decreasing and starts growing. During the valve lift event, the error cannot drop to zero due the system dynamics, but the learning process continues to modify the reference curve. At some point, the system can no longer properly follow the increasing reference value, and the error starts increasing. The problem can be avoided in several ways, such as using a dynamic compensator instead of a learning gain and zero-phase low-pass filter cutoff [Longman 2006], or using a model-type ILC where the cost function can handle unwanted changes of parameters [Dorner 2012]. Good results are also achieved if the valve event is divided into phases (lift, dwell, close, dwell), and if learning of each phase is handled individually [Longman 2006]. This procedure would also allow individual delay compensation for each phase. The schematic block diagram of the used ILC method is shown in Figure 50.

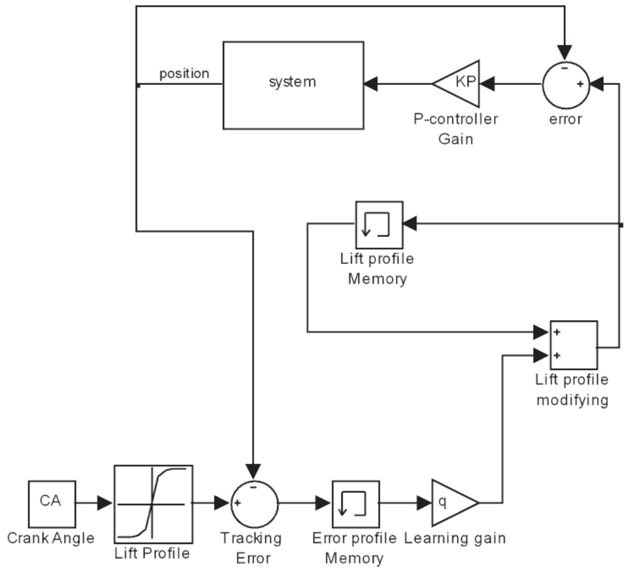


Figure 50 Schematic block diagram of Iterative Learning Control [Herranen2010b]

7.4 Model-Based Control with adaptive parameters

The Model-Based Predictive type controller was invented in the 1960s, and developed further in the 1970s and 1980s. In gas exchange valve applications, MBC type controllers have been used for example in [Hathout 2004]. Adaptive control systems have been used earlier in flexible valvetrains for example in [Hanks 2005], [Tai 2000]. The MBC algorithm consists of a prediction model, cost functions, and search space selection. Future values of the system are predicted by calculated outputs of the model. Because of the mathematical model of the system, MBC is relatively robust to interferences and changing process parameters.

The MBC used in this study is based on the mathematical model of the control valve. The basic control structure of the system is the sum of P-control and feedforward control loops. Based on this, the velocity request of the MBC was created. The selection of control parameters was handled by an adaptation mechanism, which chooses parameters based on reference signal. The MBC calculates different flow rates of the valve when the opening of the valve is changed. The calculation is based on the equation presented in Equation 13.

$$Q_{valve} = uK_{vPA}\sqrt{\Delta p_{valve}} \quad \text{(Eq.13)}$$

The valve opening is a user defined vector, which forms the search space of the controller. Basically it consists of discrete steps between 0 and full valve opening (1). The number of steps can be freely defined, and it affects the flow resolution of the control valve (Figure 51).

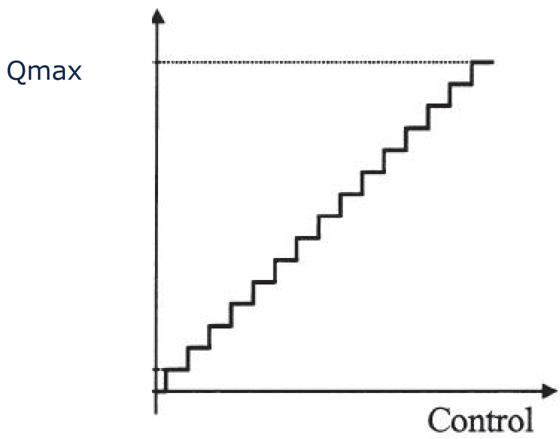


Figure 51 Discretized flow states of the control valve

The calculation time of the controller is dependent on the number of steps. The cost function gets a set of possible flow rates, which are converted into piston velocities. The cost function compares these values with the velocity request and choses the closest opening state. The cost function can include several penalty definitions like deviations in desired chamber pressures, cross-flow (energy loss) through the valves, and change of the valve state. All penalty factors could be weighted with different ratios. Now the only used penalty is velocity error, because only one spool valve is used and each flow states have only one possible opening. Valve state change is usually needed to punish when several, approx. more than five on/off valves per control edge are used or flow state is possible to achieve with several different opening combinations. Chamber pressure control is important if pressure and tank control valves are opened at the same time (cross-flow). With simple system like this, velocity of the actuator provides accurate enough tracking error.

The schematic block diagram of the MBC is shown in Figure 52, which also shows the adaptive control part.

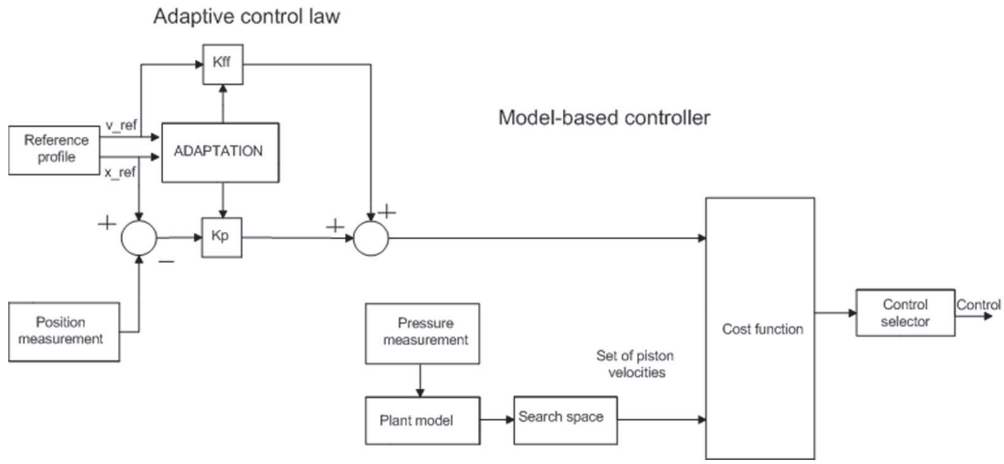


Figure 52 Schematic block diagram of the MBC control [Herranen2012]

Gain values of the P-control and feedforward loop are adapted according to the reference profile. P-gain and Feedforward gains are defined in three sections. Example of controller parameters is shown in Table 4.

Table 4 Adaptive values of MBC parameters

GEV lift [m]	RPM [min^{-1}]	Kp open	Kp close	Kff open	Kff close	Start velocity [m/s]
0.017	820	130	95	0.86	1.2	1.2
0.0125	820	135	95	0.86	1.15	0.9
0.0085	820	125	95	0.86	1.1	0.7
0.017	400	130	95	0.86	1.1	0.4
0.0125	400	135	95	0.86	1	0.4
0.0085	400	125	95	0.86	1	0.36

8 COMPARISON OF ALTERNATIVE CONTROL CONCEPTS BY MEANS OF SIMULATIONS AND MEASUREMENTS

The controllers introduced in the previous chapter, have been tested first by means of simulations. Then test rig measurements were carried out.

8.1 Simulation results

The investigation is started with a P-controller. Delay of the system is compensated in results afterwards by shifting the timelines of the curves. As a result of the simulations, it can be discovered that the closing part of the valve stroke is problematic (Figure 53), as seen in Chapter 6, but in simulations the difference is more clear. The start of the closing is delayed heavily and the tracking is very poor at the end of the closing. When direction dependent gain is added to the system, also the closing part is better under control (Figure 54).

