11TH INTERNATIONAL FLUID POWER CONFERENCE

19th - 21st MARCH 2018 | PROCEEDINGS

Vol. 1

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VOLUME 1 - SYMPOSIUM:
MONDAY, MARCH 19TH

11TH INTERNATIONAL FLUID POWER CONFERENCE
19th - 21st of March 2018
Aachen, Germany

Volume 2
- Conference: Tuesday, March 20th
- Scientific Poster Session

Volume 3
- Conference: Wednesday, March 21st
Fluid Power Networks has drawn a lot of attention and the organizing team and all collaborators at IFAS are proud that you followed our invitation to join the 11th International Conference on Fluid Power in Aachen. Welcome to the academic and truly intercultural city in the heart of Europe!

The traditional IFK is one of the world’s largest scientific conferences on fluid power and unites scientists with industrial delegates at an international forum to exchange knowledge in the area of fluid power drives and systems. The first conference (1. AFK, Aachener Fluidtechnisches Kolloquium) was conducted in 1974 by Prof. Wolfgang Backé. Since 1998 the Institute for Fluid Power Drives and Controls (IFAS) at RWTH Aachen University and the Institute of Fluid Power (IFD) at TU Dresden alternately organize the International Fluid Power Conference (IFK) every two years. This year we host 141 scientific contributions and speakers. Attendees from 30 countries are registered. 36 companies exhibit their products in a designated floor area in the Eurogress. A special highlight of the 11th IFK is the 50th anniversary of Fluid Power Research at RWTH Aachen University which will be celebrated on Wednesday, March 21st. It is accompanied by a change in directorship at IFAS. Prof. Katharina Schmitz will be introduced as my successor.

The program starts on Monday morning with a symposium where researchers from mainly universities and other research facilities but also from companies have the opportunity to present their research projects to a wide international community of scientists. In the evening of the first day all participants are invited to the opening event that marks the start of the exhibition.

The second day begins with the opening address of Mr. Christian Kienzle of VDMA followed by two plenary lectures to actual IoT subjects as they concern fluid power. On Tuesday there are six groups in two parallel sessions of presentations covering a wide variety of application and technology oriented topics in the time of digitalization. The banquet is held at the Coronation Hall of the Aachen town hall and will be followed by an after show in the Aula Carolina. On Wednesday the program continues in three parallel sessions. The Rector of RWTH, Prof. Schmachtenberg will talk in the concluding plenary session, arrange the hand over in the directorship of IFAS and initiate the anniversary events.

A traditional cultural program will be offered in the surrounding of Aachen as well as an excursion following the conference on Thursday and Friday.

Finally, we would like to express our thanks to all members of the program and organizing committee, scientific advisory board, plenary and keynote speakers, speakers, reviewers, chairmen and exhibitors for their time and commitment helping to conduct another successful conference and we hope that you will enjoy the 11th IFK in Aachen.

H. Murrenhoff
GROUP A - MOBILE APPLICATIONS

Nicolas Brötz Integrated Fluid Dynamic Vibration Absorber for Mobile Applications 14
Jun.-Prof. Jörg Edler A new Approach on a Hydrostatic Motor for Applications in Mobile Cranes 26
Jun.-Prof. Ajit Kumar Performance Investigation of a Hydro-pneumatic type Accumulator used in a Hydrostatic Drive System of Off-road Vehicles 36
PhD Min Cheng Active damping improvement of the electrohydraulic control system with dual actuators for mobile machinery 50

GROUP B - ENERGY MANAGEMENT

PhD Bin Yu Accurate Control Method of Vane Direction Based on Pressure Difference Feedback in Active Yaw System for Wind Turbines 60
Linart Shabi Investigation of Potentials of Different Cooling System Structures for Machine Tool 82
PhD Chong Liu An energy efficiency evaluation method based on least squares combination weighted in refrigeration system 96

GROUP C - SYSTEMS

PhD Tatjana Minav Adaptive Control for direct-driven hydraulic drive 110
PhD Alexander Mitov Identification and synthesis of linear-quadratic regulator for digital control of electrohydraulic steering system 120
Jan Siebert Marco Wydra Development and Implementation of a Control and Regulation Concept for a Hydraulic Load Unit 130
Matti Linjama Qi Zhong Fault-Tolerant Control of a Multi-Outlet Digital Hydraulic Pump-Motor 144
Design of control system for independent metering valve 158

GROUP D - DESIGN PROCESS

Ryan Jenkins A Semi-Empirical Lumped Parameter Model of a Pressure Compensated Vane Pump 168
Tobias Speicher Process-driven component adjustment in variable speed pump drives – development of a strategy to increase the overall energy efficiency 182
Artemi Makarow Holistic Approach to the System Optimization of a Proportional Valve 198
Andrea Lucchi System optimization by means of an integrated design: the Dana Brevini case 212
Enrico Pasquini Pressure Loss in Unsteady Annular Channel Flow 222

GROUP E - COMPONENTS

Florian Schoemacker Piston slippers for robust water hydraulic pumps 236
James Marschand Comparison of a Variable Displacement 3-Piston Inline Digital Pump using Electrically and Mechanically Actuated Poppet Valves 248
Prof. Robert Castilla Fluid Dynamic Effects of Interteeth and Sideway Clearances on a Mini Gerotor Pump using Dynamic Meshing Decomposition 260
Ying Li Experimental study on churning losses reduction for axial piston pumps 272
Prof. Bulent Sarlioglu Investigation of the Aerodynamics Characteristics of the Integrated Motor-Compressor 282

GROUP F - MOBILE APPLICATIONS

PhD Tatiana Minav Control strategy for a direct driven hydraulics system in the case of a mining loader 294
Prof. Andrea Vacca Combining Control and Monitoring in Mobile Machines: the Case of an Hydraulic Crane 306
PhD Ruqi Ding Fault-tolerance Operation for Independent Metering Control Valve 320
Kerstin Ritters Efficiency studies on double pump supply units 334
PhD Lei Ge High Energy Efficiency Driving of the Hydraulic Excavator Boom with an Asymmetric Pump 346
### 02:00 - 03:45 p.m.

**GROUP G - TRIBOLOGY & FLUIDS**

<table>
<thead>
<tr>
<th>Author</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tobias Mielke</td>
<td>Entrainment of free water into a hydraulic system through the rod sealing</td>
<td>358</td>
</tr>
<tr>
<td>Joep Nijssen</td>
<td>Development of an interface between a plunger and an eccentric running track for a low-speed seawater pump</td>
<td>370</td>
</tr>
<tr>
<td>Alexander Terwort</td>
<td>Bubble nucleation in hydraulic systems</td>
<td>380</td>
</tr>
<tr>
<td>Lars Brinkschulte</td>
<td>An approach to wear simulation of hydrostatic drives to improve the availability of mobile machines</td>
<td>392</td>
</tr>
<tr>
<td>Yuan Chen</td>
<td>Investigation of Laser surface texturing for Integrated PV (pressure×velocity)-value-decreased Retainer in an EHA Pump</td>
<td>408</td>
</tr>
</tbody>
</table>

**GROUP H - PNEUMATICS**

<table>
<thead>
<tr>
<th>Author</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stephan Merkelbach</td>
<td>Development of a rotary pneumatic transformer</td>
<td>420</td>
</tr>
<tr>
<td>David Straub</td>
<td>Experimental and Theoretical Investigation of Lightweight Pumps and Fluid Reservoirs for Electrically Driven Vacuum Systems in Automated Handling Processes</td>
<td>434</td>
</tr>
<tr>
<td>Annabell Effner</td>
<td>Fast Switching Pneumatic Valves Driven by Magnetic Shape Memory Materials</td>
<td>446</td>
</tr>
<tr>
<td>PhD Miha Pipan</td>
<td>Closed-loop control algorithm for fast switching pneumatic valves</td>
<td>460</td>
</tr>
<tr>
<td>Filipp Kratschun</td>
<td>Transient simulation of a pneumatic sharp edged L-shape fitting</td>
<td>472</td>
</tr>
</tbody>
</table>

### 04:15 - 06:00 p.m.

**GROUP I - NEW & SPECIAL APPLICATIONS**

<table>
<thead>
<tr>
<th>Author</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>PhD Vito Tič</td>
<td>Low compressibility of ionic liquids and its effects on pulsation within hydraulic system</td>
<td>486</td>
</tr>
<tr>
<td>Marcel Rückert</td>
<td>High Pressure Falling Cylinder Viscometer-Error Analysis and Improvement Proposal</td>
<td>494</td>
</tr>
<tr>
<td>Nils Preuß</td>
<td>Accumulators with sorbent material – an innovative approach towards size and weight reduction</td>
<td>504</td>
</tr>
<tr>
<td>Andreja Poljšak</td>
<td>Polymer composites materials for water hydraulic seat on/off valves</td>
<td>518</td>
</tr>
<tr>
<td>PhD Niels Diepeveen</td>
<td>Field tests of the DOT500 prototype hydraulic wind turbine</td>
<td>530</td>
</tr>
</tbody>
</table>

**GROUP J - TRIBOLOGY & FLUIDS**

<table>
<thead>
<tr>
<th>Author</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dominik Krahl</td>
<td>Burning Hydraulics – Experimental Investigations of the Micro-Diesel Effect and Gas Discharge within Models of a Valve and a Pump</td>
<td>538</td>
</tr>
<tr>
<td>Paul Michael</td>
<td>An Investigation of the Effects of Fluid Composition on Aeration, Efficiency, and Sound Generation in an Axial Piston Pump</td>
<td>550</td>
</tr>
<tr>
<td>Julian Angerhausen</td>
<td>Influence of transient effects on the behaviour of translational hydraulic seals</td>
<td>562</td>
</tr>
<tr>
<td>Tobias Corneli</td>
<td>Reduction of bearing load capacity due to measured wall slip</td>
<td>574</td>
</tr>
<tr>
<td>Lizhi Shang</td>
<td>Advanced Heat transfer model for piston/cylinder interface</td>
<td>586</td>
</tr>
</tbody>
</table>
The setup of a suspension always leads to a compromise between comfort and safety. In order to counteract this in a passive approach, one could attach a structural extension in the form of a dynamic vibration absorber to the axle. Thus, energy of the wheel vibrations is diverted into the vibration absorber instead of the body. In comparison to a classic dynamic vibration absorber, which is not in the sense of lightweight construction due to the additional mass, our Fluid Dynamic Vibration Absorber (FDVA) reduces the dynamic mass by using a hydrostatic transmission.

Keywords: dynamic vibration absorber, driving safety, suspension strut, hydraulic transmission

Target audience: mobile hydraulics, automotive, hardware-in-the-loop

1 Introduction and Motivation

The essential requirements for a suspension strut of modern vehicles are high driving safety and high driving comfort at the same time. The control of the driver over the vehicle can only be ensured if there is a contact between the wheel and the ground at any time. Because only the tire has contact to the road, the vertical force of the suspension strut has to be sufficient to ensure the best transmission behavior of horizontal maneuvers /1/.

According to Mitschke /2/, the dynamic wheel load fluctuation is a measure of driving safety. The driving comfort is influenced decisively by acceleration acting on the occupants. In reality, the evaluation of driving comfort is very complex and can be evaluated as better or worse by the personal feeling of the occupants. The driving comfort is expressed in simplified terms by the effective value of the amplitude of the vertical acceleration of the body. The suspension strut of a vehicle is the connection between wheel and body, thus affects both driving safety and driving comfort. The setup of a classical suspension system with spring and shock absorber leads to a compromise between driving safety and driving comfort. This compromise becomes obvious in the so-called conflict diagram, where the fluctuation of the body acceleration is plotted over the relative wheel-load fluctuation, Figure 1. The solid lines in this figure represent a constant damping, the dashed lines constant spring stiffness.

Moving to a higher solid line in the conflict diagram, the damping rate increases. A higher damping rate leads to a higher driving safety but on the other hand, it reduces the driving comfort because shocks are transmitted more strongly from tire to body. Therefore, the Pareto line, which represents the absolute optimum, cannot be undercut by any combination of damping coefficient and spring stiffness for this system.

There are several approaches to shift the Pareto line. On the one hand, there are active systems that adapt the force transmitted by the wheel, such as the active air spring /4/ or Daimler’s ABC system /5/. On the other hand, there is also the possibility of using passive systems with different structures.

One well-known structural extension of a passive system is a vibration absorber, which is capable of reducing vibrations of the wheel in a defined frequency band without affecting the body acceleration. This is realized by redirecting the energy of the accelerated wheel into the structure extension.

A further aim is to make the dynamic vibration absorber as light as possible in order to follow the trend of lightweight construction and not to attach unnecessarily high additional masses to the vehicle.

In this paper, we examine how a dynamic vibration absorber, which reduces the heavy mass of the absorber by means of a hydraulic transmission at the same absorber inertia, can be integrated into the suspension strut /6, 7, 8/. Furthermore, we show a mitigation of oscillations by using the fluid dynamic vibration absorber.

For this purpose, firstly the theory of the Fluid Dynamic Vibration Absorber (FDVA), a vibration absorber with a hydraulic transmission, is shown. Secondly, we describe the structure of the functional demonstrator and the experimental set-up (component and HIL tests), which we use to validate the theoretical model. Thirdly, we discuss our results and give a short outlook.

2 Theory

A vibration absorber consists of a capacitance and an inertia and is used in many systems to reduce vibrations. In the case of a mechanical vibration absorber, a comparatively small mass is attached with a spring to the oscillation system. This structural extension takes the energy out of the oscillating basic system, in our case the wheel. Due to the extended Den Hartog principle, the natural frequency of the dynamic vibration absorber in a suspension strut has to be close to the natural frequency of the wheel /9/. An additional damping of the vibration absorber enables its usage over a wide frequency range but also reduces the vibration isolation in the natural frequency. The vibration energy is dissipated in the damper. Damping is therefore desirable, as long as the system is not only operating stationary in the natural frequency of the vibration absorber. The transfer functions for a vibration absorber with and without damping are shown in Figure 2.

Figure 2: Single mass system with a vibration absorber amplitude and its transfer function.

The closer the absorber mass is to the system mass, the more energy can be absorbed /10/. On the contrary, a higher absorber mass is in conflict with the sense of lightweight construction. This conflict can be overcome by using the
The piston rod protrudes on both sides of the hydraulic volume to keep it constant indecently of the strut deflection.