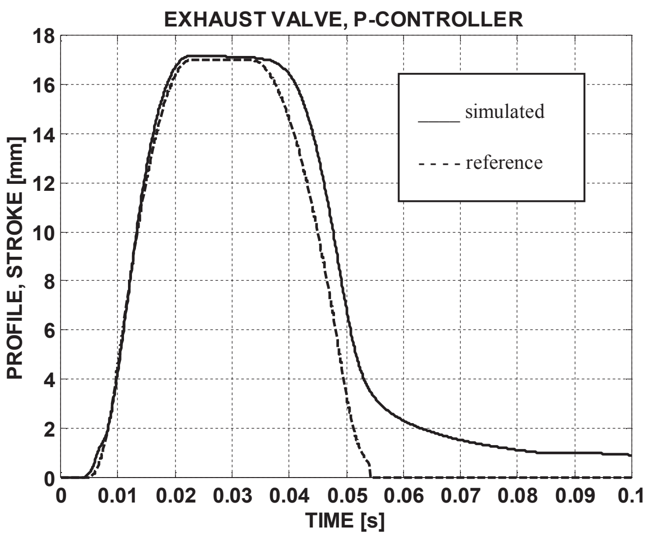


Figure 53 Simulated P-controlled actuator stroke, symmetric gain [Herranen2009]

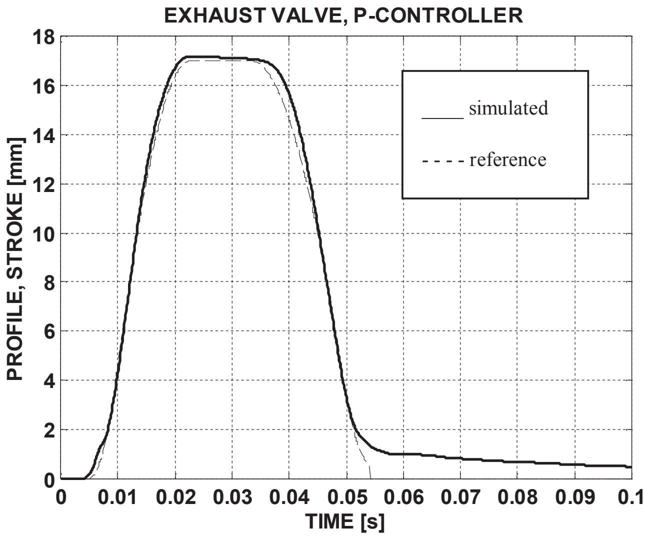


Figure 54 Simulated P-controlled actuator stroke, asymmetric gain [Herranen 2009]

The difference between the P- and State Feedback Controller (Figure 55) is small according to the results. Some minor improvement can be seen in the beginning of the opening as well as closing.

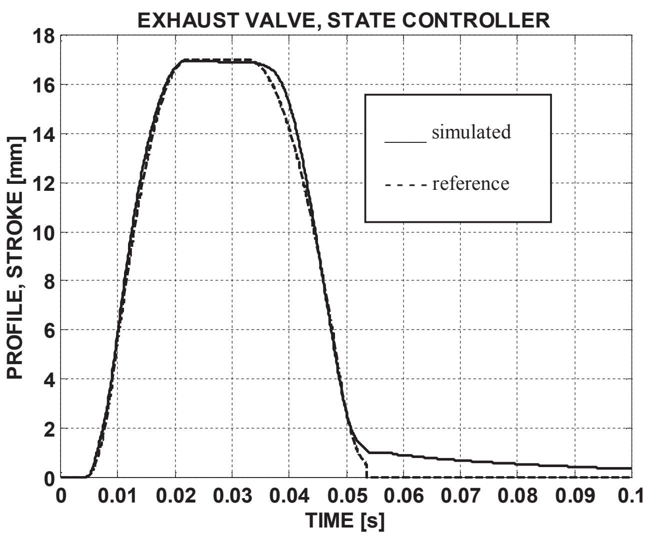


Figure 55 Simulated State controlled actuator stroke [Herranen 2009]

One reason is that the capacity of the state controller is limited by the dynamics of the hydraulic control valve. This is due to the much higher nominal frequency of the hydraulic system (ca. 1800 rad/s with the full opening of the gas exchange valve, in later versions even up to 2500 rad/s, Eq.6) than the nominal frequency of the control valve (ca. 700 rad/s with a 50% input signal). It is recommended that the control valve should be at least three times faster than the system in order to achieve full benefit of the State Feedback controller and feedforward loops [Virvalo 2008].

In these simulations there is a 1 mm backlash modelled between the gas exchange valve and the inner end position of the actuator. The clearance is designed to compensate for the temperature expansion of the valve lift system. The effect of the backlash can be seen as a very slow closing of the hydraulic actuator in the very end of the stroke. The springs of the gas exchange valves are not helping any more the movement of the actuator in the clearance range, and the movement of the actuator becomes slower caused by the sudden change of the force balance. This also means that the gas exchange valve closes about 1 mm before the actuator reaches its end position. The backlash is neglected in later simulations.

Next, the ILC simulations takes place. The fully learned profile and resulted lift curve are shown in Figure 56. The modified shape of the reference is clearly seen, also the most difficult spots in the both deceleration places which are mostly modified. Very good matching between the actuator movement and the target is found. Asymmetric gain is still used in the P-control loop of the ILC.

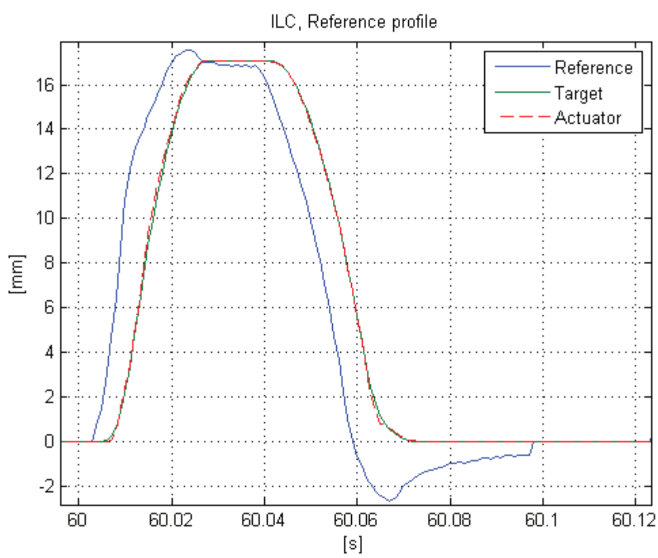


Figure 56 Simulated lift curves after ILC finished [Herranen 2010b]

Figure 57 shows a situation where learning is not stopped and the system has become unstable. This happens, as said earlier, due to points where the error cannot be reduced. Learning continues to modify the reference until it results in either an under- or overshoot after the problematic point. This error must be corrected, but the final result is the situation where the reference oscillates heavily and also the real lift curve is in a similar state. This shows the importance of the learning process control.

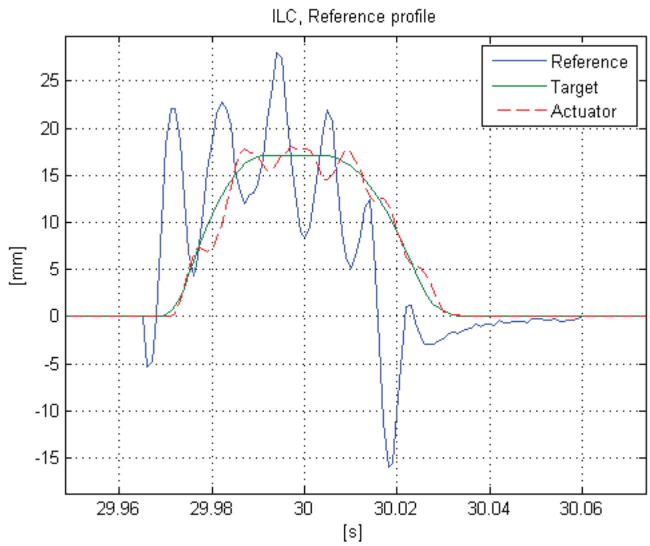


Figure 57 Simulated lift curves, over learning [Herranen 2010b]

Simultaneously with the ILC controller, an individual controller is developed for the delay compensator. The difference between the target and measured position is calculated in one certain lift point. After that the shift of the CA is made in the GEV lift data, if necessary. The delay compensating is made separately from the ILC process, so that it would not disturb the learning process by shifting and confusing data values in the learning. When the initial delay is compensated, the learning process is started. If the delay increases out of the range during the learning process, the delay is again controlled during a stroke and learning is set off. Figure 58 shows the effect of the delay compensation and learning process on the tracking error of the actuator. First, the system compensates the delay during the first 2.5 seconds. The ILC is started after 6 seconds of running. After this, the ILC and delay controller take turns (they are not simultaneously on). Finally the required error range is reached after about 8 seconds. In real application, waiting time after delay is compensated and before learning begins is not needed.

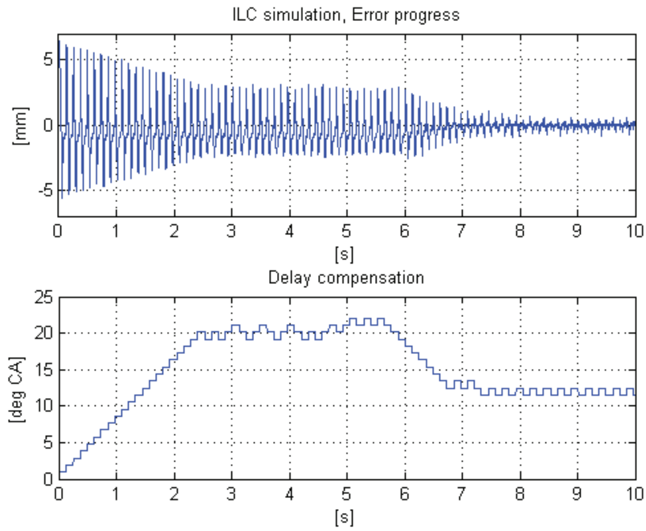


Figure 58 Delay compensation and effect on error [Herranen 2010b]

In Model-Based Controller simulations, first the control valve resolution is defined. Figure 59 shows the controlled displacement of the actuator, when the resolution of the discrete step is 2%, 4% and 10% of the full valve opening. It can be seen that with 2% resolution the accuracy is good enough, and further tests also showed that significant improvement is not achieved if smaller resolution is used. Thus the chosen 2% value equals 50 discrete opening states of the control valve.