3.1 Functional Demonstrator

The functional demonstrator is adapted to the dimensions of the installation in the SFB demonstrator, a load-bearing structure for the examination of uncertainty /3/, and is no optimized solution for a vehicle installation. The rod end of the FDVA on the left-hand side in Figure 4 is connected to the wheel axle. At this point, the vibrations are introduced into the absorber. The force is transmitted to the piston rod via two compression springs that are tensioned against each other. The mutual tension allows oscillation in the z-direction. Springs with four stiffnesses from 302 N/mm to 63 N/mm can be used for the tests.

The piston rod protrudes on both sides of the hydraulic volume to keep it constant indecently of the strut deflection. A piston, which is linked to the rod and driven by the spring force, presses the oil of one chamber through ducts into the other chamber. Thus, the hydraulic transmission consists of a double-acting hydraulic cylinder whereby the chambers are connected via several ducts on the outer side.

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Figure 7: Two-mass oscillator model with FDVA.

Table 3: Parameter of the virtual two-mass oscillator.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>body mass $m_B$</td>
<td>290 kg</td>
</tr>
<tr>
<td>wheel mass $m_w$</td>
<td>40 kg</td>
</tr>
<tr>
<td>body spring stiffness $k_B$</td>
<td>19.7 N/mm</td>
</tr>
<tr>
<td>tire stiffness $k_W$</td>
<td>200 N/mm</td>
</tr>
<tr>
<td>body damping $d_B$</td>
<td>1170 Ns/m</td>
</tr>
</tbody>
</table>

The excitation with frequency shares up to 25 Hz corresponds to a drive over a federal highway (Bundestraße) at a driving velocity of 100 km/h /4/. The term “wheel mass” describes the unsprung mass, thus parts of the suspension, brake and the wheel /3/. With this excitation and the measured forces at the FDVA, the simulation model calculates the compression of the strut. This signal is transmitted to the MTS test damper system, which applies the actual compression. The simulation model is based on the equation

$$m_B + (1 + a^2)m_f \quad 0 \quad -a(1 + a)m_f \quad 0 \quad 0$$
$$-a(1 + a)m_f \quad 0 \quad [m_f + a(2f + a)m_f]d_B$$
$$z_W$$

where $p_v$ is the pressure losses.

4 Results

The simulation results of the FDVA show a significant decrease of the wheel oscillation at almost constant body oscillation, Figure 8. The transfer function is defined by $V(x) = \frac{z_W}{z_B}$ in the following section.

Figure 8: Simulation results of a quarter vehicle model at a harmonic excitation of 5 mm amplitude.

The measurements are used to validate the virtual model of the FDVA. The results are divided into damping, natural frequency and HiL-Simulation.
4.1 Damping of the Hydraulic Transmission of the FDVA

An important component of the FDVA is the hydraulic transmission. This acts almost like a damper. In contrast to a damper, the FDVA has a significantly larger inertial mass and therefore an additional force acts at the reversal point of a harmonic excitation, as shown exemplary in Figure 9. For the calculation, the body motion is set to zero and the force on the body is calculated using the motion equation of the body. It follows the force on the body is

\[ F_B = -\alpha(1 + \alpha)\dot{z}_B + p_u a, \]

(2)

where \( \dot{z}_B \) is the acceleration of the excitation \( z_B \). The measurement results fit qualitatively and quantitatively well to the simulation results, whereby Coulomb's friction was not taken into account in the simulation, which is why the curve of the simulation is not widened at the turning points. The Coulomb's friction, which is identified experimentally, is approximately 30 N.

![Figure 9: Body force of the FDVA at a harmonic amplitude of 25 mm at 1 Hz.](image)

The harmonic excitation at 1 Hz and 25 mm amplitude corresponds to the maximum velocity and thus forces occurring in the HiL test. Consequently, it can be shown that the attenuation in this range can be described with the simulation model. The oscillations of the measurement result from a measuring noise, as well as the control of the testing machine.

4.2 Natural Frequency of the FDVA

To ensure that the hydraulic transmission is working properly, the FDVA is regarded as a single-mass oscillator with excitation via the spring, shown in Figure 10. The transfer function for this case is

\[ V_\text{SIM}(s) = \frac{1}{\sqrt{1 - \eta^2}} \cdot s^2 \theta_\text{sim}, \]

(5)

according to Magnus /11/. The phase is calculated with

\[ \tan \psi = 2D\eta/(1 - \eta^2). \]

(4)

The natural frequency of the analytical solution is shown at the intersection with the phase shift of \( \pi/2 \) and also at maximum of amplification function. Figure 11 shows the measured values in comparison to the analytical model for one opened duct. With a parameter identification, the natural frequency of the FDVA can be estimated for the phase shift of \( \pi/2 \). At \( f_{\text{real}} = 15 \text{ Hz} \), the measured natural frequency is slightly higher than that of the analytical model, which is designed for a natural frequency of \( f_{\text{sim}} = 13.1 \text{ Hz} \). The higher frequency \( f_{\text{real}} \) can be explained by the fact that Coulomb's friction is not taken into account. The difference in shape of the curve, especially for the amplification, can also be explained by the non-linear friction, but has no influence on the determination of the natural frequency by means of the phase shift.

Figure 12 shows the same curves as Figure 11 for 2 opened ducts. By opening a second duct, the inertia changes as shown in Table 1. With a lower inertia, the natural frequency increases. This behavior can also be well reproduced.

The equation for the natural frequency can be used to calculate the existing inertia, which leads to an inertia of \( \theta_{\text{real,1}} = 33.1 \text{ kg} \) for one open channel and \( \theta_{\text{real,2}} = 19.1 \text{ kg} \) for two open channels. In this case, the simulation value is higher. Further investigations of Coulomb's friction and leakage currents at the piston seal must be carried out. The real functional demonstrator does not quite reach the simulated value, but is much larger than the heavy mass of \( m = 1.6 \text{ kg} \). The inertia thus is translated hydraulically.

4.3 HiL Simulation Results of the Functional Demonstrator

The vehicle is modeled virtually in the real-time simulation environment and coupled with the real FDVA on the test damper system. This makes it possible to examine a component independently of other components in the vehicle. The measurement of the FDVA in the HiL simulation not only provides results for the actual component but also for the quarter vehicle. This makes it possible to show the structure vibrations, wheel vibrations, as well as body acceleration and wheel load. The results for a drive over a federal highway described by Hedrich /4/ can be seen in Figure 13. The considered quantities are related to the excitation.

The two graphs show a particular agreement in the natural frequency range of the FDVA. Outside this range, deviations can be detected, but no model adjustment has yet been carried out. Remarkable are the better results of...
In the next step, we plan to optimize the design of the vibration absorber, which can be achieved by simply adjusting the vibration absorber spring. An adjustment of the natural frequency of the vibration absorber is necessary, because a vibration absorber that is not optimally designed for the system natural frequency could even worsen the vibration behaviour.

Furthermore, it is conceivable to realize a semi-active version of the Fluid Dynamic Vibration Absorber by designing the closing of the channels with electromagnetic valves. This means that the natural frequency of the absorber can be adapted to the driving situation.

The advantage of the Fluid Dynamic Vibration Absorber is the increase of driving safety. This is particularly important in motor racing. Downsizing of the vibration absorber is necessary to integrate it into a racing car. If the vibration absorber springs are located inside of the hydraulic transmission and only the piston moves on the piston rod, it is possible to avoid the extension of the piston rod, which leads to a significant reduction of the length.

6 Acknowledgements

The authors would like to thank the German Research Foundation (DFG) for funding this research within the Collaborative Research Centre (CRC) 805 “Control of Uncertainty in Load-Carrying Structures in Mechanical Engineering” (TU Darmstadt, speaker Prof. Dr.-Ing. Peter F. Pelz). Furthermore, the authors especially would like to thank the project cooperation partner ZF for supporting this project.

Nomenclature

The first column of the following table shows the symbols utilized for physical and mathematical quantities. The second column shows the meaning of each quantity. The dimension of each physical quantity is denoted in the third column, based on the generic quantities length (L), mass (M) and time (T).

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Dimensions</th>
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</thead>
<tbody>
<tr>
<td>a</td>
<td>duct surface</td>
<td>L^2</td>
</tr>
<tr>
<td>A</td>
<td>piston surface</td>
<td>L^2</td>
</tr>
<tr>
<td>D</td>
<td>damping coefficient</td>
<td>L</td>
</tr>
<tr>
<td>d_B</td>
<td>body damping</td>
<td>MT^-1</td>
</tr>
<tr>
<td>f_B</td>
<td>natural frequency</td>
<td>T^-1</td>
</tr>
<tr>
<td>F_B</td>
<td>body force</td>
<td>MLT^-2</td>
</tr>
<tr>
<td>l</td>
<td>duct length</td>
<td>L</td>
</tr>
<tr>
<td>L</td>
<td>chamber length</td>
<td>L</td>
</tr>
<tr>
<td>k_A</td>
<td>absorber spring stiffness</td>
<td>MT^-2</td>
</tr>
<tr>
<td>k_B</td>
<td>body spring stiffness</td>
<td>MT^-2</td>
</tr>
<tr>
<td>k_W</td>
<td>tire stiffness</td>
<td>MT^-2</td>
</tr>
<tr>
<td>m</td>
<td>mass</td>
<td>M</td>
</tr>
<tr>
<td>m_f</td>
<td>fluid mass</td>
<td>M</td>
</tr>
<tr>
<td>m_B</td>
<td>body mass</td>
<td>M</td>
</tr>
<tr>
<td>m_vA</td>
<td>vibration absorber mass</td>
<td>M</td>
</tr>
<tr>
<td>m_W</td>
<td>wheel mass</td>
<td>M</td>
</tr>
</tbody>
</table>

Figure 13: Comparison of a HiL measurement of the FDVA with the corresponding simulation results and a simulation without FDVA.

5 Summary and Conclusion

In this article, a new concept of a hydraulically transmitted vibration absorber for installation in a suspension strut was presented. The functionality of the Fluid Dynamic Vibration Absorber was shown and first component measurements as well as measurements in the Hardware-in-the-Loop environment with a virtual quarter car were used to validate the axiomatic model. The simulation model has already shown that it is possible to improve driving safety without, or only slightly, affecting driving comfort and thus shifting the Pareto line. The measurements presented show that the vibration absorber fulfils its function and that the hydraulic transmission of the vibration absorber inertia is working quite well.
\[ m_p \] piston mass \[ \text{M} \]

\[ p_v \] pressure losses \[ \text{M L}^{-1} \text{T} \]

\[ x_0 \] excitation \[ \text{L} \]

\[ x_b \] body excitation \[ \text{L} \]

\[ x_{VA} \] vibration absorber excitation \[ \text{L} \]

\[ x_{W} \] wheel excitation \[ \text{L} \]

\[ \alpha \] ratio between piston and ducts \[ \text{L} \]

\[ \eta \] ratio of frequency \[ \text{L} \]

\[ \rho \] oil density \[ \text{M L}^{-3} \]

\[ \Theta \] inertia \[ \text{M} \]

\[ \omega_0 \] natural frequency \[ \text{T}^{-1} \]

References


10/ Neubauer, M., Schwingungsschutz, Schwingungsmessung, Aktive Systeme, Lecture Notes, Leibnitz University Hannover, Hannover, Germany, 2009.


12/ Magnus, K., Popp, K., Sextro, W., Schwingungen, Springer, Wiesbaden, Germany, 2013.
A new Approach on a Hydrostatic Motor for Applications in Mobile Cranes

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Mobile hydraulic linear actuators are a fixed part of many applications. Especially in mobile cranes, they are used for the movement of the booms and are characterized with a light power to weight ratio. The kinematics can be seen as a restriction of linear motor in mobile cranes. On one side the possible range of the motion and on the other side the unfavourable constellations of the triangle of force are essentially restrictions in the construction of mobile cranes. For the avoidance of these restrictions exists approaches by in the joints arranged hydrostatic rotational motors. At present these solutions fails by a to high weight to force ratio, or they give no benefit in the kinematics. In this article, a new approach of a hydrostatic rotational motor will be presented, which is characterized by low weight and high torque. By the possibility to rotate endless these rotation hydrostatic motors are predestined as a direct drive in the joints of mobile cranes, to get new possibilities in the kinematics and the construction of the booms.

Keywords: mobile cranes, hydrostatic rotary motor
Target audience: mobile hydraulics, components

1 Introduction

The kinematics of mobile cranes, especially in applications for the agriculture and forestry industries, also at mobile concrete pumps is mostly restricted by the linear motors of the booms. The use of lever to increase the swivel range of single booms for more freedom in motion leads to a higher weight of the crane and to unfavourable forces at the linear motors (see figure 1).

The special requirements of the concrete industry represent a big challenge on the kinematics of automotive concrete pumps. When a customer buys an automotive concrete pump, he must decide the length of the crane and how the crane is folded on the vehicle. Depending on the type of folding, there are advantages when working outdoors or when working in buildings or in pits. The booms of concrete pumps can be fold in two different types. By the z-fold, the booms are applied either from above or from below to the previous boom (see Figure 2 left). The roll folding type applied all booms from below to the previous boom (see Figure 2, right). Depending on the type of folding, the kinematics result in different working areas of the concrete pump crane.

By replacing the linear motors in the individual booms through rotary motors in the respective joints of the booms many advantages are given in the kinematics. The main advantage is, that the customer must make no decision which type of folding he wants to use. Depending on the design, the individual booms can cover large angular areas, or even rotate endless. Thus, there are no restrictions in the kinematics. However, the implementation of rotary motors in the joints of the booms of cranes cannot be implemented without problems. The greatest problem results in limitations in the kinematics and in the power weight (see Table 1). The above-described concepts can only be used in the last joint of the booms of automobile concrete pumps, because the power weight or the maximal achievable torque is a limit in the use in all joints of the booms of automobile concrete pumps.

The in this article presented hydrostatic motor don’t have the above described limitations in the kinematics, power weight or in the achievable torque and can be used in all joints of the booms of automotive concrete pumps.