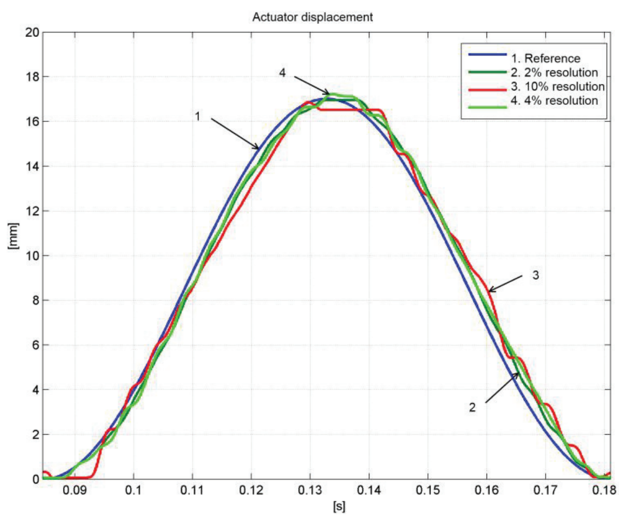


Figure 59 Effect of the resolution on the tracking. Edited [Enlund 2011]

The valve state selection of the MBC between the simulated and measured system is shown in Figure 60. It can be seen that the simulation matches quite well to the measurements, and also that the system uses only half of the valve states available. Thus the system is capable of faster movements than used in this experimental simulation.

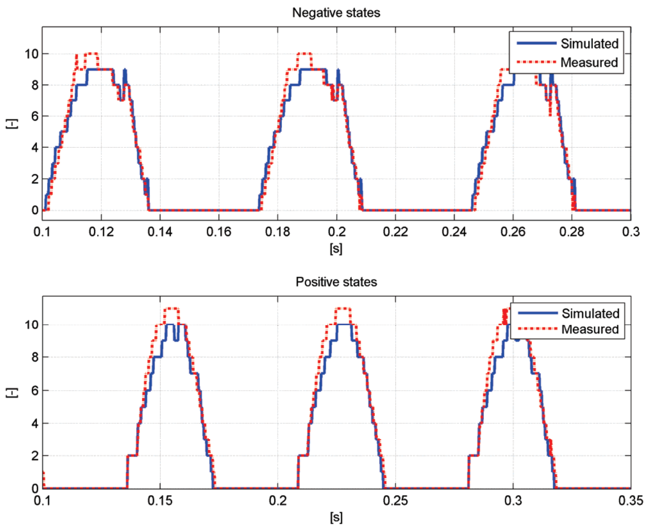


Figure 60 Comparison of the MBC control signals [Herranen 2012]

One problem with the MBC system is that in the beginning of the stroke, opening is delayed due to the simulated cylinder load (Figure 61). The MBC controller does not allow the state of the control valve to change before the step is reached, and this causes delayed opening. The solution is to add the “boost control” in the beginning of the stroke. In boost control, start velocity of the reference signal (5^{th} element of velocity vector), is detected. In the beginning of the GEV lift, start velocity value forces a control valve state higher than the initial calculation would require, resulting faster opening of the GEV (Figure 62). Boosting is then one parameter of the adaptive control.

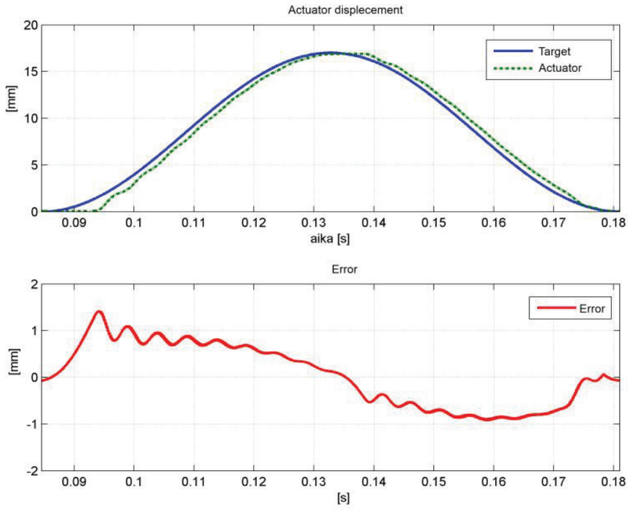


Figure 61 Actuator displacement without start boost control. Edited [Enlund 2011]

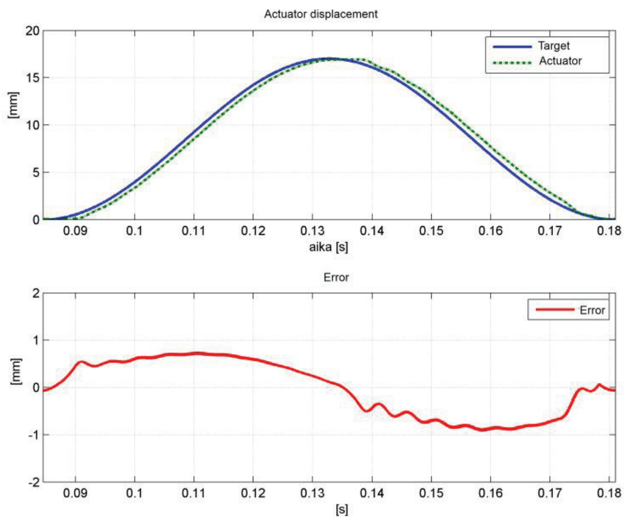


Figure 62 Actuator displacement with start boost control. Edited [Enlund 2011]

Next, the delay compensation is applied separately to the rising and falling part of the valve lift. The result is already very good in the current simulated working point (Figure 63).

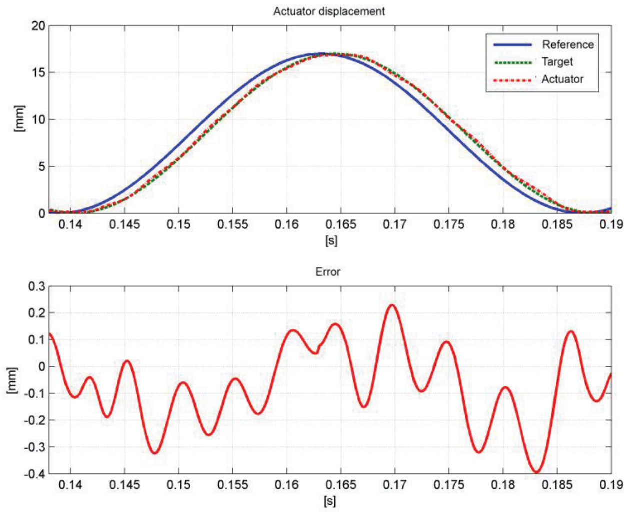


Figure 63 Actuator displacement with separate open and close delay compensation. Edited [Enlund 2011]

The adaptive control system changes six P-control and six Feedforward control parameters like shown in Table 4 according to the reference lift, start velocity, and rpm. This leads to good sensitivity against these environmental variables, for example to lift frequency (Figure 64).

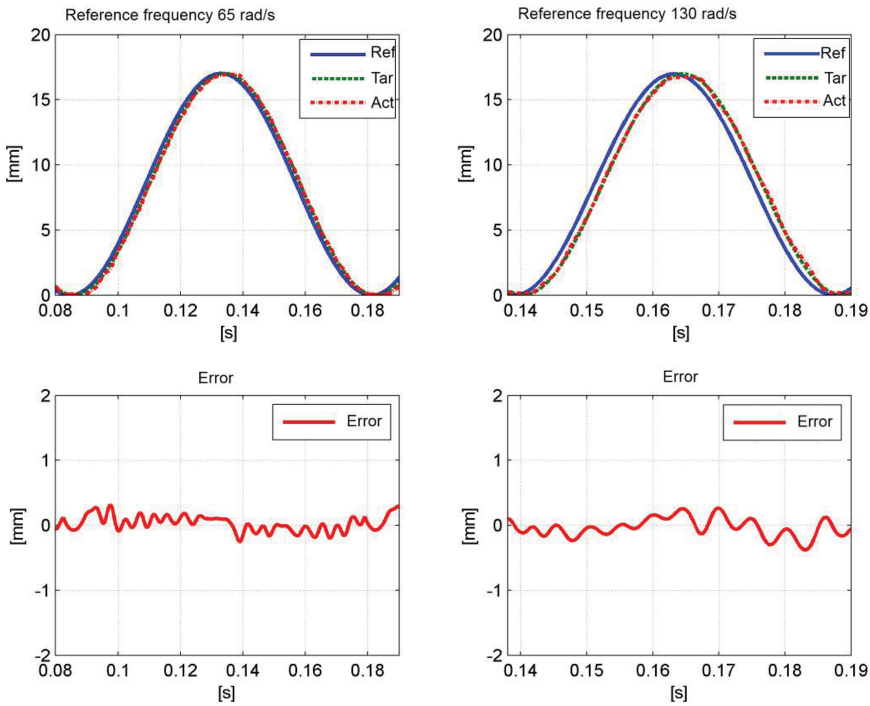


Figure 64 Actuator displacement sensitivity to lift frequency, adaptive control. Edited [Enlund 2011]

The MBC system is then tested with different loads, hydraulic supply pressures, and hydraulic dead volumes. Dead volume has been found to be the most sensitive parameter, while the system is insensitive to other variables (Figure 65, Figure 66, Figure 67).