Figure 1: Principle sketch of implementation of linear actuators on mobile cranes, source: own illustration

Figure 2: Different sequences to fold the booms of mobile cranes, source: own illustration
2.2 Kinematics

The linear motion of the individual pistons is converted into a rotary movement by the plane toothing (see figure 4). Both pistons are mechanically connected to each other via the splined shaft (see figure 3, shaft (3)). This means, that the pistons can move independently linearly, but any twist which a piston undergoes through the plane toothing is also transmitted to the second piston. This mechanical coupling is especially in the reversal points necessary.

<table>
<thead>
<tr>
<th>Hydrostatic Rotary Motor</th>
<th>Torque in kNm</th>
<th>Max Operating Pressure in bar</th>
<th>Weight in kg</th>
<th>Operating Angle in °</th>
</tr>
</thead>
<tbody>
<tr>
<td>Eckart SM4/300</td>
<td>85</td>
<td>250</td>
<td>1456</td>
<td>360</td>
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<tr>
<td>Hägglunds Drive CB280</td>
<td>92</td>
<td>350</td>
<td>705</td>
<td>∞</td>
</tr>
<tr>
<td>Direct Drive, Schwing GmbH</td>
<td>120</td>
<td>300</td>
<td>120</td>
<td>∞</td>
</tr>
</tbody>
</table>

Table 1: Comparison of different construction principles of hydrostatic rotary motors.

In Table 1, three hydrostatic rotary motors are compared with respect to power weight and swivel angle. The motor marketed by the company Eckart /5/ as the construction of Martin /6/ which is also described by Holmes /7/ is also mentioned in the patent of the Putzmeister Engineering GmbH /3/. In the points of power weight and kinematics the hydrostatic rotary motor of the company Hägglunds is close to the here presented direct drive of the company Schwing GmbH.

2 Direct Drive

2.1 Construction

The design of the direct drive is shown in Figure 3. The rotary motor consists of two hydraulic cylinders (2) with the same area ratio, each of them has plane gear teeth on the two piston surfaces. The housings (1) are likewise designed with plane gear teeth on the planar surface. Both pistons are mechanically connected to each other via a splined shaft (3). Either the toothings of two associated housing or the pistons have an offset by half a tooth width. (see Figure 4). By the splined shaft have both pistons an offset of a quarter tooth. While the offset by a half tooth is necessary to generate the torque of one cylinder, the offset by a quarter of a tooth is necessary to generate torque in the death point of the cylinders during the reversal of the direction.

Figure 4 shows a simplified piston chamber of a cylinder with plane toothing. The simplified piston moves along the spline shaft (1). Due to the plane toothing, the longitudinal movement of the piston is converted into a rotational movement of the housing (2). When the piston is at a dead point, it has no longer any contact with the opposite plane toothing of the housing. The piston has threaded out of the toothing. In order to be able to thread into the next tooth gap during the movement reversal, the second piston is necessary. During the reversal movement of a piston is the second piston in its middle position between both housings. Therefore, the second piston can generate the necessary torque to move the boom. Thereby, the second piston ensures that the piston which is just in the movement reversal, can thread into the next tooth gap.

The resulting motion profile is shown in Figure 5. It shows the stroke of both pistons over the time. It can be seen, that the directional reversal of the pistons must take place in an infinitely short time (peak at 0 mm and 20mm strike). The following measure is taken to ensure a secure threading into the next opposing tooth gap. The pistons are not moved into the maximum possible position (dashed line in Figure 5), thereby the teeth have more space which is necessary to thread into the opposing teeth gap. As a result, it is also necessary to reduce the height of the individual teeth as well (dash-dotted line in Figure 5). This results in a trapezoidal movement profile (see Figure 6, top) of the individual pistons, whereby the speed of the individual pistons as well as the position of the individual pistons must be coordinated with each other.

2.2 Kinematics

The linear motion of the individual pistons is converted into a rotary movement by the plane toothing (see figure 4). Both pistons are mechanically connected to each other via the splined shaft (see figure 3, shaft (3)). This means, that the pistons can move independently linearly, but any twist which a piston undergoes through the plane toothing is also transmitted to the second piston. This mechanical coupling is especially in the reversal points necessary.
Depending on the situation, whether a piston is standing still, moving without load, moving together with the second piston, or generating the required torque alone, results different pressure differences (see Figure 7 below).

Figure 5: Movement profile of individual pistons, source: own illustration

An ideal motor is a hydrostatic rotary motor with ideal geometry, without backlash and without tolerances. In addition, an infinitely fast movement direction reversal is assumed. This results in the previously described ideal trapezoidal movement profile (see Figure 6, top). The volumetric flow can be calculated by the area of the piston and the velocity of the piston (see equation 1).

\[ Q = V_{K_{\text{tooth},1}} \times A_{K_{\text{tooth}}} + V_{K_{\text{tooth},2}} \times A_{K_{\text{tooth}}} \]  

(1)

It can also be seen, that the volumetric flow to the motor can change by a factor of 2, depending on whether both pistons are moving or when one of the pistons is in the reverse phase (see Figure 6 below). The black vertical lines represent the movement reversal and the dwell time, which is necessary to thread into the opposing tooth gap.

The torque of the motor can be calculated by the difference pressure of the piston, the area of the piston and the angle of the plane gear teeth (see equation 2–4).

\[ F_{K_{\text{tooth}},\text{assistant}} = \Delta p \times A_{K_{\text{tooth}}} \]  

(2)

\[ F_{K_{\text{tooth}},\text{radial}} = \frac{F_{K_{\text{tooth}},\text{assistant}}}{\tan \alpha} \]  

(3)

\[ M_T = F_{K_{\text{tooth}},\text{radial}} \times r_{\text{w}} \]  

(4)

On the other way with the weight and the length of the boom the difference pressure \( \Delta p \) of each cylinder can be calculated (see Figure 7). The corresponding motion profile is shown again in Figure 7 top. In this case, the black vertical lines represent the start of engagement of the individual pistons. Due to the shortening of the tooth heads, the start of engagement of the toothing deviates from the beginning of movement of the individual pistons.
3 Realized Motor

In the case of the realized motor, the pressure curves differ to the ideal motor. These deviations have “internal” constructive and manufacturing reasons, as well as “external” reasons such as long supply pipes, fluctuating loads during one piston stroke, or vibrations of the entire structure of the crane.

Furthermore, the switching points of the individual pistons must be precisely matched to one another in order to be able to achieve a smooth movement of the boom. For each moving direction of each piston results a stop point and a starting point. Each point can be set either too early or too late. In principle, it can be said, that a deviation from the ideal movement results in a blocking of the motor. Figure 8 shows the measured pressure profile of each cylinder chamber of a prototype. The corresponding movement profile of the individual pistons corresponds to Figure 7 top.

The vertical lines represent again the load transfer of the individual pistons. It can also be seen, that depending on the direction of movement of the pistons, different pressure differences are required to generating the torque. This can be explained by the manufacturing accuracies of the individual components and undesired relative movements of individual components relative to one another of the prototype. Figure 9 shows the required torque and the actual torque during an entire crane boom rotation. From this, the efficiency of the motor can be calculated with 65%. Further, the torque reversal during downward travel can be seen in a range of 120°. This is also a safety issue, because the boom can not fall down when the torque of the motor is positive. When the motor is operating the control unit prevents the boom to fall down, when the control unit is not working safety valves are needed.

Figure 7: Pressure difference of the ideal motor, source: own illustration

Figure 8: Pressure of the cylinder chambers of the real motor, source: own illustration

Figure 9: Torque profile of the real motor over 360°, source: own illustration
4 Summary and Conclusion

The consistent use of hydrostatic rotary motors in the joints of the booms of concrete pumps is with existing solutions of hydrostatic rotary motors not or only with limitations possible. Either there are limitations due to the swivel angle which can be achieved or the power weight represents the limiting factor.

The presented direct drive hydrostatic rotary motor can surpass existing solutions both with the swivel angle to be achieved and with the power weight. The combination of plane toothing and an integrated hydraulic cylinder makes the construction compact and light in weight. At the same time, however, high torques can be generated via the plane toothing.

The production of the components as well as the exact synchronization of the movement sequence certainly represent a challenge in the implementation. That is nevertheless possible to integrate the concept into automotive concrete pumps is shown with the presented prototype.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta p$</td>
<td>Pressure difference</td>
<td>[N/m²]</td>
</tr>
<tr>
<td>$A_{\text{kubem}}$</td>
<td>Piston area</td>
<td>[m²]</td>
</tr>
<tr>
<td>$r_{\text{effe}}$</td>
<td>Effective radius to generate the torque</td>
<td>[m]</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>Angle of the teeth</td>
<td>[grad]</td>
</tr>
<tr>
<td>$F_{\text{kubem,axial}}$</td>
<td>Axial force of the piston</td>
<td>[N]</td>
</tr>
<tr>
<td>$F_{\text{kubem,radial}}$</td>
<td>Radial force of the piston</td>
<td>[N]</td>
</tr>
<tr>
<td>$v_{\text{kubem,n}}$</td>
<td>Velocity of the piston n</td>
<td>[m/s]</td>
</tr>
<tr>
<td>$M_T$</td>
<td>Torque of the motor</td>
<td>[Nm]</td>
</tr>
</tbody>
</table>

References

/7/ Holmes, B.: Direct drive hydraulic motors. Engineering (London), v 236, n 1, p 16-17, Jan 1995
Performance Investigation of a Hydro-pneumatic type Accumulator used in a Hydrostatic Drive System of Off-road Vehicles

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The performance characteristics of a hydro-pneumatic type accumulator on the responses of the hydrostatic drive system are studied in this article. The physical system considered for the analysis consists of fixed displacement pump, hydro-motor coupled with a loading unit and an accumulator. By varying the capacity and precharge pressure of the hydro-pneumatic accumulator and load torque on the hydro-motor, the performance behaviour of the accumulator is determined. In MATLAB/Simulink® environment, the simulation studies are made. By comparing the simulation results with the test data, the model is validated. The studies made in this article may be useful for the proper selection of accumulators in typical mining equipment.

Keywords: Hydro-pneumatic type accumulator, MATLAB/Simulink® environment, performance characteristics, hydrostatic drive.

Target audience: Mobile Hydraulics, Mining and allied Industry, Design Process

1 Introduction

Efficient performance and energy saving in construction and mining machinery has become a preeminent issue due to the increase in fuel price and increasing demand of production. Hydraulic systems are predominantly used in several industrial applications and indispensable for mobile equipment used in mining operations.

Commonly, in a typical working cycle of mining equipment the potential energy and the kinetic energy are dissipated in form of heat. So it is required to make maximum use of regenerative energy for further improvement of fuel consumption and also to ensure higher system control performance. One of the possible solutions is the incorporation of hydraulic accumulators in a hydraulic main.

Some of the significant research works made in this area in recent past are discussed here. An energy regeneration system in hydraulic forklift trucks has been studied, concentrating on energy recovery in the main lift system with electric motor and batteries, and resulting in improved energy efficiency but shorter lifetime of components /1/. An energy recovery system with a hydraulic accumulator that could save and restore energy in a crane’s hydraulic system has also been studied, and it was found that the potential energy of the crane and load can be saved in the form of hydraulic energy and reutilized /2/. A speed control system of a variable voltage variable frequency hydraulic elevator with a pressure accumulator has been studied, and shown to have higher efficiency compared with a hydraulic elevator without a pressure accumulator /3/. The hydraulic accumulator is one of the hydraulic elements of the system that will be used to reduce pressure and speed pulsation inside it; thus, the selection and workflow modelling have a significant influence on the stability of the entire system. For proper selection of accumulators, the understanding of its dynamic phenomena is essential. This paper deals with the issue of understanding the dynamic phenomena in bladder- type accumulators by controlling its various performance parameters. The analysis has been done using two different sizes of accumulators. Using the mathematical model, MATLAB/Simulink® environment and DSH® software, simulation model of the physical system is made. By varying the capacity and precharge pressure of the hydraulic accumulator and load torque using the pump loading configuration on the hydro-motor, the performance behaviour of the accumulator is determined. The simulation results are verified with the experimental test data. Using the validated model, the parametric studies are also made to rationalize the system design. The studies made in this article may be useful for the proper selection of accumulators with respect to its physical attributes for use in hydrostatic drive system in typical mining equipment.

1.1 Principle of Operation of a Hydro-pneumatic Accumulator

The hydro-pneumatic type accumulator consists of a synthetic polymer rubber bladder like chloroprene, nitrite, etc. inside a metal (steel) shell. The bladder is filled with compressed gas. A poppet valve located at the discharge port closes the port when accumulator is completely discharged that keeps the bladder from getting out into the system. The operation of a bladder type hydraulic accumulator is explained below. Figure 1 shows the schematic representation of the working of a hydraulic accumulator.

![Schematic representation of a hydraulic accumulator](image)

Figure 1: Schematic representation of the working of a bladder type hydraulic accumulator.

Referring to Figure 1, $P_P$ = Precharge pressure, $P_1$ = Maximum working pressure, $P_2$ = Minimum working pressure, $V_1$ = Volume of the fluid chamber during Precharged condition, $V_2$ = Volume of the fluid chamber after charging, $V_3$ = Volume of the fluid chamber after discharging.

The bladder of the accumulator and the accumulator shell, both have nearly the same volume when precharged with Nitrogen gas ($P_P$) to a predetermined pressure. The pressurized fluid supplied by the pump cannot enter the accumulator shell until the system pressure exceeds the precharge pressure of the gas in the accumulator bladder. As the system pressure increases above the precharge pressure, the fluid enters the accumulator and compresses the bladder until gas and system pressure become stabilized at $P_1$, which is typically the set pressure of main PRV (Pressure Relief Valve) of the hydraulic set-up or maximum working pressure of the hydraulic system. The selection of the optimum size of the accumulator as the only energy source for a plant for a definite time interval. Puddu and Paderi /10/ have investigated the discrepancies in the thermal behaviour of ideal and real gases to determine their effects on the processes of expansion and compression of an accumulator. The performance characteristics of a hydro-pneumatic type accumulator on the responses of the hydrostatic drive used in off-road vehicles are studied in this article. The efficiency of a hydrostatic drive system depends greatly on the load conditions, showing significantly low efficiency under partial load conditions, which arises when either the desired velocity or torque is much less than its maximum value. Accumulators are used in the hydraulic system to reduce pressure pulsations, emergency operation, energy storage, surge absorption, potential energy regeneration, dampening vibrations and several other applications /11/. Energy is stored in the hydraulic accumulator by the volume of the hydraulic fluid that compresses the gas under pressure. The rate at which the compression and expansion of the gas takes place affects the gas state – which is defined by volume, pressure and temperature /12/. The hydraulic accumulator is one of the hydraulic elements of the system that will be used to reduce pressure and speed pulsation inside it; thus, the selection and workflow modelling have a significant influence on the stability of the entire system. For proper selection of accumulators, the understanding of its dynamic phenomena is essential. This paper deals with the issue of understanding the dynamic phenomena in bladder- type accumulators by controlling its various performance parameters. The analysis has been done using two different sizes of accumulators. Using the mathematical model, MATLAB/Simulink® environment and DSH® software, simulation model of the physical system is made. By varying the capacity and precharge pressure of the hydraulic accumulator and load torque using the pump loading configuration on the hydro-motor, the performance behaviour of the accumulator is determined. The simulation results are verified with the experimental test data. Using the validated model, the parametric studies are also made to rationalize the system design. The studies made in this article may be useful for the proper selection of accumulators with respect to its physical attributes for use in hydrostatic drive system in typical mining equipment.