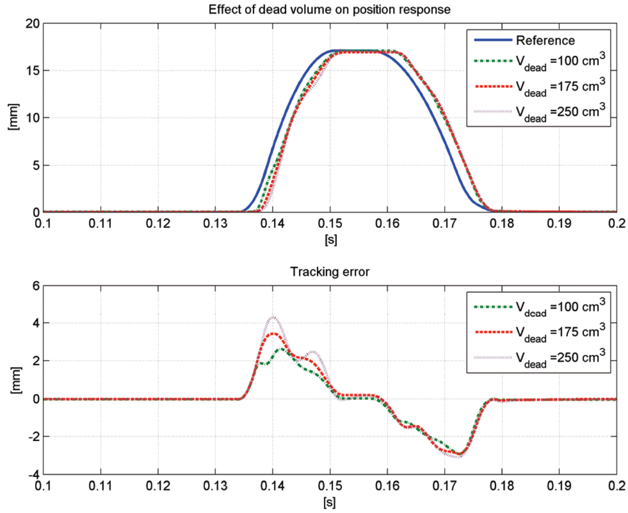


Figure 65 Actuator displacement sensitivity to dead volume between actuator and control valve [Herranen 2012]

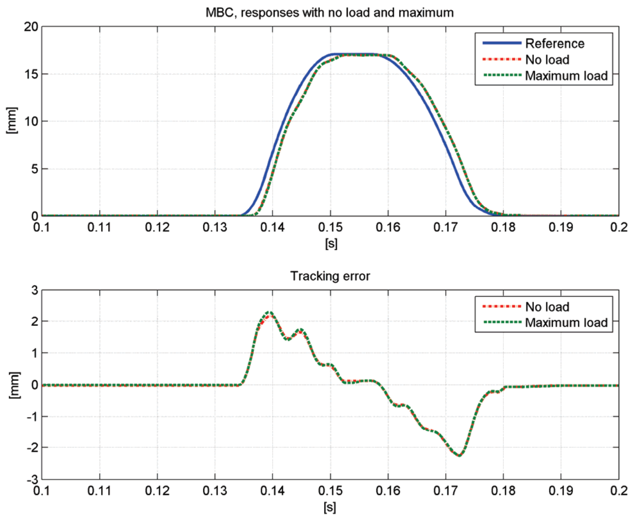


Figure 66 Actuator displacement sensitivity to external load

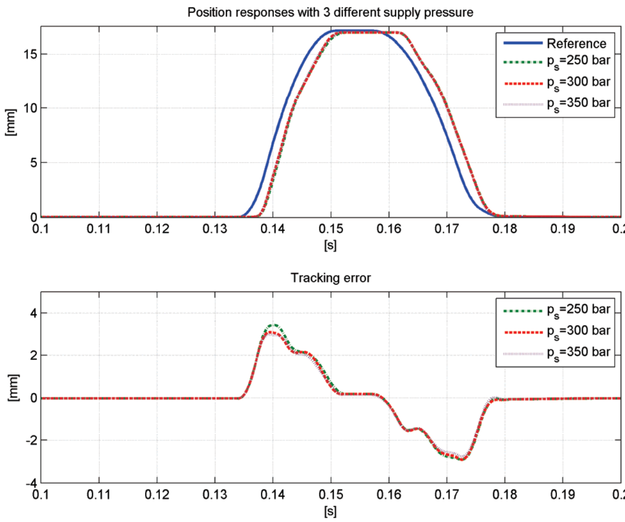


Figure 67 Displacement sensitivity to supply pressure

8.2 Performance of different EHVA systems, test rig measurements

The different EHVA control concepts are experimentally evaluated with test rig measurements. The investigated properties and demands of the EHVA are introduced in Chapter 4.1. In the beginning of the tests, the focus was on the tracking capability or in other words on tracking error minimization. Later on, the practically important characteristics (e.g. usability) and performance values (e.g. GEV timing repeatability) are measured or evaluated.

8.2.1 State Control

For the P-controller, the beginning of the stroke, end of the fully open position, and the seating velocities of the GEV have been found to be problematic, especially if the load conditions change during the stroke like in driving the engine at full load (Figure 43).There is clear improvement in these situations when the state controller is used (Figure 68) and the required error range can be reached.

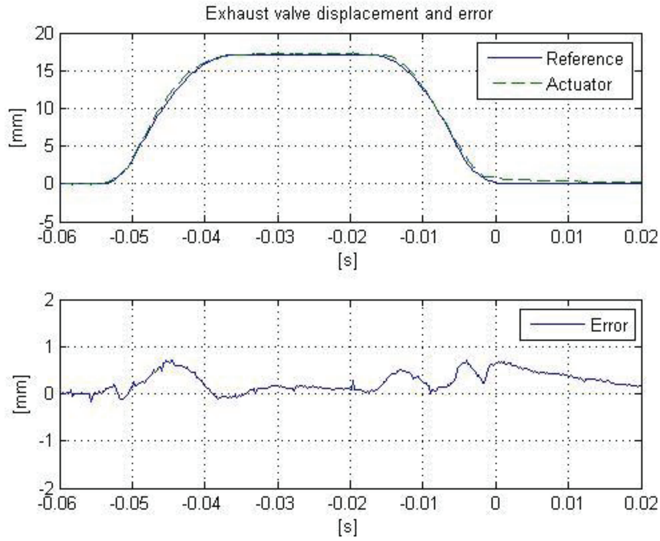


Figure 68 Tracking error of state controller [Herranen 2009]

Because the state controller gains need many different values (total number of gain parameters is up to 11 per actuator), it is very time consuming to tune up the parameters in each working point. Adaptive parameters would help the tuning, but because individual gains also have cross-effects on each other, this was very hard to apply.

Because the velocity and acceleration are calculated from the measured position signal, the quality of the calculated signal is poor, which is also stated as a problem with PID derivative term. Especially the acceleration control loop is very gain sensitive due to disturbances in the calculated acceleration signal, and it is basically useless if used without any signal conditioning. Filtering, state observer/estimators or quantization avoidance may be required for better performance of this part of the controller.

8.2.2 Iterative Learning Control

In the ILC, the number of tuning parameters can be decreased down to 3 per actuator (if asymmetric P-control in use). Of course the divergence problem caused by the continuous learning process needed a few parameters, but these can be pre-tuned and they rarely need to be re-adjusted. The initial, delay compensated situation is shown in Figure 69. The saw-tooth shape error curve is due to discrete definition of the target reference in the current control system when interpolation of the data points was not in use.

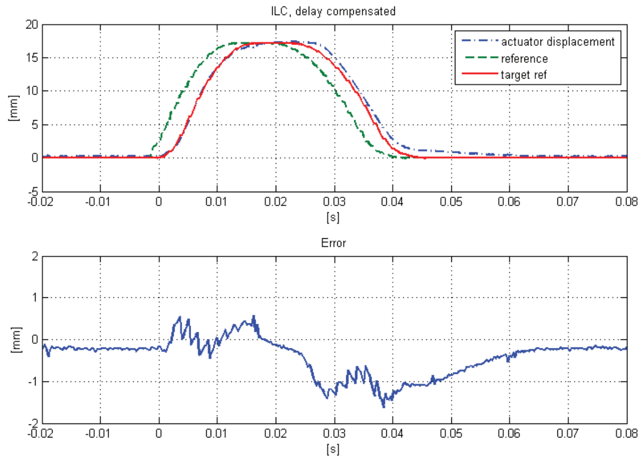


Figure 69 Tracking error of the ILC system, delay compensated [Herranen 2010b]

Figure 70 shows the tracking result when ILC learning has been on for 60 seconds. The reference curve shape is clearly re-shaped due to learning, and it results in an error range well inside ±0.5mm. In normal conditions with tuned parameters, error correction by ILC takes about 30-60 engine stroke cycles.

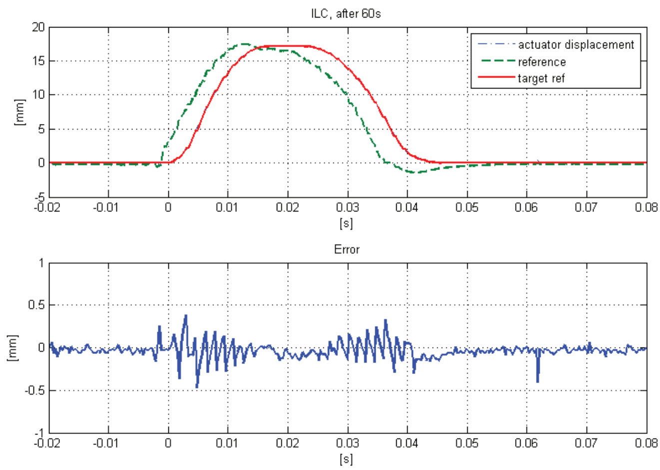


Figure 70 Tracking error of the ILC system, learning completed [Herranen 2010b]

Control of the learning process can be done in many ways, as stated in Chapter 7.2., but two very simple methods are used here together. First, the learning process is simply stopped when the error of the current data point is within a predefined tolerance. If the measured error is outside the error range, learning of that point starts again. Second, the error sum of all data points is calculated. The system records the smallest error sum gained, and if the sum grows larger than a certain percent of the minimum value, it shows signs of failed learning and the process can be stopped. If the error sum still continues to grow, the whole learning process is reset to the initial state to prevent stability problems.

Response of the system to changes is fast, and everything is automatic once the learning limits have been defined. ILC has problems with the controller time step because the structure of the reference is so that the data point must be measured and updated at same increment that it is defined in data array. Reference data definition step of 2 CA leads to the required step time being shorter than 0.35 ms at a nominal engine speed of 900 RPM. Otherwise every reference data points get no learning correction at all.

8.2.3 Model-Based Control

In model based control measurements the main goal is to find out if the MBC system could provide sufficient performance with shorter calculation time than ILC. In measurements the malfunction of the displacement sensors disturbed the performance verifications (Figure 71), but the controller calculation throughput was somehow possible to define.

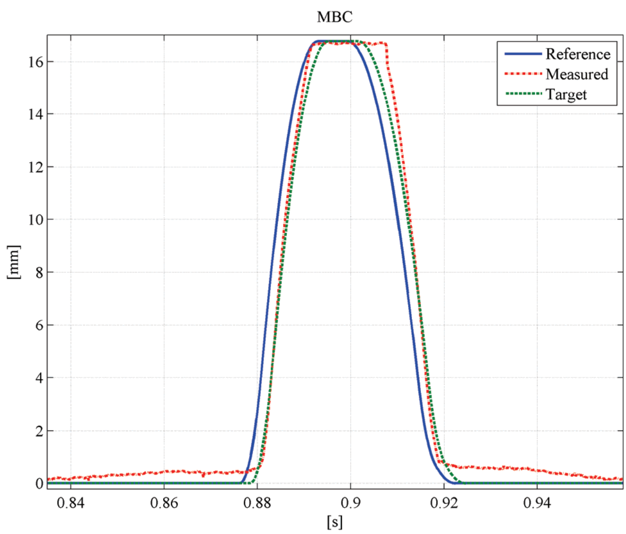


Figure 71 Tracking error of MBC

The repetition of the valve timing (as defined in chapter 4.1.3) by using the MBC is about 0.6 ms. This equals about 3 CA degrees, which is at the same level as the ILC repetition range showed before. Minimum controller calculation step time of the MBC is 0.04ms, and the time appeared to be constant during the measurements.

A negative aspect of the used adaptive model-based controller is that the number of parameters is also relatively high, and the adaptive varying range and behavior must be tuned manually.

8.3 Comparison of controllers

The controller development process has been a chronological process, and each tested controller has pros and cons. The most important features of each controller are summarized and evaluated in Table 5.

Table 5 Evaluation of controllers

	P-control	State	ILC	MBC
Number of parameters / actuator	2	11	3 + (3)	6 + (8)
Sensitivity to environmental variables	Poor	Poor	Excellent	Good
Tracking error	Poor	Good	Excellent	Good
Complexity of the controller	Good	Poor	Poor	Average
Required computation power	Good	Average	Poor	Average
User interface / Usability	Excellent	Poor	Good	Good
Repeatability	Average	Good	Good	Good

There is some variation in the numbers of tunable parameters because ILC and MBC require some predefined parameters for the learning and adaptive control selection limits. However, once tuned these parameters can remain mostly constant.

ILC is superior when environmental parameters change, while MBC and state controllers can adapt to some changes. ILC is the most complex controller and it causes the highest need of calculation power. Manual parameter tuning of the state controller makes it difficult and slow to use.

8.4 Optimization of EHVA energy consumption

Like any auxiliary component of the combustion engine, power consumption of the EHVA has an important role because all energy required by the EHVA reduces the diesel engine output power. Thus the energy balance of the EHVA and different improvements are tested by means of simulations.

Some of the energy based simulations have already been done in concept simulations, but that simulation model is very coarse, and gives just suggestive information about the differences between hydraulic circuits and mechanical construction of the actuator. More accurate simulations are done now when the whole EHVA system has been designed. First, the energy consumption of the different actuator actions are simulated, as well as losses of the EHVA hydraulic components. The energy effect between the current 3-way system and the 4-way system without the return spring is found out.

The hydraulic energy taken during the stroke event is calculated by integrating the product of the hydraulic pressure and flow rate of the actuator chamber. The energy is calculated separately in the open (Equation 14) and close (Equation15) direction. This is the amount of energy which is eventually needed to produce the stroke of the electro-hydraulic gas exchange valve system. Total energy taken from pump is calculated in Equation 16.

$$E_{hyd\ open} = \int_0^{t_{open}} p_a \times Q_+ \, dt \qquad \text{(Eq.14)}$$

$$E_{hyd\ close} = \int_{t_{open}}^{t_{close}} p_b \times Q_- \, dt \qquad \text{(Eq. 15)}$$

$$E_{hyd\ total} = \int_0^{t_{close}} p_{pump} \times Q_{pump} \, dt \qquad \text{(Eq.16)}$$

In the system with the return spring, the energy stored in the spring produces the return stroke, and therefore any extra closing energy is futile. However, the actuator has to push out the hydraulic fluid from above of the actuator and this needs some energy. The energy of the opening and closing events include the energy needed to overtake energy losses in returning hydraulic lines and components. Without the return spring, the actuator has to draw the whole gas exchange valve system to the upper position, and therefore the energy consumption is much higher. The needed energy levels are presented in Table 6. The difference in overall energy rate is explained by a higher average flow rate of the non-return spring system.

Table 6 Energy rates of the systems

Energy	3-Way + Return spring	4-way No Spring
Opening (hydraulic)	179 J	132 J
Closing (hydraulic)	28 J	116 J
Pressure relief valve	2 J	12 J
Energy from the pump	309 J	410 J
System efficiency	67%	60%

The hydraulic control valve has a significant effect on the system efficiency. Table 7 shows the relative energy losses over the control valve compared to the energy produced by the pump during the event. The 4-way system has naturally higher losses because flow through all control edges of the control valve is needed, and furthermore input and output flow cannot be controlled individually but via the same spool opening.

Table 7 Energy loss in the control valve

	Return spring	No spring
P-->A	35%	21%
A-->T	23%	27%
B-->T	-	8%
P-->B	-	10%
Total	58%	66%

The length of the hydraulic lines between the components produces some energy loss due to friction losses. In the simulations, the basic pipeline length between the control valve and the actuator is 0.2 m and diameter 10 mm. Then the different l/d ratios are simulated and energy loss during the stroke is calculated from pipe flow and pressure drop over the pipe. Figure 72 presents the energy consumption of the line with different l/d ratios.

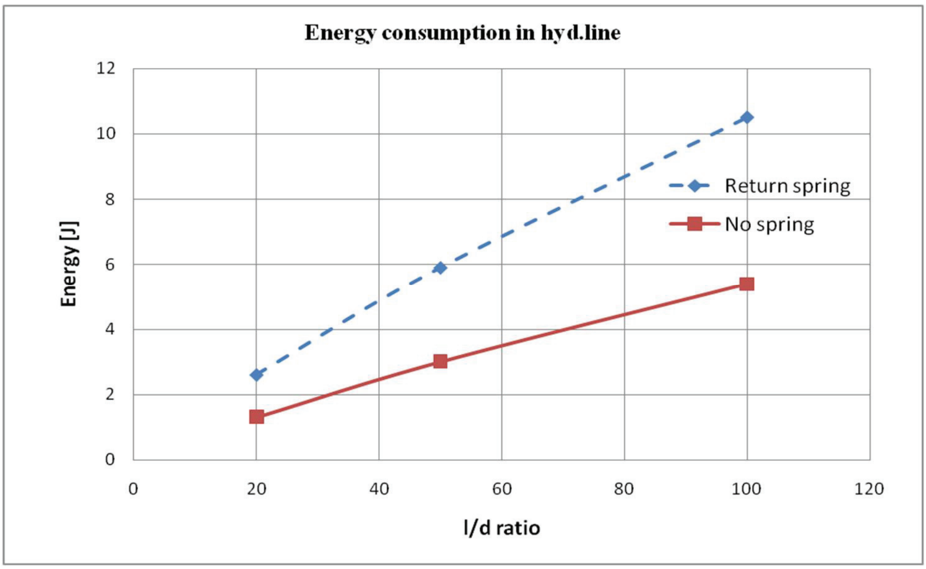


Figure 72 Energy consumption in the hydraulic line (control valve ↔ actuator) [Herranen 2010a]

The energy consumption is increased linearly when the l/d-ratio increases. Single line energy consumption is under 1% of the total energy consumed. Total energy consumption of all pipe lines can be assumed to be around 7-8% in both investigated hydraulic circuits.