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volume now stored in the accumulator becomes \( V_2 \). After the poppet valve is opened, the pressurized fluid of volume \( V_1 \) stored in the accumulator drives the hydro-motor. The system pressure reduces as the energy stored in the form of pressurized fluid is released to drive the hydro-motor; the accumulator bladder expands releasing the stored volume and the system pressure drops to \( P_2 \) which is typically the minimum operating pressure of the system to drive the hydro-motor. At this typical minimum operating pressure of the system, the volume of the pressurized fluid stored in the accumulator is \( V_1 \) which drives the hydro-motor at a constant pressure of \( P_2 \). After the fluid stored in the accumulator gets exhausted, the hydro-motor speed gradually falls to zero.

2 Experimental Test Set-up

The schematic representation of the experimental test set-up is shown in Figure 2 and its pictorial view is depicted in Figure 3.

![Figure 2: Schematic representation of the physical system incorporating hydro-pneumatic type accumulator.](image)

![Figure 3: Pictorial view of the experimental set-up showing front and rear view.](image)

A 7.5 kW electric motor (1) drives a fixed displacement vane pump (2) that supplies pressurized fluid to the accumulators (3) of capacities 20 l and 10 l through check valve (4). The pump flow is also supplied to the bent axis hydro-motor (5) through solenoid operated proportional flow control valve (6). By adjusting the command signal (from 0 – 10 V dc) given to the flow control valve, the pump flow supplied to the hydro-motor is varied; thereby the speed of the hydro-motor is controlled. The motor in-turn drives a fixed displacement pump (7) in the loading circuit that supplies flow to the sump through a PRV (8). By regulating the set pressure of the PRV, the load torque on the hydro-motor shaft is controlled. Before the pump supply is made available to the hydro-motor, both the accumulators were charged by opening their respective poppet valves and closing the proportional flow control valve. After they are fully charged, the poppet valves were switched off and the load torque on the poppet valves and unloading the main pump i.e. by switching off the unloading PRV (9) simultaneously, the flow was supplied to the hydro-motor from the accumulator.

The experiments were performed in a well-ventilated lab. In order to keep the viscosity of the oil constant with reasonable accuracy, the test set-up is fitted with appropriate oil cooler to maintain the temperature at 55 ± 2°C. The pressures, flow rates and the speed of the hydro-motor were measured through respective sensors and recorded through data logger /13/. To examine the repeatability of the system, the experiments were conducted several times before collecting the test data. Table 1 list the major components and instruments used in the test set-up.

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Items</th>
<th>Specifications</th>
<th>Make and Model</th>
</tr>
</thead>
<tbody>
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<td></td>
<td></td>
<td>Max. pressure: 150 bar</td>
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<tr>
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<td>Bladder Type Capacity: 20 l and 10 l</td>
<td>Parker Hannifin Corporation, USA;</td>
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<td></td>
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<td>3.</td>
<td>Hydro-Motor</td>
<td>Fixed Displacement Bent Axis</td>
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<td>Motor Displacement: 12 cc/rev</td>
<td>A2FM12/61WVBB040</td>
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<td>4.</td>
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<td>5.</td>
<td>Proportional PRV</td>
<td>Max. Pressure: 200 bar</td>
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<td></td>
<td>(Pilot Operated)</td>
<td>Max. Flow: 30 lpm</td>
<td>DBEE6-21/200YG24K31A1</td>
</tr>
</tbody>
</table>

Table 1: List of the major components and instruments used in the physical system.

3 System Modelling

Referring to Figure 2, the model of the test set-up is discussed in this section considering the actual features of the critical components used in the physical system. While developing the model, following assumptions are made:

- The detail dynamics of the pump, hydro-motor and the valves are ignored,
- The gas compression of the accumulator bladder is determined on the basis of the thermodynamic behaviour of ideal gases,
- The charging and discharging processes of accumulator are assumed to be polytropic,
- Losses in the accumulator due to heat transfer are ignored,
- Fluid compressibility is not taken into account. The fluid considered has Newtonian characteristics.

3.1 Mathematical Model

In order to perform a dynamic simulation of the system, the mathematical model of the system is developed defining the behaviour of the critical components.

1. Fixed-Displacement Pump. The pump flow rate is given by

\[
q_p = D_p \omega_p - k_{st} \Delta P_p
\]

where \( k_{st} = \frac{\pi r^4}{8 \eta_p} \)
2. Accumulator. Hydro-pneumatic type accumulators of capacities 20 litres and 10 litres are used in the physical system for analysis. The rate of change of accumulator absolute pressure is given by /14/:

$$\dot{P}_a = \frac{\rho}{\beta} \frac{\mathrm{d}V}{\mathrm{d}t}$$  \hspace{1cm} (3)

The accumulator is considered to be working in an adiabatic process. The value of the polytropic index is determined and discussed in Appendix I. The oil flow rate discharged by the accumulator is expressed as:

$$q_{acc} = -\frac{\rho}{\beta} \frac{\mathrm{d}V}{\mathrm{d}t}$$  \hspace{1cm} (4)

3. Hydro-motor and Loading system. Fixed displacement bent axis hydro-motor is used in the proposed system. The describing equations of the hydro-motor considered for the analysis are expressed below.

The inlet flow to the hydro-motor is expressed as:

$$q_M = \frac{D_M}{\pi} \frac{\mathrm{d}\theta_M}{\mathrm{d}t} + \frac{C_L}{\beta} \Delta P_L + \frac{V_M}{\beta} \frac{\mathrm{d}\Delta P_L}{\mathrm{d}t}$$  \hspace{1cm} (5)

The torque load on the hydro-motor is given by:

$$T_d = J_M \omega_M + B_M \omega_M + T_L$$  \hspace{1cm} (6)

where load torque $T_L$ is given by;

$$T_L = \frac{D_P L}{\beta} \Delta P_P$$  \hspace{1cm} (7)

The value of $B_M$ is determined by investigating the steady state characteristics of the hydro-motor as studied by N. Kumar et al. /15/.

4. Control Valves. With the movement of the valve spool due to the pressure acting on it, the valve port starts opening at its cracking/set pressure and at the full open pressure, the movement of the valve spool is arrested. The model of the control valves considers for both the laminar and turbulent flow states by analysing the Reynolds number (Re) and correlating its value with the critical Reynolds number ($Re_c$). The flow rate $q_{CV}$ across the control valves is determined by the following equations.

$$q_{CV} = \begin{cases} 
C_D \frac{A}{\rho} \sqrt{2} |\Delta P_{CV}| \sqrt{\frac{2}{\beta}} & \text{for } Re \geq Re_c \\
2 C_{Diam} \frac{A}{\rho} \frac{\beta}{\beta_0} \Delta P_{CV} & \text{else}
\end{cases}$$  \hspace{1cm} (8)

where $C_{Diam} = \left(\frac{4 L}{D_{CV}}\right)^2$  \hspace{1cm} (9)

3.2 Simulation Model

The proposed physical system is modelled in MATLAB/Simulink® environment using Sim-hydraulics blocks and DSH software. The model block consists of basic components like pump block connected with a variable angular velocity source, accumulator block, hydro-motor block and valve blocks. The output of sine wave is given to variable angular velocity source to introduce the pressure-surge in the system. A PRV is set across the main pump to relieve excess pressure of system. The upstream pressure comes in the form of pulsation which is sensed by pressure sensor. A flow sensor is also connected in system line to check flow ripple. Accumulator is installed between the pump and hydro-motor line to absorb pressure pulsation.

Referring to the physical system discussed in section 2, the DSH model and simulation model of the proposed system is shown in Figure 4 and Figure 5. The parametric values of the major components of the physical system are taken from the datasheet of the components specified in Table 1. The features pertinent to each component are explicitly detailed in the Sim-Hydraulics component library /16/.

4 Results and Discussions

The analysis is performed to investigate the performance behaviour of a hydro-pneumatic type accumulator used in hydrostatic drive of off-road vehicles on the responses of hydro-motor speed. Using the experimental set-up discussed in section 2, the experiments were performed. After the accumulators were fully charged at a particular precharge pressure, the cartridge valves were opened instantaneously to supply pressurized flow to the hydro-motor and the performance characteristics of accumulator were studied. Comparisons between simulation and experimental results are made for the dynamic performance of the accumulator discharge on the drive system at different load pressure using 10 litres and 20 litres capacity accumulators. The results are shown in Figures 6 and 7. The validation of the model is also made by comparing the simulation and experimental results for the accumulator discharge pressure at different resistive loads and capacities of the accumulator as shown in Figures 8 and 9.
The energy stored for a particular capacity of the accumulator depends on the effective bulk modulus of the fluid in the line connecting the accumulator and the hydro-motor (circuit). It is shown in Figures 6 and 7 that the hydro-motor drives the hydro-motor at a higher speed for the smaller load (50 bar set pressure of the accumulator) whereas for load given by 80 bar set pressure of PRV of the loading circuit, it starts from 759 rpm in case of 10\(\text{l}\) and 2257 rpm in case of 20\(\text{l}\) accumulator whereas for load given by 80 bar set pressure of PRV of the loading circuit, it starts from 759 rpm in case of 10\(\text{l}\) and 1008 rpm for 20\(\text{l}\) accumulator. Referring to Figure 6, in case of smaller load, by the time the speed of the hydro-motor decreases from 1750 rpm to 1208 rpm at 5.22s, the energy stored in the accumulator is almost exhausted and thereafter its speed falls abruptly in 2.78s. In case of higher load, the maximum speed of the hydro-motor starts from a lower value of 759 rpm. After the speed decreases from 759 rpm to 300 rpm in 7.9s, considerable amount of energy remains in the accumulator. The remaining stored energy drives the hydro-motor from 7.9s to 42.7s almost at a speed of 185 rpm for the constant load applied on it by the 80 bar pressure in the loading circuit. In case of 20\(\text{l}\) accumulator for the same loading conditions and precharge pressure, as the larger amount of energy is stored in the accumulator compared to 10\(\text{l}\) capacity accumulator, the running time of the hydro-motor increases. The deviation observed between the experimental and the simulation results in Figures 5 and 6 at lower hydro-motor speed may be due to the characteristics of the PRV of the loading circuit at low flow range. However, the simulation and experimental responses are very similar in nature. The percentage variation between the simulation and experimental results is within ±3% to 5%. Due to the minor fluctuations of the actual speed compared to the simulation results and the measured noise of the speed sensors, the smooth curves of the experimental responses could not be obtained. The Coulomb friction/ Stiction effect could not be realized that would have been expected at lower speed. However, the close agreement between the predicted and the experimental responses of the system validates the model.

Figures 6 and 7 compare the experimental and the simulation results for the hydro-motor speeds. The time delay \((t_d)\) shown in Figures 6 and 7 is the time required to open the cartridge valve. By comparing the simulation and experimental results for 10\(\text{l}\) accumulator shown in Figure 6, it is found that the peak time \(t_{peak}\) for the experimental result is 0.20s for 50 bar and 0.22s for 80 bar loading pressure which is higher than that of the simulation result \(t_{sim}\). This is due to the damping effects that are not considered in the model; like, seal friction of the hydro-motor. With the increase in the load resistance (50 bar to 80 bar set pressure) controlled through PRV of the loading circuit discussed in section 2, the steady state pressure of the physical system rises and the maximum speed of the hydro-motor falls from 1750 rpm to 759 rpm. For the identical load resistances, the hydro-motor speeds are also compared for 20\(\text{l}\) capacity of the accumulator and it is shown in Figure 7. With the opening of the cartridge valve, there is instantaneous flow supplied from the accumulator to the hydro-motor. Therefore, there is sudden rise in the speed of the hydro-motor. The overshoot of the hydro-motor speed depends on the effective bulk modulus of the fluid in the line connecting the accumulator and the hydro-motor as well as the load inertia on the motor shaft. Once the inertia is overcome, the hydro-motor speed decreases from its peak value following mostly the polytropic process of expansion of the gas in the accumulator. In case of 50 bar set pressure of PRV in the loading circuit, the discharge of the fluid from the accumulator drives the motor for the time up to 7.5s as shown in Figure 6. With the release of the fluid from the accumulator, its pressure reduces, the hydro-motor speed also decays till 5.22s and thereafter the speed falls to zero in 2.24s. Similar trend is observed for the load resistance given by 80 bar pressure of the loading circuit where the speed of the hydro-motor falls from its maximum value of 759 rpm to 185 rpm in 42.7s and thereafter the speed fall to zero in 2.54s.