Next, the energy recovery is investigated. The nominal energy required to move the actuator is defined by calculating first product of the force produced by the actuator upper chamber and the velocity of the actuator. Calculation is done only at the moment when the velocity is increasing and acceleration is positive in the moving direction (Figure 73). Then the energy is calculated when momentary product is integrated as shown in Equations 17. E_{open} is the amount of energy needed to make the open stroke. Close direction movement energy E_{close} is calculated respectively from lower chamber. On the contrary, the deceleration part of the stroke can be assumed to be the energy which could be recovered ($E_{rec\ open}$, $E_{rec\ close}$). In this way the estimation of the maximum recovered energy is better brought out, because some energy is still fed or stored to the system. Because of the frictions, the actuator will need some energy during the movement, but this matter is ignored in this study. The energy needed to compress the return springs during the deceleration is subtracted from the opening recovery part because it can be used in closing direction recovery. The maximum recovered energies during the stroke are presented in Table 8.

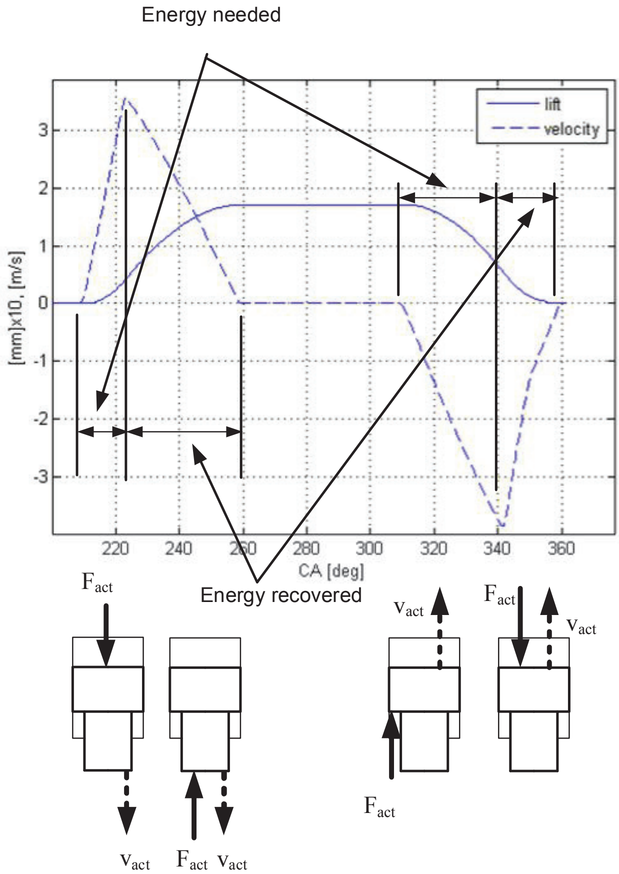


Figure 73 Lift event moments of energy need and recovery

$$\left.\begin{aligned}
E_{open} &= \int_{0}^{t_{open}} F_{act} \times v_{act}\, dt \;, when\; a_{act} > 0 \\
E_{close} &= \int_{t_{open}}^{t_{close}} F_{act} \times v_{act}\, dt \;, when\; a_{act} < 0 \\
E_{rec\ open} &= \int_{0}^{t_{open}} (F_{act} - F_k) \times v_{act} dt \;, when\; a_{act} < 0 \\
E_{rec\ close} &= \int_{t_{open}}^{t_{close}} F_{act} \times v_{act}\, dt \;, when\; a_{act} > 0
\end{aligned}\right\} \qquad \text{(Eq.17)}$$

Table 8 Energy in the acceleration and deceleration phase of the actuator

Energy	Return spring	No spring
Opening	118 J	85 J
Closing	36 J	49 J
Recovery in opening	9 J	45 J
Recovery in closing	47 J	71 J

Next, the energy consumption is investigated as a function of engine load (cylinder pressure). Three different load points are simulated. Then, the hydraulic working pressure and/or actuator pressurized area is changed in order to find out some effects of the fluid compressibility or flow frictions.

Figure 74 shows the relative energy consumption of the 3-way actuator, when exhaust valves only are taken into account. The built EHVA system is a constant pressure system, and it is designed according to the maximum cylinder load, but now in the simulations the system pressure is optimized to correspond to the load points. The area of the actuator is increased by 36% and the pressure level is optimized again. Then, raising the working pressure by 40% and 68% and optimizing the actuator area in all the investigated load points are simulated. The results show if the working pressure is raised and the diameter of the actuator is correspondingly decreased, or the actuator area is increased and the pressure is adjusted, similar efficiency of the system is achieved. Thus the required flow is smaller with higher pressures (and flow resistances lower), but evidently the effect of the fluid compressibility counter that. So there is no big difference which kind of "optimization" method is used; energy needs from the pump (and from the engine) are close to each other in each load condition compared to initial system optimizations that can be done in lower load points.

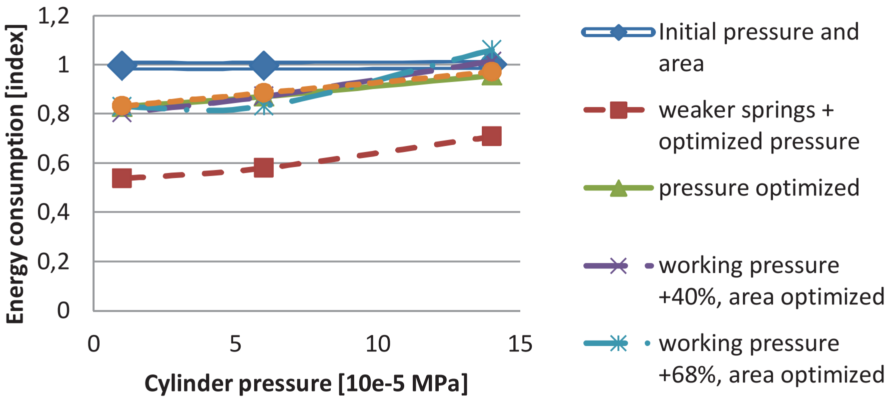


Figure 74 Relative hydraulic energy required by exhaust actuator as a function of cylinder load

In these simulation cases, the used return spring is stiffer (3kN pre-tension, 100N/mm spring constant). Next the pre-tension and spring constant is decreased back to the standard level, where the maximum force is about 40% of the previously used one. This 'weaker spring' case is then simulated with optimized pressure levels. Now it is clearly seen that energy consumption is much lower in all load points. Stiffness of the return spring could be dropped further to the point where the controlled return stroke of the GEV can be barely made. This would give an approx. 40% relative energy need of the actuator energy compared to the initial value (result not shown in the Figure), but this alternative is not investigated further because keeping the GEV closed should be ensured otherwise if a sufficient spring force is not present.

Both actuators are included in the next simulations. In valvetrain optimization, first the diameters of the actuators are decreased to a minimum possible. Because the exhaust actuator had to overcome the largest force, the actuator area is already optimized in the initialization design process when the working pressure is set to 24 MPa. The intake actuator diameter could be decreased by about 10 % due to the lack of cylinder pressure at the moment of the intake valve opening.

The optimized intake area gives about a 10% unit improvement in total energy consumption. Working pressure adjustment gives only minor improvement in the middle load point, but pressure must be maintained at the same level in the lower load point too because the force of the return spring increases toward the fully open position and requires enough hydraulic pressure force.

Due to the nature of the cylinder pressure force, the highest force is needed in the beginning of the exhaust opening. When the exhaust valve opens, the pressure difference over the gas exchange valve rapidly drops, and the required force will be smaller. If the exhaust actuator had a two-staged pressurized area, the hydraulic pressure force could be changed. After 5mm GEV lift the cylinder pressure is dropped, a smaller actuator effective area is activated (Figure 75), and flow to the actuator is thus decreased during the end of the stroke (area of the lower chamber remains unchanged). This 2-staged decreasing area can be utilized only to the exhaust actuator, because the force variation during the stroke is more or less opposite (due to the spring force) of the intake valve. The 2-stage method gives an approximately 20% decrease of energy consumption, but it is valid through the whole cylinder load range (Figure 76).

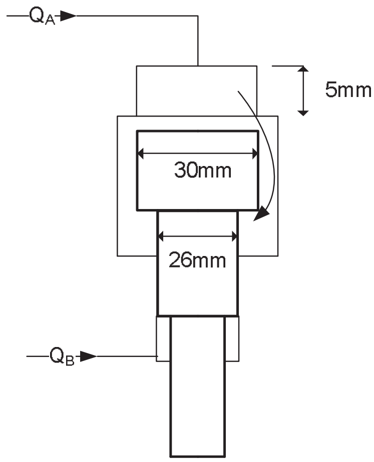


Figure 75 2-stage actuator area concept

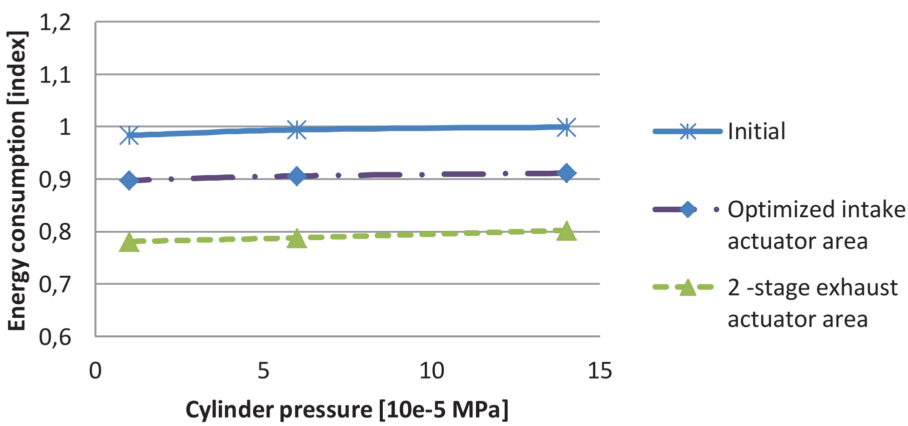


Figure 76 Relative energy consumption of EHVA valvetrain

Because the studied EHVA system is a constant pressure based system, basically all flow needed from the pump takes maximum energy from the power source (engine), even if the actuator pressure need was less than maximum pressure. This is the reason why a 4-way controlled system usually cannot compete with a 3-way controlled system in energy consumption. The usage of the 3-way controlled system and the return spring is lowering the hydraulic flow rate, and thus lowering the pressure/energy losses up to 25% compared to the 4-way system. The system efficiency of the non-return spring system is also lower due the multiple pressure losses over the hydraulic control valve.

8.5 Cam shaft vs. EHVA

According to simulation results, the optimum redesign of the actuator and reduction of return spring stiffness and pre-load would give up to a 20% reduction in power consumption compared to the initial design. With the help of this modification, an acceptable level of total power consumption is easily realized. At maximum cylinder load, the theoretical EHVA power consumption can thus be estimated as 2% of the engine cylinder power, which is about the level what camshaft driven engine is consuming at full engine power. Low cylinder loads are problematic to the EHVA system, because a mechanical cam needs power only according to the counter force of the exhaust valves and return springs, and return springs may help the camshaft to rotate during the GEV closing event. Comparative measurements of camshaft power consumption in an idle engine are not however available.

9 DISCUSSION AND FUTURE WORK

The EHVA system has been driven in a test rig environment, where all basic parameters but real cylinder load, interaction of hydraulic pressure waves between the cylinders, and engine mechanical vibrations can be tested. The test rig phase has revealed some drawbacks, indicating where some minor constructions or materials of the actuator parts are needed to be redesigned. Performance and limits of the system have been investigated, and the system proved sufficient performance and reliability in engine tests. The engine environment puts additional challenging requirements on the system. The most important are reliability and reaction in malfunctions. Mechanical vibrations are giving stress to electronic components. Hydraulic pressure peaks are also shortening lifetime of the components, and they may disturb the actuator control. Safety features try to avoid the system ending up in a critical situation, and they try to help the system go down with minimum damage if a fault already occurred.

9.1 Engine tests

During the engine tests it was quickly noticed that the state controller is very complicated to tune when actual engine performance tests are applied because each cylinder requires slightly different parameters, and therefore the state controller is not so effective to use as expected. Due to this ILC was taken into use in the short run and after that the EHVA system has been able to be used as a ‘standard’ cam actuator system of the test engine, regardless of what kind of test procedures the engine has gone through.

The EHVA system with ILC control system has been running in 4-cylinder (Picture 2) and 6-cylinder test engines in laboratory. Very different engine development tests (not only valvetrain tests!) have been applied during the time frame, and EHVA operation has been found very helpful and time saving in most of the circumstances. At the time of writing, about 2500 cumulative running hours of the two separate EHVA systems have been performed.

Picture 2 EHVA on the 4-cylinder W20 engine

Figure 77 shows the intake valve tracking error of one cylinder in a running 4-cyl engine. ILC is engaged after two seconds, and it can be seen how the controller efficiently decreases the error inside the required range during three seconds. Finally the error range ±0.5mm can be reached, even if it temporarily jumps higher. The minimum achievable error range has been ±0.25mm in the intake valve, because the load condition of the intake GEV is not varying as much as in exhaust valve, and actuator tracking in the beginning and end of the stroke is then much better (Figure 78).

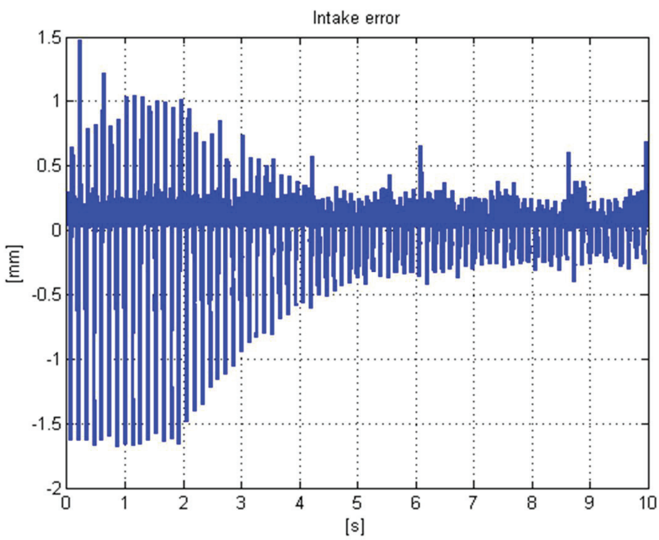


Figure 77 Error development in real engine run [Herranen 2010b]

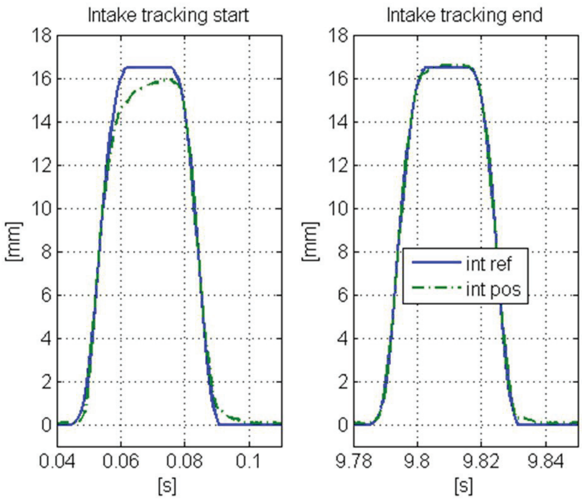


Figure 78 Intake GEV displacement before and after the learning process [Herranen 2010 b]

Engine tests have also revealed problems in displacement sensor reliability (due to mechanical vibrations and hydraulic pressure peaks), which has been investigated and improved with the help of the sensor manufacturer itself. Pressure wave of the tank line caused some trouble in the end of the GEV closing, but the effect was decreased by hydraulic accumulators.

9.2 Safety features

As stated, safety features are naturally important when talking about a critical system in vital parts of the engine. A malfunction of a single component should not cause more damage and the engine should be drivable for as long as possible. Due to this, different safety systems have been built inside the controller.

- Initialization and engine start-up ensure that all sensors give sensible readings and have the right offsets. Also the control system follows the engine start procedure and starts with reduced valve lifts without danger to piston collision etc.
- The virtual combustion piston position is calculated, and the system ensures that the wanted GEV lift does not intercept with the piston curve. Also if the measured GEV lift goes too close to the piston (safe zone), the system gives an alert.
- Because crank angle data is essential to the whole EHVA, the angle sensor output is monitored and in an emergency case can be shortly shunted by a virtual angle.
- The tracking error of the GEV is monitored, and different trigger levels can be set. Here the malfunction of any EHVA component is seen in the first place.

All faults above can then trigger different actions, where the engine (and GEVs) is either shut down as fast as possible, the engine is shut down under control, or the engine is maintained to run but in a limp mode where full load is not possible.

9.3 Future work

In future, the EHVA system could be applied with a digital hydraulic based system and model based control. Hydraulic control valves are on/off valves which are installed parallel in flow lines. The Model-Based Digi Control (MBDC) is basically similar to MBC but with the difference that each valve has only one step, and there are several parallel valves. The valves are simple on/off valves, which are sized and combined so that the wanted discrete flow characteristic as a function of the control input can be achieved (Figure 51). The difference in the digital version compared to analogue is that the transient time between any flow state is the same as the individual valve reaction time. The valve control methods can vary, as can the structure of the valve combination and number of valves. MBDC and its features are more accurately explained in [Linjama 2003a]. On/off valves have been used in gas exchange valve application studies earlier, but the benefit of the full digivalve series has not been investigated until [Herranen 2011].