The energy stored for a particular capacity of the accumulator was same for the same precharge pressure driving different loads. Therefore, in the event of the opening of the cartridge valve, the pressurized fluid of the accumulator drives the hydro-motor at a higher speed for the smaller load (50 bar set pressure of the loading circuit). It is shown in Figures 6 and 7 that the hydro-motor speed for smaller load starts from 1750 rpm in case of 10\(\text{l}\) and 2257 rpm in case of 20\(\text{l}\) accumulator whereas for load given by 80 bar set pressure of PRV of the loading circuit, it starts from 759 rpm in case of 10\(\text{l}\) and 1008 rpm for 20\(\text{l}\) accumulator. Referring to Figure 6, in case of smaller load, by the time the speed of the hydro-motor decreases from 1750 rpm to 1208 rpm at 5.22s, the energy stored in the accumulator is almost exhausted and thereafter its speed falls abruptly in 2.78s. In case of higher load, the maximum speed of the hydro-motor starts from a lower value of 759 rpm. After the speed decreases from 759 rpm to 300 rpm in 7.9s, considerable amount of energy remains in the accumulator. The remaining stored energy drives the hydro-motor from 7.9s to 42.7s almost at a speed of 185 rpm for the constant load applied on it by the 80 bar pressure in the loading circuit. In case of 20\(\text{l}\) accumulator for the same loading conditions and precharge pressure, as the larger amount of energy is stored in the accumulator compared to 10\(\text{l}\) capacity accumulator, the running time of the hydro-motor increases. The deviation observed between the experimental and the simulation results in Figures 5 and 6 at lower hydro-motor speed may be due to the characteristics of the PRV of the loading circuit at low flow range. However, the simulation and experimental responses are very similar in nature. The percentage variation between the simulation and experimental results is within ±3% to 5%. Due to the minor fluctuations of the actual speed compared to the simulation results and the measured noise of the speed sensors, the smooth curves of the experimental responses could not be obtained. The Coulomb friction/ Stiction effect could not be realized that would have been expected at lower speed. However, the close agreement between the predicted and the experimental responses of the system validates the model.
Using the validated model, the responses of the hydro-motor speed are also analysed at different precharge pressure of 10 litres and 20 litres capacity accumulators shown in Figures 10 and 11.

Figure 10: Characteristics of the hydro-motor speed at different precharge pressures of 10 litres accumulator.

Figure 11: Characteristics of the hydro-motor speed at different precharge pressures of 20 litres accumulator.

Figure 10 shows the characteristics of the hydro-motor speed for different precharge pressures at a particular load resistance provided by the 50 bar pressure of the loading circuit. Similar characteristics are obtained for the higher capacity of the accumulator for the same precharge pressure and load as shown in Figure 11. The characteristics curves indicate that with the decrease in precharge pressure of the accumulator, the hydro-motor running time increases. This is due to the fact that at lower precharge pressure or higher volume of accumulator, larger amount of energy is stored in the accumulator.

Also, by using DSH software, simulation graphics of the physical system is made and the effects of the variation of the accumulator parameters on the hydro-motor speed fluctuations have been observed. The results have been shown by the characteristics curves in Figure 13.

The DSH simulation program is used for the analysis of system dynamics, system revision, component selection, component development, fault diagnosis. Employing an interactive graphical environment and libraries of components, DSH software offers several interfaces to MATLAB and Simulink that enables seamlessly taking advantage of the power of the two simulation tools together. It performs a co-simulation with Simulink and exports the validated simulation model as Simulink functions.

The simulation analysis using the DSH program has been carried out to study the hydro-motor speed fluctuations caused by varying the critical parameters of the hydraulic accumulator such as nominal volume and pre-charge pressure. Figure 13 indicates that with the increase in the nominal volume of the accumulator, the percentage fluctuation of hydro-motor speed reduces. Similar trend has been observed with the variation in precharge pressure of the accumulator.

5 Summary and Conclusion

This paper analyses the characteristics of a hydro-pneumatic type accumulator on the responses of a hydraulic drive system typically used in off-road vehicles. In this respect, a typical open circuit hydrostatic system with accumulator used in mining vehicle is considered. The simulation model of the physical system has been made using MATLAB/Simulink® environment and DSH software. The model has been validated through experiments at different characteristics of the system. The analysis has been performed for two different sizes of accumulators at different loads as well as at different precharge pressures of the accumulator. Comparing the simulation results with the test data, it is observed that the variation is within ± 2% to 3% and therefore it is ascertained that the model fairly depicts the characteristics of the accumulator used in the hydrostatic drive. Using the validated model, the characteristics of the accumulator is also analysed for the variation of some of its critical parameters.

The following conclusions are drawn with respect to the observations made in the analysis

1. For a particular capacity of accumulator, at the same precharge pressure, the pressurized fluid stored in it drives the hydro-motor at a higher speed for a shorter period of time for the smaller load. With the increase in the capacity of the accumulator, the hydro-motor rotates for a longer period of time.
2. With the increase in the precharge pressure of the accumulator, less energy is stored in it which results in the decrease in decay time of the hydro-motor speed.
3. The small size accumulator shows the quicker response in minimizing the pressure surge as compared to the large size accumulators. However, the energy stored and the discharge characteristic of large size accumulator is much better as compared to the smaller accumulator.
4. Discharging of accumulator is poly-tropic process, for proposed setup it follows the equation \( PV^n = \text{Constant}. \)
5. Drive efficiency decreases when differential pressure across the flow control valve increases. System efficiency increases with the accumulator in the hydraulic system.

The optimum sized accumulator should be used at optimum pre-charge pressure (near about system operating pressure). The future scope of this work will be to analyse the energy stored in the accumulator owing to pressure surge which needs both simulation and experimentation work, to be carried out. The authors believe that the studies made in this article may be useful for selecting a proper size of accumulator for a given application.

6 Acknowledgements

Authors would like to acknowledge the staff members of Mining Machinery Engineering Department for their help in carrying out the research work.
Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\theta_m$</td>
<td>Angular displacement of the hydro-motor</td>
<td>[rad]</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Mass density of the fluid</td>
<td>[kg/m³]</td>
</tr>
<tr>
<td>$\nu$</td>
<td>Kinematic viscosity of the fluid</td>
<td>[m²/s]</td>
</tr>
<tr>
<td>$\omega_m$</td>
<td>Angular speed of the hydro-motor</td>
<td>[rad/s]</td>
</tr>
<tr>
<td>$\omega_p$</td>
<td>Angular speed of the pump</td>
<td>[rad/s]</td>
</tr>
</tbody>
</table>

References

3/ Bing, X., Jian, Y., Yang, H. Y., Comparison of energy-saving on the speed control of the VVVF hydraulic elevator with and without the pressure accumulator, In: Mechatronics, Vol. 15, No. 10, pp. 1159-1174, 2005
of the accumulator, pump is in unloaded condition. Hence, two boundary conditions exists at initial and final state of discharging of the accumulator.

\[ 2416.6 \text{ rpm} = 40.3 \text{ rev/s} = \omega_1 \]
\[ 900 \text{ rpm} = 15 \text{ rev/s} = \omega_2 \]

Therefore, Total revolution = 3362 cc/ (DM) = 3362 cc/12 cc/rev = 280.2 rev

So, from equation (10)

\[ \frac{\omega_1 \times t_1 - \omega_2 \times (t_1 + t_{ds})}{\gamma - 1} = 280.2 \text{ rev} \]
\[ 40.3 \times t_1 - 15 \times (t_1 + t_{ds}) = 280.2 \text{ rev} \] (11)

Also;

\[ \omega_1 \times t_1^\gamma = \omega_2 \times (t_1 + t_{ds})^\gamma \] (12)
\[ 40.3 \times t_1^\gamma = 15 \times (t_1 + t_{ds})^\gamma \] (13)

Substituting the value of \( t_1 \) from equation (13) in equation (11) and taking the value of \( n \) as 1.4 considering the discharging of accumulator as an adiabatic process, the value of \( t_{ds} \) is calculated as

\[ t_{ds} = 11.49 \text{ s} = 11.50 \text{ s} \]

From the experimental investigation, it has been found that the time required during the discharge of 20 litres capacity accumulator; when pump is in unloaded condition is 11.4 s and during charging; when pump is in loaded condition is 55.8 s. Substituting the value of \( t_{ds} \) in equation (13), the actual path of discharging of accumulator is obtained as;

\[ \gamma = 1.72 \]
Active damping improvement of the electrohydraulic control system with dual actuators for mobile machinery

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Low damping property of hydraulic systems has been a remarkably troublesome issue for a few decades. The poor damping with two actuators or more is still intractable and pendent due to the complex coupling effect of different loads. A decoupling compensator based on pump/valve combined control is proposed for the system with dual actuators for mobile machinery. Using decoupling control of different load branches, the coupling hydraulic circuit with dual cylinders is transformed into two separate single-cylinder circuits with dynamic compensation. Compound motion tests on a 2-ton hydraulic excavator were carried out. The results indicated that the proposed compensator reduced velocity and pressure oscillations under different working conditions.

Keywords: Decoupling compensation; Damping control; hydraulic system; mobile machinery

Target audience: Mobile Hydraulics, Control system

1 Introduction

Owing to flexible structure and low damping of hydraulic manipulators, the oscillation tendency is a remarkably evident issue for mobile machinery characterized by large size and heavy load, such as excavators, hydraulic cranes and turntable ladders. There are disadvantages of safety hazards, actuator wear and physical discomfort of the operators caused by system oscillations. To improve damping and reduce oscillations, active damping control is a classic and powerful method by adding an offset compensator into the valve or pump controller using oscillation signals (structure acceleration, cylinder velocity, and chamber pressure [1]), such as dynamic pressure feedback [2] and input shaping [3]. To obtain enough stability and fast response, linear analysis tools have been adopted to optimize the damping based on the mathematical model, such as pole placement [2] and LQR method [4]. Also, auto-tuning methods (e.g., extremum seeking method [5]) were proposed to obtain the optimized damping under different load conditions.

However, existing methods are almost used to improve the dynamic behaviour with only a hydraulic actuator, while the system with two or more actuators is seldom mentioned, as illustrated in Figure 1. The main challenge of damping improvement about two actuators is that different loads are coupled and interacted with each other for the hydraulic system of mobile machinery [6]. Generally, there are two kinds of coupling for the system with dual actuators. Firstly, the two load flow rates are coupled with each other since the actuators are supplied by only one pump, so the flow into one load increases theoretically if the flow into the other decreases. Secondly, the higher load pressure is transmitted into the pressure compensator with lower load, leading to pressure and velocity fluctuation of the actuator with lower load. Therefore, the dynamic behaviour of the other actuator is unpredictable if one actuator is damped by traditional active damping methods with pump or valve control.

In this paper, a decoupling compensator based on pump/valve combined control is proposed for active damping improvement of the electrohydraulic system with dual actuators. The remaining of this paper is structured as follows: The studied system and the problem dealt with are described in Section 2; the mathematical model and simplification are drawn in Section 3; the compensator design is introduced in Section 4; the compound motion test on a hydraulic excavator is shown in Section 5; finally, the conclusion and future work are given in Section 6.

2 Introduction

The studied system with two actuators is a typical electrohydraulic circuit for mobile machinery, as shown in Figure 1. The system consists of a prime mover, an electronically controlled pump, control valves, pressure compensators and hydraulic actuators. In contrast to industrial hydraulic systems, the main feature is the flow sharing function through the pressure compensator, which means that the distributed flow of the actuators only depends on the valve opening but not relates to the load. The higher pressure \( p_{in} = \max(p_{in1}, p_{in2}) \) is transmitted into the pilot chamber of the compensators, where \( p_{in1} \) and \( p_{in2} \) are the load pressures.

![Figure 1: Schematic of the hydraulic system with dual actuators](image)

Through force balance of the pressure compensator, the expression \( p_{in} = p_{in1} = p_{in2} \) is obtained if the compensator dynamic is neglected as well as its friction and viscous damping. The pressure drops of the two meter-in orifices are consistent with each other, which can be expressed as

\[ \Delta p_{in1} = p_{in1} - p_a = p_{in2} - \max(p_{in1}, p_{in2}) = \Delta p_{in2} \]  \hspace{1cm} (1)

\[ \Delta p_{in2} = p_{in2} - p_a = p_{in1} - \max(p_{in1}, p_{in2}) = \Delta p_{in1} \]  \hspace{1cm} (2)

The two control valves are assumed to be symmetrical with matched orifices. The nominal flow rates across the control valves can be written as

\[ Q_i(u_i) = C_iA_i(u_i) \frac{\Delta p_{in}}{\rho} \hspace{1cm} i = 1, 2 \]  \hspace{1cm} (3)

where \( C_i \) is the flow coefficient, \( \Delta p_{in} \) the nominal pressure drop, \( \rho \) the oil density. Since the pressure drops of the meter-in orifices are equal with each other, so the flow rates over the orifices are only related to the valve...
cross-sectional areas. Based on the flow matching concept [7], the pump displacement is calculated directly through the flow requirement of the valves. Then the pump control signal is drawn from Eq. (3) as:

\[ u_{p} = C_{p}A(u_{v})\sqrt{\frac{k_{p}p_{v}}{\rho}} + k_{p}p_{v} \]

(4)

Additionally, the energy efficiency can be improved by fully opening the valve with higher flow demand and proportionally enlarging the valve opening with smaller demand. Then, the valve control signals satisfy the following equations:

\[ \begin{align*}
   u_{i} & = u_{m} \\
   Q_{i}(u_{i}) & = \frac{Q_{i}(u_{m})}{Q_{i}(u_{i})}Q_{i}(u_{i})
\end{align*} \]

(5)

The flow distribution is consistent with the original one, while pressure losses over the valves are reduced. However, the existing literature [8-9] indicates that the damping ratio is reduced remarkably since the valve opening is maximized to decrease the throttling loss, leading to more oscillations and impacts compared with traditional control. As mentioned above, the damping performance can be improved by introducing the DPF method, but the main challenge for system damping is the coupling relationship between the two loads. Generally, there are two kinds of coupling for the system with dual actuators. Firstly, the two flow rates are coupled with each other since the actuators are supplied by only one pump, so the flow into one load increases theoretically if the flow into the other decreases. Secondly, the higher load pressure is transmitted into the pressure compensator with lower load, leading to pressure and velocity fluctuation of the actuator with lower load. Therefore, the dynamic behaviour of the other actuator is unpredictable if one actuator is damped by traditional DPF methods with pump or valve control. Thus, the damping problem for the system with two actuators is much more difficult to cope with. Fortunately, multiple input variables can be regulated to improve the damping performance, including the pump displacement and the valve openings of different load branches. To address the Multiple Input Multiple Output (MIMO) control issue, the mathematical model is firstly established and the decoupling compensator is introduced through pump/valve combined control afterwards.

3 System modelling

3.1 Mathematical model

The mathematical model is established involving the pump, the pipe, control valves and the actuators. If the relief valve is closed, the continuous equation of the pipe chamber between the pump and the valve can be expressed as

\[ p_{i}(s) = \frac{\beta_{i}}{q_{i}} \left[ q_{i}(s) - \sum_{n} q_{i}(s) \right] \]

(6)

The oil into the pilot chamber of the compensator is small enough to be neglected. In the linearization form, the flow rates across the meter-in orifices and the meter-out orifices can be expressed as

\[ q_{i}(s) = \dot{k}_{i}x_{i}(s) + \dot{k}_{i\epsilon}(p_{i}(s) - p_{e}(s)) \]

(7)

\[ q_{e}(s) = \dot{k}_{e}x_{e}(s) + \dot{k}_{e\epsilon}(p_{e}(s)) \]

(8)

Compared with the system dynamic, the valve dynamic is faster enough so that it can be neglected in the dynamic model. Then it can be considered that \( x_{i}(s) = \dot{k}_{i}u_{i}(s) \). The cylinder leakage is also neglected since it contributes to improving the damping ratio. The continuous equations of the cylinder chambers are expressed as

\[ p_{i}(s) = \frac{\beta_{i}}{q_{i}} \left[ q_{i}(s) - A_{i}V_{i}(s) \right] \]

(9)

\[ p_{o}(s) = \frac{\beta_{o}}{q_{o}} \left[ q_{o}(s) - q_{o}(s) - F_{o}(s) \right] \]

(10)

Neglecting the load stiffness, the force balance equation of the hydraulic pistons can be expressed as

\[ (m_{o} + b_{o})v_{o}(s) = A_{o}p_{o}(s) - A_{o}p_{o}(s) - F_{o}(s) \]

(11)

3.2 Model simplification

Considering Eqs. (6)-(11), it is seen that the control plant is a coupled high-order one. Several assumptions are made to simplify the compensator design as follows:

- The dynamic behavior of the pressure compensator is fast enough when the higher load pressure varies, so the compensator spool reaches the desired position promptly when the load pressure fluctuates;
- The pipe chamber between the pump and the valve is small enough so that its dynamic can be neglected. Actually, the pipe chamber is a design variable for the system so it can be selected as small as enough to improve the dynamic behavior;
- The load viscous damping and the pump leakage are neglected in the compensator since they contribute to enlarging the damping ratio of the whole system.

If the actuator is exerted by a passive load, the load flow across the two actuators can be expressed according to the assumptions and Eq. (2), as

\[ q_{i} = \frac{A(u_{v})}{A(u_{v}) + A(u_{v})} \sum_{n} q_{i} = \alpha q_{i} = \alpha q_{i} \]

(12)

\[ q_{o} = \frac{A(u_{v})}{A(u_{v}) + A(u_{v})} \sum_{n} q_{o} = (1 - \alpha)q_{o} = (1 - \alpha)q_{o} \]

(13)

where \( \alpha = A(u_{v})[A(u_{v}) + A(u_{v})] \), so it is seen that the flow rates are only related to the supplied flow rate and the cross-sectional areas of meter-in orifices. Thus, the hydraulic schematic is simplified as Figure 2. The valve with higher flow demand is fully open so that the energy consumption is reduced, and also the valve opening with lower flow demand is proportionally enlarged to make the flow distribution consistent with the original one.
4 Compensator design

From Figure 2, it is seen that the dynamic performance of the two actuators are both affected if regulating the pump displacement or anyone of the two valve openings. The basic damping concept is thereby proposed to decouple the hydraulic system as two separate damped circuits shown in Figure 3.

The hydraulic circuit with higher flow demand is controlled with a pump/valve combined compensator, and the other one with lower flow demand is controlled with a valve-based DPF compensator. The damping effects of the DPF compensators have been proven by the existing references, so damping improvement of two actuators can be obtained synchronously for the studied electrohydraulic system. According to the DPF concept, the steady-state pump displacement and valve opening in Figure 3 are consistent with that in Figure 2, so the steady-state velocity remains the same as that without compensation. Therefore, the control task is transformed into designing a pump/valve combined compensator in order to obtain the decoupled and damped circuits in Figure 4. The oscillation frequency usually falls within the range of 1–3 Hz in mobile machinery [1]. According to several simplified manipulations (detailed information is given in the equations (5)–(19) of ref. [10]), the capside pressure and the rodside pressure in the circuit with the maximum valve opening can be drawn as respectively:

\[ p_1(s) = \frac{\beta k}{k_{w12}}(V_{p1} + V_{e1}) \]

\[ p_2(s) = \frac{\beta k}{k_{w21}}(V_{p2} + V_{e2}) \]

Thus, the cylinder velocity with higher flow demand is expressed as Eq. (16). The cylinder velocity with the lower flow demand can be drawn as Eq. (17) from Eqs. (6)–(11).

\[ v_1(s) = \frac{\beta}{k}k_{w12}V_{e1} + A_{V1}V_{p1}k_{w12}k_{s1} - k_{w12}V_{e1}sF(s) \]

\[ v_2(s) = \frac{\beta}{k}k_{w21}V_{e2} + A_{V2}V_{p2}k_{w21}k_{s2} - k_{w21}V_{e2}sF(s) \]

There are three input variables in the electrohydraulic system, including input signals of the two control valves and the pump. The damping capability of the valve with higher flow demand is suppressed, since it should be fully open to reduce the pressure loss. A decoupling pump/valve compensator is thereby designed so that the two cylinders can be better damped to reduce oscillations. Assuming that \( \Delta A(u_{i1}) > \Delta A(u_{i2}) \), the displacement

\[ V_c \] and the valve signal \( u_c \) are regulated as \( V_c \) and \( u_c \) by the proposed compensator, while the valve signal \( u_{i2} \) remains the same. If the expression Eq. (18) is satisfied, it is seen that the system can be transformed as the one in Figure 4, in which the two circuits are damped individually by valve and pump compensation.

\[ \alpha V_c^\alpha = \alpha [u(s)]G_p(s) - k_{w12}G_p(s)(p_1) \]

\[ \beta A_c[V_{p1} + k_{w12}A_{V1}][1 - \alpha]u(s) V_{p1} + A_{V1}V_{p1}k_{w12}k_{s1} - k_{w12}V_{e1}sF(s) \]

\[ \beta A_c[V_{p2} + k_{w21}A_{V2}][1 - \alpha]u(s) V_{p2} + A_{V2}V_{p2}k_{w21}k_{s2} - k_{w21}V_{e2}sF(s) \]

Where \( G_p(s) = V_c(s) / u_c(s) \) describe the pump dynamic. In Eq. (18), the transfer functions \( G_p(s) \) and \( G_o(s) \) are the high-pass filters, which are expressed as

\[ G_p(s) = \frac{k x}{s / \omega_n + 1} \]

\[ G_o(s) = \frac{k x}{s / \omega_n + 1} \]

Where \( k_p \) is the control gain of the pump compensator, \( k_o \) the control gain of the valve compensator. The cutoff-off frequency \( \omega_n \) and \( \omega_n \) should be selected below the natural eigenfrequency of the hydraulic system. The cross-sectional area of the control valve with the compensator can be drawn from Eq. (18) as

\[ A(u_{i1}) = \frac{\alpha k}{k_{w12}}[u(s) - G_p(s)(p_1)] A(u_{i2}) A(u_{i1} + A(u_{i2}) \]

\[ A(u_{i1}) = \frac{\alpha k}{k_{w21}}[u(s) - G_p(s)(p_1)] A(u_{i2}) A(u_{i1} + A(u_{i2}) \]

\[ A(u_{i1}) = \frac{\alpha k}{k_{w21}}[u(s) - G_p(s)(p_1)] A(u_{i2}) A(u_{i1} + A(u_{i2}) \]

\[ A(u_{i1}) = \frac{\alpha k}{k_{w21}}[u(s) - G_p(s)(p_1)] A(u_{i2}) A(u_{i1} + A(u_{i2}) \]

Thus, the damping performance of dual actuators is improved synchronously with the decoupling compensator. Besides, it is found that \( G_p(s) = A_{u1}(s) = 0 \) in steady-state conditions, so the expressions \( A(u_{i1}) = A(u_{i2}) \) and \( V_c(s) = V_c(s) \) are obtained. Therefore, the steady performance is consistent with that without compensation.

5 Experimental results

5.1 Test rig

The test rig with a 2-ton hydraulic excavator [10] was used to validate the proposed decoupling compensator, as given in Figure 5. The compensator was implemented on the MATLAB xPC target platform and the MATLAB/ Simulink environment. The sampling frequency is selected as 2 kHz. The proportional valve (Type PVG 100 from Sauer Danfoss, Inc.) is used to distribute the flow into multi-actuators. The valve spool locates at the center, the negative maximum position and the positive maximum position under the signal 6 V, 3 V and 9 V, respectively. The nominal flow rate under the 2.0 MPa pressure drop is almost 50 L/min, and its flow
characteristic has been obtained by the manufacture. The effective bulk modulus of the oil was identified as 550 MPa from its definition. The main parameters of the test rig and the hydraulic excavator are listed in Table I.

<table>
<thead>
<tr>
<th>No.</th>
<th>Parameters</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Maximum displacement of the pump</td>
<td>45.6</td>
<td>mL/r</td>
</tr>
<tr>
<td>2</td>
<td>Rotational speed of the motor</td>
<td>1500</td>
<td>r/min</td>
</tr>
<tr>
<td>3</td>
<td>Diameter of the boom cylinder</td>
<td>0.07</td>
<td>m</td>
</tr>
<tr>
<td>4</td>
<td>Piston diameter of the boom cylinder</td>
<td>0.04</td>
<td>m</td>
</tr>
<tr>
<td>5</td>
<td>Diameter of the bucket cylinder</td>
<td>0.06</td>
<td>m</td>
</tr>
<tr>
<td>6</td>
<td>Piston diameter of the bucket cylinder</td>
<td>0.035</td>
<td>m</td>
</tr>
</tbody>
</table>

Table 1: Main parameters of the test rig

5.2 Boom/bucket compound motion test

The boom/bucket compound motion test is carried out to validate the proposed compensator under the heavy load with violent oscillation. The bucket firstly retracted at t=10s, and then the boom and the bucket retracted simultaneously at t=13s. As mentioned in the Introduction, there are many methods to determine the control parameters to obtain good dynamic. The purpose of this paper is to discuss and verify the decoupling method experimentally. The optimization of the control parameters could be carried out afterwards since the electrohydraulic system has been decoupled as two damping circuits separately. Therefore, the try-and-error method is adopted in this paper to select the parameters and verify the proposed compensator. The boom/bucket compound movement with and without the compensation is shown in Figure 4.

![Figure 4 Compound movement of the boom and the bucket](image_url)

The test results are shown in Figure 5. The boom cylinder is with higher flow demand when the two cylinders move simultaneously, so the pump signal and the boom valve are compensated to damp the hydraulic system. Generally, it is observed that more oscillations are generated when the boom cylinder starts to move. The reason can be easily found that the boom cylinder is with heavier load and poorer damping than the bucket. The discussion on the test results are listed as below.

- **System pressure:** When the boom started to move at t=13s, the system pressure rose up and oscillated. The pressure oscillation and the overshoot are reduced with the proposed compensation. The integrated time absolute error (ITAE) indicator is calculated between t=13-16s (0.538 MPa without compensation and 0.268 MPa with compensation) when the boom moves. Thus smoother build-up process could be obtained by the compensator. Moreover, the pressure reaches the steady stage later without the compensation.

- **Boom movement:** Due to heavier load and lower damping, the velocity and the pressure of the boom oscillates more violently than the bucket. The ITAR indicators of boom velocity with and without compensation are calculated as 0.006 m/s and 0.0037m/s, and also the boom pressure is less oscillated with the proposed compensator. Therefore, the cylinder motion with the compensation behaves with better dynamic performance than that without compensation.

- **Bucket movement:** It is seen that the start-up movement of the boom cylinder results in the fluctuations of the bucket velocity and pressure. The bucket velocity reduced nearly zero due to the instant short supply flow from the pump. The bucket movement was adversely affected by the boom motion due to the coupling effect. Moreover, pressure and velocity fluctuations are also reduced by the compensator.

The boom/bucket motion test results showed that the velocity vibrations and pressure peaks of the dual actuators are reduced simultaneously. Movements of the two cylinders are decoupled and damped theoretically so that the damping performance is improved. Therefore, smoother movement and better dynamic behaviour are achieved by the proposed decoupling compensator.

6 Summary and Conclusion

To cope with the low-damping issue of electrohydraulic control system with dual actuators, this paper proposed a decoupling compensator with pump/valve coordinated control. Through the proposed compensator, the control...
system is equivalent to two separate damped circuits: the actuator with higher flow demand is damped by pump-based compensation; and the actuator with lower flow demand is damped by valve-based compensation. The pump and valve compensators are designed according to the dynamic pressure feedback method, so the damping performance of dual actuators is improved synchronously. The compound motion test was carried out with a 2-ton hydraulic excavator. The boom/bucket motion test results shown that the velocity vibrations and pressure peaks of the actuators are reduced simultaneously, so dynamic performance of the two loads can be both improved. The decoupling control concept also opens up an opportunity to solve the poor damping issue of multiple actuators to reduce vibrations of mobile machinery. Future work will be focused on the auto-tuning decoupled methods to deal with parameter uncertainties so that the dynamic behaviour in multi-actuator systems can be optimized under resistive or overrunning load conditions.