An example of EHVA systems with MBDC is shown in Figure 79. The presented system is a basic digital hydraulic system, without any choices for differential regenerative or energy recovery connections. In the showed example circuit, there are two individual hydraulic control valves which control flow from the pressure side into the actuator chamber, and thus move the actuator and gas exchange valves. Also, two individual valves control the outflow from the actuator chamber to the tank. Thus the definition ‘2+2 valves’ is used for this particular system. According to the binary coding of the valve sizes, this system can perform 3 different flow states.

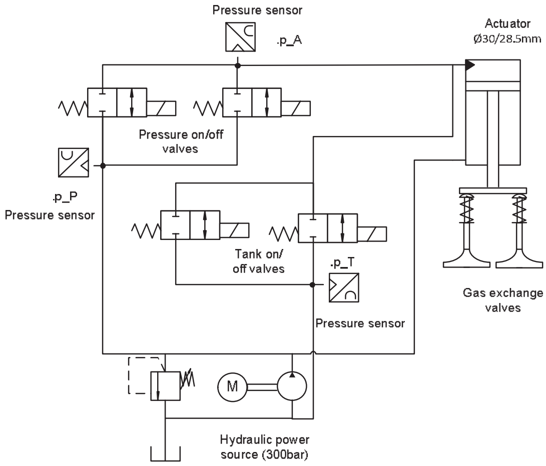


Figure 79 EHVA with a digital hydraulic circuit, MBDC 2 + 2 valves version [Herranen 2011]

MBDC has not been tested because it was not possible to produce a suitable prototype. A lack of commercial high performance on/off valves restricts the overall development of digihydraulic applications, where fast and accurate movements are required. However, simulations can be done, and here only the required number and sizes of the investigated valves, and the simplest and most practical system, were sought. The system was tested also with changing environment parameters, like cylinder load force, valve lift, hydraulic working pressure, and running frequency (engine RPM).

Valves sizes (flow performances) and controller parameters were tuned until reasonable tracking ability in the initial conditions was found in the initial working point. Response time of the hydraulic valves was assumed to be 1ms (1kHz).

Simulations showed that the 2+2 digivalve EHVA system is capable of a tracking error under ± 1mm, where valve sizes are tuned according to the given working point. The requirement was ± 0.5mm, which can be achieved with very careful tuning of the control parameters and only in a narrow working point range at a time. Next, the number of the digivalves was increased. Now three valves were modeled on both the pressure and tank side of the hydraulic system. Pre-tuned valve sizes were 3.1 L/min, 4.7 L/min, and 9.5 L/min at 0.5 MPa in both pressure and tank lines. Now the tracking ability is improved. In addition, sensitivity to the environment parameters is decreased; tracking error is smaller than in the 2+2 valve system in most of the studied cases. With parameter tuning, the target error range of ± 0.5mm could be reached (Figure 80) through the investigated environment parameter range. Fault tolerance of the system in case of one control valve malfunction is still not very good.

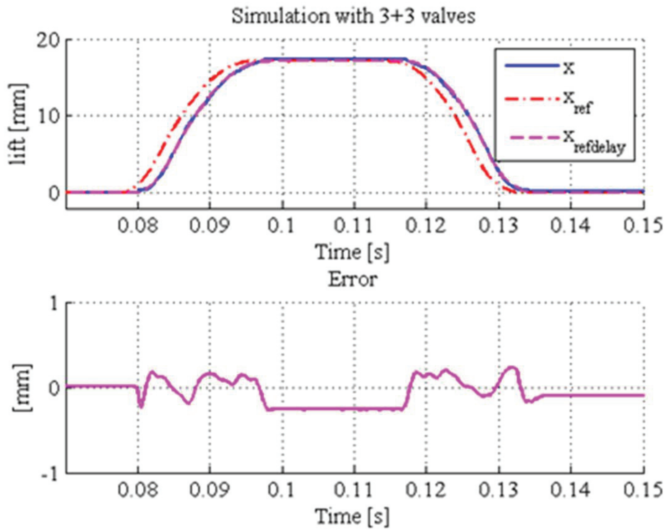


Figure 80 Tracking error of the 3+3 digivalve EHVA [Herranen 2011]

It is clear that an increased valve number results in better controllability, better failure behavior and better energy recovery options. A simple system with the fewest possible parts is naturally the goal when designing industrial applications. Simulations showed that the 3+3 valve system was the ‘smallest’ system which fulfilled the tracking requirements of the studied system, despite changing environment variables.

10 CONCLUSIONS

It can be stated that the developed variable valve actuation system is very suitable for the cyclic diesel combustion process. The chosen hydraulic actuator can provide both high forces and long displacement, and the power density of the actuator is good. The 3-way controlled actuator gives good energy efficiency and simple construction. The control valve is the most critical component of the system. It defines most of the system controllability and response, but also causes the highest energy lost in a conventional resistance controlled hydraulic system. It is possible to implement the depicted EHVA system to medium speed, large bore diesel engines, and the system usability is good. GEV timing and event can be freely controlled inside the diesel engine and combustion process specified limits. Among others, it is possible to apply freely defined gas exchange lift curves, including multiple lift events, and stroke by stroke control where sequential strokes have different lift and timing, including 2-stroke events between the 4-stroke operation. The developed EHVA system can fulfill the requirements on the tracking accuracy and repeatability of the GEV lift.

Applied controller algorithms are tested, and it is found that the P-controller is not good enough for tracking purposes. The state controller can provide a smaller tracking error, but the high number of cross effective parameters makes the usage very time consuming. The model-based controller is relatively light and robust, but with excessive environmental parameter changes or a difficult shape of GEV lift curve, the tracking error limit may not be reached during the whole GEV lift. If a specific GEV lift curve is not required and the dynamics of the system can naturally self-define the wanted curve, MBC is a good choice. The Iterative Learning Controller is then the best choice. It has best tracking capability and it adapts automatically to changes within the system's dynamic response limits. Only the learning process must be handled tightly, and the calculation time of the controller is relatively high.

The return spring of the GEV has a very big effect on the energy consumption, mainly because energy stored in the spring compression is much higher than needed for the GEV closing. The current hydraulic system then wastes this extra energy on the tank line. However, it is not recommended to remove the return spring totally from the system; the spring stores the required energy with good efficiency, and the spring force is needed to keep the intake valve closed if a hydraulic locking force cannot be applied.

Energy consumption of the studied EHVA is challenging when compared to the conventional camshaft driven system, and there is still work to do. A power consumption reduction of up to 20% could be reached with a 2-stage actuator design, and even more redundancy and energy optimization might be reached with a different multi chamber actuator, controller and control valve.

It's not known do the energetic improvements of the whole combustion process overcome the higher power consumption of the EHVA. However, even with the current EHVA, many valuable features can be realized if slightly larger energy consumption is accepted and the versatility of the system is emphasized. In EHVA's present state, it is already a very flexible, useful and powerful tool for laboratory test engine purposes. In the testing environment, varying lift curves against varying engine parameters are often required, and EHVA can respond to these demands well. EHVA's full potential is not revealed yet, and more research for example in the combustion process area is needed in order to utilize the full benefit of the EHVA. Power consumption of the EHVA can be decreased, and in the future this will show whether EHVA is also compatible in overall cost effectiveness, which is needed in commercial production.

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APPENDIX I

Main parameters of the AMESim simulation model:

Mass friction endstops	
Mass [kg]	actuator 0.5, yoke 0.6, GEV 2x0.8 (incl. 1/3 of the spring mass)
Stiction force [N]	actuator 500, yoke 20, GEV 2x20
Coefficient of viscous friction [N/(m/s)]	actuator 100, yoke 25, GEV 2x20
Coulomb friction force [N]	actuator 75
Pipe (fluid inertia + pipe friction)	
Dimensions between control valve – Actuator chamber and manifold – control valve	Diameter 10mm, pipe length 0.2m, relative roughness 1e-5
Dimensions between Pump – manifold, control valve - tank	Diameter 19 mm, pipe length 2m, relative roughness 1e-5
Actuator Diameters [mm]	upper piston 30, lower piston 30, lower rod 28.5
Elastic_endstops	
contact stiffness [N/m]	1,00E+08
contact damping [kN/(m/s)]	50
penetration for full damping [mm]	0,01
GEV	
Valve diameters [mm]	exhaust 66 (rod 14.5), intake 73 (rod 13.8)
Poppet diametres [mm]	exhaust 55.5, intake 57.7
Flow coefficient	0,72
Spring stiffness [N/mm]	60, (Miller 100)
Spring pre-tension [N]	1000. (Miller 3000)
Fluid	
Temp [C]	40
Density [kg/m^{3}]	850
Bulk modulus [bar]	17000
absolute viscosity [cP]	40
air/gas content [%]	0,1
Control valve	
Valve natural frequency [Hz]	200
Damping ratio	0,5
Flow rate at the valve at maximum valve opening [L/min]	100
Corresponding pressure drop [bar]	35
Accumulator	
Volume [L]	0,35
Pre-charge pressure 0.8 x working pressure	

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