7 Acknowledgements
This work was supported in part by the National Natural Science Foundation of China under Grant 51605050 and Grant 51475462, in part by the Fundamental Research Funds for the Central Universities under Grant 106112017CDJXY110002, and in part by the Chongqing Research Program of Basic Research and Frontier Technology under Grant cstc2016jcyjA0253, and in part by the Fundamental Research Funds for the Central Universities under Grant 106112017CDJXY110002.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A_{c}, A_{r})</td>
<td>Capside areas, rodside areas of hydraulic cylinders</td>
<td>m²</td>
</tr>
<tr>
<td>(A_{c}(\omega_{c}))</td>
<td>Cross-sectional area of the control valves</td>
<td>m²</td>
</tr>
<tr>
<td>(b_v)</td>
<td>Viscous damping coefficients of the loads</td>
<td>N·s/m</td>
</tr>
<tr>
<td>(F_e)</td>
<td>External force of the loads</td>
<td>N</td>
</tr>
<tr>
<td>(k_p)</td>
<td>Pump leakage</td>
<td>m³/(s·Pa)</td>
</tr>
<tr>
<td>(k_w)</td>
<td>Displacement gain</td>
<td>mL/(r·V)</td>
</tr>
<tr>
<td>(k_{vi}, k_{vo})</td>
<td>Flow-pressure gains of the meter-in orifices, the meter-out orifices</td>
<td>m³/(s·Pa)</td>
</tr>
<tr>
<td>(k_{vi}, k_{vo})</td>
<td>Flow gains of the meter-in orifices, the meter-out orifices</td>
<td>m³/s</td>
</tr>
<tr>
<td>(k_s)</td>
<td>Spool displacement gain</td>
<td>m/V</td>
</tr>
<tr>
<td>(m_a)</td>
<td>Load masses</td>
<td>kg</td>
</tr>
<tr>
<td>(n_e)</td>
<td>Rotational speed of the prime mover</td>
<td>r/min</td>
</tr>
<tr>
<td>(p_{ci}, p_{ri})</td>
<td>Capside pressure, Rodside pressure of the cylinders</td>
<td>MPa</td>
</tr>
<tr>
<td>(p_{ci}, p_{ri})</td>
<td>Chamber pressures between the meter-in orifices and the pressure compensators</td>
<td>MPa</td>
</tr>
<tr>
<td>(p_s)</td>
<td>System pressure</td>
<td>MPa</td>
</tr>
<tr>
<td>(q_{vi})</td>
<td>Flow rates across the meter-out orifices 1,2</td>
<td>L/min</td>
</tr>
<tr>
<td>(q_{vo})</td>
<td>Flow rates across the meter-out orifices 1,2</td>
<td>L/min</td>
</tr>
</tbody>
</table>

References

Accurate Control Method of Vane Direction Based on Pressure Difference Feedback in Active Yaw System for Wind Turbines


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In this paper, an active yaw system with valve-controlled hydraulic motor is designed. Correspondingly, the accurate control method of vane direction based on pressure difference feedback is presented. Then the simulation analysis is conducted in AMESim®. The simulation results show that the control method presented in this paper is efficient. Moreover, the control accuracy can be improved by decreasing the friction torque or adding a friction compensation link into the controller. At last, an experimental platform is built to verify the feasibility of the control method presented. The achievements provide theoretical and practical guidance for the design of wind turbine active hydraulic yaw systems.

Keywords: Wind turbines, active yaw, differential pressure feedback, accurate control method of vane direction

Target audience: Energy Industry, Wind Generator, Hydraulics

1 Introduction

The wind turbines yaw systems, including passive and active systems, can align the vane direction to the wind direction rapidly and steadily, which improves the generating efficiency. The passive yaw system is only adopted on small off-grid wind turbines. Although having a complicated structure, the active yaw system makes the yaw process controllable. So it has been widely used [1,2]. Usually, the motors are used as the driving unit of active yaw systems. Besides, the hydraulic cylinders and hydraulic motors can also be adopted.

Because of the varying wind direction, there exists some errors in the anemoscope detection results, which makes the active yaw system difficult to control the rotation position of engine room. That reduces the utilization rate of wind energy and exerts asymmetric force on symmetric vanes of wind turbines, which results in engine room’s vibration and vane fatigue [3,4].

To eliminate the negative influence from the error of anemoscopes, the traditional method is to introduce power control within small control angles (usually within 15 degrees). Though a better control effect can be achieved, the power control method needs a complicated control strategy, which increases the difficulty in system design and control [5]. Aimed at the deficiency of traditional methods, an active yaw system with valve-controlled hydraulic motor is designed. Correspondingly, the accurate control method of vane direction based on pressure difference feedback is presented. Then the simulation analysis and experimental verification are conducted. It provides theoretical and practical guidance for the design of active hydraulic yaw systems and accurate control methods of vane direction.

2 Hydraulic design of the active yaw system

2.1 Load calculation of the active yaw system

When wind blows the vane, the kinetic energy of wind per unit of time can be expressed as follows:

\[ P_r = \frac{1}{2} m v^2 \]

where, \( m \) —— wind mass flow (kg/s),
\( v \) —— wind speed (m/s).

The wind mass flow per unit of time can be expressed as follows:

\[ m = \rho A \bar{v} \]

where, \( \rho \) —— air density (kg/m³)
\( A \) —— swept area of the wind wheel in one rotation (m²)

The energy transferred to wind wheel per unit of time can be expressed as follows:

\[ P_T = C_p P_r \]

where, \( C_p \) —— wind energy efficiency of wind wheel

The average pressure on the swept area of wind wheel is expressed as follows:

\[ P_H = \frac{P_T}{A \bar{v}} \]

Combine equation (1)-(4), the following result can be obtained:

\[ P_n = \frac{1}{2} C_p \rho \bar{v} A \]

The wind blows the vane with a certain angle, the force diagram of wind wheel is shown in Figure1.

\[ \text{Figure 1: Force diagram of wind wheel.} \]

The torque exerted on the wind wheel tower can be reached:

\[ M_n = F_y A \theta = \frac{P_n A \cos \theta \sin \theta}{\ell_n} = \frac{1}{2} P_n A L_2 \sin \theta \]

where, \( L_2 \) —— friction torque (Nm);
\( M_v \) —— viscous torque (Nm);
\( I_m \) —— inertia moment on the axis of wind wheel tower (m⁴);
\( \alpha \) —— angle accelerate of cabin while yawing (rad/s²);
$M_e$ — other torques during yawing, including the eccentric torque resulted from aerodynamic force and the torque caused by gust load (Nm)

2.2 Hydraulic principles of active yaw system

The active yaw control can align the vane direction to the wind direction automatically when the wind speed is available and make the vane direction vertical to the wind when the wind speed is unavailable. Besides, the function of untwisting the cables automatically is also required to make wind turbines operate steadily and efficiently. The relative parameters of 850kW wind turbine is shown in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated generation power</td>
<td>$P_r$</td>
<td>850</td>
<td>kW</td>
</tr>
<tr>
<td>Rated wind speed</td>
<td>$v_r$</td>
<td>13</td>
<td>m/s</td>
</tr>
<tr>
<td>Best aerodynamic coefficient</td>
<td>$C_f$ max</td>
<td>0.4496</td>
<td></td>
</tr>
<tr>
<td>Swept area of vanes and wind wheel hub</td>
<td>$A$</td>
<td>1559.9</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Vertical distance between the center of Tower and wind wheel</td>
<td>$L_{WH}$</td>
<td>3.177</td>
<td>m</td>
</tr>
</tbody>
</table>

Table 1: Parameters of 850kW wind turbine.

According to the load calculation result and the requirement of active yaw, a hydraulic system is designed according to the parameters of 850kW wind turbine. The system principle diagram is shown in Figure 2.

In parallel connection, four hydraulic motors (12) with low speed and big torque can provide enough torque to rotate the cabin through internal-gearred ring (16). Energy accumulator (7) provides the system with hydraulic power. The start and stop of the motor are controlled by the pressure relay which comes after the energy accumulator (7). This can realize the oil supply for the energy accumulator. Relief valve (17) is used to ensure the safety and check valve (10) is used to avoid the lower pressure cavity from sucking air.

To achieve position closed loop control, the yaw system uses proportional valve-controlled hydraulic motors. According to the error between coder (15) and the detective values of anemoscope (18), the opening gap of proportional valve (9) is controlled to align the vane direction to the wind direction rapidly. The system will untwist the cables automatically when counter (17) give the corresponding signal; when the wind speed detected by anemoscope (18) is beyond the available scale, the system will stop generating.

If there is a certain nonzero angle between vane direction and wind direction, the force on the swept plane of vane will lead to a pressure difference between the two ends of hydraulic motors. According to the pressure difference between pipe L1 and pipe L2 detected by pressure sensor (14), the proportional valve (9) is controlled to align the vane direction accurately.

3 The simulation model of active yaw hydraulic system

3.1 AMESim® simulation model of hydraulic system

According to the hydraulic principle diagram, a simulation model shown in Figure 3 is built in AMESim® software.

![Figure 3: AMESim® simulation model of active yaw hydraulic system.](image)

1. motor 2. fixed displacement pump 3. check valve 4 energy accumulator 5 proportional direction valve 6. pressure sensor 7. hydraulic motor 8. relief valve 9. tank

3.2 AMESim® simulation model of load

According to the torque on wind wheel tower exerted by wind, the load simulation model of the system is built in AMESim® with the consideration of the gear transmission ratio, system inertia, friction and viscous damping etc. The model is shown in Figure 4.
Parameter | Value
--- | ---
Transmission ratio (actual value) | 16.7
Inertia torque of cabin and axis of wind wheel tower (calculated from the 3D model) | $8.96 \times 10^5$ kgm$^2$
The mass of vane and wheel hub (calculated from the 3D model) | $1.14 \times 10^4$ kg
Total mass of wind wheel and cabin (calculated from the 3D model) | $4.5 \times 10^4$ kg
Motor speed | 990 r/min
Pump displacement | 4ml/r
Rate volume of energy accumulator | 30 L
Opening pressure of check valve | 0.15 MPa
Rate flow of proportional direction valve | 8 L/min
Displacement of hydraulic motor | 940 ml/r
System pressure | 25 MPa

Parameters of pipes | Estimated according to the actual situation
oil | Software default

**Table 2: Parameters of AMESim® Simulation model.**

---

3.3 AMESim® model of accurate vane direction control

The accurate vane direction control is achieved based on the detection of the pressure difference between both ends of the hydraulic motor. An inertia link is designed to eliminate the influence from the medium-high frequency noise of pressure signals. The dead zone is designed to stop the vane direction control within the permissible error. PID control method is adopted in the system. Meanwhile, the amplitude limitation of the control signals and the dead zone of proportional direction valve are also considered. The AMESim® simulation model of accurate vane direction control is shown in Figure 5.

3.4 AMESim® simulation model of the whole system

In the simulation model, the friction torque is simplified to a constant. Some factors such as the extra eccentric torque, the gust torque and the leakage of hydraulic motors are also ignored. The simulation sampling time is 10ms and the simulation duration is 20s. Other parameters are shown in Table 2.

4 Simulation analysis of accurate vane direction control

4.1 The influence of the angle between wind and vane on system characteristics

The friction torque during yaw is estimated as 5000N [6]. The initial angle between vane direction and wind direction is zero. Then the step change signals, which are set as 5, 10 and 15 degrees, are input into the system respectively. Using the accurate control method of vane direction, the variation curves of the pressure in both cavities of hydraulic motors and the angle between vane direction and wind direction are achieved, which are shown in Figure 7 and Figure 8.
The angle between wind wheel direction and wind direction/degree

![Figure 7: Pressure curves of hydraulic motor cavities.](image)

Figure 7: Pressure curves of hydraulic motor cavities.

![Figure 8: Angle curves between wind and wind wheel.](image)

Figure 8: Angle curves between wind and wind wheel.

The following things can be seen from the curves. Because of the angle between vane direction and wind direction, the inclined wind flow exerts a torque on engine room, which results in the pressure difference between hydraulic motor’s two cavities. The larger the angle is, the bigger the pressure difference is. The vane direction control method based on pressure difference feedback is applied to control the pressure difference between hydraulic motor’s two cavities by adjusting the opening gap of the proportional valve. The less pressure difference means the smaller angle error between vane direction and wind direction, which achieves the accurate vane direction control.

Moreover, when the step changes of the initial angle are 5, 10 and 15 degrees, the adjust time of vane direction is 5.1s, 6s and 7s respectively. It meets the performance requirement of control speed.

4.2 The influence of engine room’s friction torque on system control accuracy

The pressure difference between hydraulic motor’s two cavities resulted from friction torque leads to system overshoot. So simulations with friction torques of 7000Nm, 5000Nm and 3000Nm are conducted, from which the curves of the angle between wind wheel direction and wind direction are obtained. It can be seen from the curves in Figure 9 that friction torque has great effect on vane direction control accuracy. The smaller the friction torque is, the higher the control accuracy is.

![Figure 9: Angle curves between wind and wind wheel.](image)

Figure 9: Angle curves between wind and wind wheel.

4.3 Compensation control of friction torque

The friction torque is estimated as 5000Nm. The influence on the pressure of hydraulic motor’s two cavities is calculated. The friction torque direction is confirmed by detecting the rotation direction of engine room. The friction compensation link is added between dead zone link and PID controller. The simulation model of friction compensation link is shown in Figure 10.

![Figure 10: AMESim® simulation model of friction torque compensation.](image)

Figure 10: AMESim® simulation model of friction torque compensation.

Conduct the simulation with friction compensation link, the angle curves shown in Figure 11 can be achieved. It can be seen from the curves that the friction compensation link can eliminate the steady-state error and improve the system accuracy.
5 Simulation experiments of accurate vane direction control

A simulation experiment platform of 850kW hydraulic wind turbine yaw system is built to verify the feasibility of the vane direction control strategy.

5.1 Hardware composition of simulation experimental platform

The experimental platform is composed of load simulation system, yaw system, control system and detection system. The overall structure is shown in Figure 12.

5.1.1 Detection system of wind turbine yaw simulation

The wind direction is decided by the input signal. The vane direction is detected by angle sensor. Driven by wind, the wind wheel generates a yaw load, which is detected by the torque sensor. The pressure of both ends of hydraulic motor is detected by pressure transmitter. The adopted sensors are shown in Figure 13.

5.1.2 Hydraulic system of experimental platform

The yaw simulation hydraulic system is composed of a variable pump, a constant delivery pump with low speed and large torque, hydraulic control valves, a control system, detection components, valve blocks and pipes. Its semi-physical simulation experimental platform is shown as Figure 14.

5.2 Experimental scheme of accurate vane direction control

On the yaw simulation experimental platform, when wind direction changes a small angle, the pressure difference feedback principle is used to control the vane direction. The specific step is expressed as follows:

(1) Different signals of small angles are input into the system. Then a yaw torque is generated and it produces the pressure in both ends of hydraulic motors.

(2) The pressure is output to computer through pressure transmitter. Then the pressure difference between both ends of hydraulic motors is calculated. The difference control signal is achieved by comparing that pressure
difference with the normal difference without wind. That signal is input into the control system to align the accurate control on vane direction.

5.3 Experiments of accurate vane direction control simulation
The system pressure is set to 10MPa. When the input angle signal is 10 degrees and 15 degrees, the yaw angle curves of the simulation yaw system are shown in Figure 16.

In Figure 16, there is a step change of wind direction at 2s. By using pressure difference feedback control method, the yaw process lasts about 13s. The steady-state angle error meets the accuracy requirement. The accurate control of vane direction is achieved.

6 Conclusions
1. An accurate vane direction control method is presented and it is verified to be efficient through simulation and experiments.
2. The larger angle between vane direction and wind direction brings the larger pressure difference in two cavities of hydraulic motor and longer yaw time.
3. Engine room’s friction torque has great effect on control accuracy. The smaller the friction is, the higher the control accuracy is. The friction torque can be reduced by improving the lubrication of yaw bearings.
4. The pressure difference signal in friction compensation link can be obtained by measuring the actual friction torque in wind turbines. An appropriate friction compensation torque can greatly improve the system accurate.
5. Based on the principle of pressure difference feedback, the accurate vane direction control strategy is verified to be feasible and efficient.

References
This article presents an innovative technology of energy management for a conventional hydrostatic-split power transmission (CH-SPT) system used in front end loader (FEL). A fuel efficient controller and a DC generator are additionally connected in parallel with the load shaft of the drive to prevent the engine and the major hydraulic components from over-loading or under-loading conditions. Detailed simulation model of the system, so called Regenerative Hydrostatic-Split Power Transmission (RH-SPT) system is made in the MATLAB®/Simscape environment. The performance analysis and the fuel consumption of the RH-SPT drive is compared with that of the CH-SPT drive through simulation. It is observed that with increase in 10% fuel consumption, the electric power regeneration through the DC generator increases by 21% of maximum power generated in CH-SPT drive.

Keywords: Fuel consumption, Energy Management, Energy Regeneration, Regenerative Hydraulic-Split Power Transmission (RH-SPT) Drive, PID controller, BSFC.

Target audience: Mobile Hydraulics, Mining Industry, Design Process

1 Introduction

Hydrostatic Transmission (HST) drive powered by a prime mover can be used widely in stationary as well as mobile equipment, because of their higher power to weight ratio, better reliability and easy control. In mobile hydraulic transmission system, the engine acts as prime mover, where the wheels are connected to the hydraulic motor shaft through the gear box. The torque on the wheels varies according to the profile of external load demand. This variable external torque demand experienced by the HST drive in turn applies variable load on the engine, which deviates the engine from its efficient working zone. This results in increased energy loss and causes increased fuel consumption during high as well as low-load demand. Hence, the energy management system using a suitable controller is needed to prevent the major components from under-loading/over-loading conditions. This may be obtained by constraining the fluctuating load demand or by modifying the drive using energy saving and regeneration concepts.

Numerous projects undertaken by researchers and major Heavy Earth Moving Machinery (HEMM) designers came up with innovative technologies to upgrade the development of efficient hydraulic types of machines /1/. Some of the research works relevant to the present article are described below:

Considering the energy reclamation benefits in the automotive segments /2-4/, many researchers have exaggerated the hybrid concept of regeneration in the HEMM, where the load change rate and its amplitude are considered essentially higher than that in the automotive vehicle. Cheong et al. /5/ studied the optimal design of power-split transmissions for hydraulic hybrid vehicles and observed that compound and input coupled power-split drives are comparatively compact and have better fuel economy as compared to the output coupled power-split drive. Another research work carried out by Kumar et. al /6/ developed an instantaneous optimization technique for power management strategy in the output-coupled power-split drive. They used two accumulators of different sizes for maximizing the energy capture and meeting the driving performance for some standard drive cycles.

Further research carried out by Tianliang et. al. /7/ compared the motor generator energy regeneration system (MGERS) and the accumulator motor generator energy regeneration system (AMGERS) with the conventional controlled system. They experimentally found that the dynamic response for AMGERS is almost same as that of the conventional controlled system and recovery efficiency is about 22% in the experimental set-up which can be improved up to 45% with proper selection of components. Wang et. al /8/ developed the potential energy regeneration system (PERS) and compound potential energy regeneration system that combines the regeneration device with throttle together. It was concluded that conventional PERS without throttle control has poor damping characteristics, but has higher efficiency than the compound PERS. They also suggested that some energy saving can be compromised for the sake of better control. Another research carried out by Xiao et. al /9/ developed an optimal design of a compound hybrid system using torque coupling and boom energy regeneration for an excavator. They compared the effectiveness of torque coupling and compound energy regeneration strategies and concluded that the compound hybrid system is 133% more effective than the system with torque coupling as far as fuel economy is concerned. Another study carried out by Backas et. al /10/ proposed an optimal fuel controller and tested on 5-ton hydraulically operated wheel loader. It was also concluded that the fuel consumption is reduced to 16.0% with the improved controller as compared to a conventional rule-based controller. In the study carried out by Vukovic et. al /11/, a comprehensive analysis of reducing fuel consumption in hydraulic excavators has been investigated. Using the Willans approximation, they categorised the fuel consumption of the excavator into fixed and variable component. It was concluded that increasing the hydraulic system efficiency only lower the variable fuel consumption. It was suggested that efficiency investigation and improvement of the HST drive considering engine operation and its parameters are more suitable as compared to consideration of the hydraulic components’ efficiency. Another work carried out by Schneider et. al /12/, developed and tested a green wheel loader and observed the increase in fuel economy by 10-15 % through efficient control of the drive.

In this respect, this article presents another type of innovative energy regeneration method in a split power HST system. The variation of load-torque with respect to engine’s optimum efficient torque is reduced by applying additional external load on the DC generator through a PID controller. This facilitates the engine to operate in its most efficient zone and generates additional electrical energy which can be used as a power source for lightening and air conditioning of operator’s cabin.

2 System Description

An experimental test set-up is fabricated which consists of a four-stroke, four-cylinder SI engine driving a variable displacement pump, two bent-axis hydro-motors, a loading pump unit, a DC generator for energy regeneration and a battery bank for energy storage. The schematic representation of RH-SPT closed-circuit HST drive is shown in figure 1.

Referring to figure 1, the four-stroke SI engine drives the main pump through the gear box (Gear ratio 1:6:1). The main pump supplies the pressurized hydraulic fluid through Direction Control Valves (DCV’s) to drive the two hydro-motors (M1 and M2) which are connected in parallel. The hydro-motors drive the loading pump and the DC generator through gear unit (gear ratio 1:1). The load on loading pump and the DC generator is applied through the proportional pressure relief valve (PPRV) and the battery, respectively. The DCV fitted in the supply and return line of the hydro-motor is used to disconnect the motor M2. The switching over from the double-motor mode to single-motor mode and vice-versa is performed with respect to the engine load torque through the controller. The controller also applies varying load on the DC generator that depends on the
difference between engine load torque and its optimal efficient torque. The charge pump and flushing valve are used to compensate the leakage flow and maintain the oil temperature about (50 ± 2)°C.

3 Modelling of the System

The simulation model of the proposed RH-SPT drive is considered for analysis is shown in figure 2. The model consists of an engine connected with a variable displacement bi-directional pump, four 2/2 DCVs, two fixed displacement bi-directional hydro-motors, the loading unit and the DC generator with the controller unit. The pressure and flow rate in the supply and return line of the drive are measured by using hydraulic pressure and hydraulic flow rate sensors, respectively.

Referring to figure 2, the loading unit generates torque demand equivalent to the load cycle of the FEL. The load cycle is considered with respect to the Y-cycle of the FEL /13/, which is shown in figure 3.

The fluctuating load demand of the loading unit of the simulation model causes the engine to operate in under-loading and over-loading conditions; which results in inefficient operation of the engine during under-loading condition /14/. In this respect, the fuel efficient controller is innovated to prevent the hydraulic components from over-pressurisation and the engine from over-loading by switching over from single-motor mode of operation to double-motor mode and additional loading of the DC generator. The energy generated through the DC generator is used for operating the electrical appliances. The detailed energy management strategy for the same is described in the subsequent section.

While analysing the simulation model of the RH-SPT drive, the system parameters are obtained from their product catalogue; whereas their response behaviour and the loss-coefficients are characterized experimentally. The responses of the engine and the DC generator are shown in figure 4. Also, the volumetric efficiency of the pump and the hydro-motor at varying speed and load pressure are characterised in figure 5. They are used in the simulation model of the system made in the MATLAB®/Simscape environment.

The specifications of major components used in MATLAB/Simhydraulics model are given in table:

<table>
<thead>
<tr>
<th>Major Components</th>
<th>Parametric values</th>
</tr>
</thead>
<tbody>
<tr>
<td>SI engine (4 stroke 4 cylinder)</td>
<td>Max. o/p power: 15 Kw</td>
</tr>
<tr>
<td></td>
<td>Max. speed: 3750 rpm</td>
</tr>
</tbody>
</table>
4 Comparing Energy Efficiency and Quantifying Losses during Single-Motor mode

Figure 6 shows the energy flow diagram of RH-SPT and CH-SPT drive for a same load cycle as shown in figure 3. More than 75% of energy contained in fuel is lost during combustion process which is 75.66% in RH-SPT drive and 77.7% in CH-SPT drive. This difference in loss percentage is due to combustion process is more efficient in RH-SPT drive than in CH-SPT drive. Approximately 12–13% of energy is consumed to overcome pump losses. Approximately, 11.5% of energy is lost due to pressure loss in control valves and other hydraulic losses such as pipe friction. The losses in hydro-motors accounts for 13.5% and 16.1% in RH-SPT and CH-SPT drive, respectively.

5 Control Strategy

The selection of an SI engine is done according to its maximum load torque and speed demand from the drive. Due to fluctuating load demand of the duty cycle, the power loss of the engine increases with its deviation from optimal efficient zone. The energy management strategy for prevention of over-loading and under-loading conditions is explained in subsequent subsection.

5.1 Energy management algorithm in the proposed RH-SPT drive

Flowchart for the energy management algorithm of the proposed RH-SPT drive is shown in figure 7. During the initial stage of the system, it is operated in single-motor mode only. According to the given load profile, the load torque on the engine (τel) increases. When the load-torque on the engine starts increasing above its optimum torque level (τopt) the drive shifts to double-motor mode of operation to prevent the engine from stalling. If the load torque on the engine (τel) is less than or equal its optimum torque level (τopt) the system will check the state of charge (SoC) of the battery. Depending upon the state of charge (SoC) of the battery, the load-torque of the engine continues to drive the load or the DC generator is loaded through the PID controller. The strategy for loading the DC generator through PID controller is shown in figure 7.

6 Results and Discussions

While interpreting the benefits of the RH-SPT drive over the CH-SPT drive, the simulation model is executed with respect to the varying load cycle of the FEL, shown in figure 3; such that the CH-SPT drive operates by switching-over condition whereas the RH -SPT drive operates by switching-over condition coupled with the generator loading. Switching over from single-motor mode to double-motor mode occurs at engine load torque higher than 52 Nm and equivalent load pressure 79.5 bar. Also, loading of the DC generator is carried out with respect to the torque demand at the pump of 83.4 Nm. The characteristic variation of differential pressure across the hydro-motor M1 (as it is connected throughout the duty cycle) and engine load torque for the CH-SPT and the RH-SPT drives are shown in figures 9(a) and 9(b), respectively.

Table 1: Specifications of the major components used in the RH-SPT model

<table>
<thead>
<tr>
<th>Component</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Displacement of Main Pump</td>
<td>28 cc/rev</td>
</tr>
<tr>
<td>Flow area of Direction control valve</td>
<td>5E-4 m²</td>
</tr>
<tr>
<td>Displacement of Hydro-motors M1 &amp; M2</td>
<td>16 cc/rev</td>
</tr>
<tr>
<td>Displacement of loading pump</td>
<td>28 cc/rev</td>
</tr>
<tr>
<td>Density of hydraulic oil</td>
<td>890.8 kg/m³</td>
</tr>
<tr>
<td>Viscosity of hydraulic oil</td>
<td>55.44 cSt</td>
</tr>
<tr>
<td>Bulk Modulus of hydraulic oil</td>
<td>1.459E9 N/m²</td>
</tr>
</tbody>
</table>

Figure 7: Energy management algorithm for the proposed RH-SPT drive

Referring to figure 8, the algebraic summation of the optimum load torque demand (τlpd) and actual load torque (τlp) at the pump shaft multiplied by transmission ratio of the drive provides the load torque demand from the DC generator (τlg). The error signal i.e. the difference between the demand torque from the DC generator (τlgd) and the load torque on generator (τlg) is the input for the PID controller. The controller minimizes the error by varying the field coil voltage (Vf) of the DC generator.

Figure 8: Signal flow of the controller for loading the DC generator

Figure 9: Comparison of motor pressure and engine load torque between the RH-SPT and the CH-SPT drive

Referring to figure 9, the observations are:
Switching over of the CH-SPT drive from single-motor mode to double-motor mode reduces the motor pressure and engine torque approximately by half. This is due to equal sharing of load torque by the hydro-motors in double-motor mode. Also, switching over from double-motor mode to single-motor mode of the drive increases the load pressure and the engine torque by twice. However, the magnitude of overshoot of pressure and torque during switching over from double-motor mode to single-motor mode is higher than the switching over from single-motor mode to double-motor mode. This is due to the sudden increase in load on the hydro-motor M1.

The motor-pressure and the engine load torque in the RH-SPT drive is higher than that of the CH-SPT drive. This is due to the additional load applied on the DC generator through the PID controller to operate the engine in optimal efficient zone.

Switching over of the RH-SPT drive from single-motor mode to double-motor mode and vice-versa exhibits almost similar variation of pressure and torque. However, the reduction in the motor-pressure and the engine load torque while switching over from single-motor mode to double-motor mode is less than half. This is due to the instantaneous increase in load on DC generator to prevent the engine from under-loading.

While operating the CH-SPT and the RH-SPT drives, the variation of engine performance characteristic in terms of its speed, the brake specific fuel consumption (bsfc), fuel consumption rate and power output are compared and are shown in figures 10(a), 10(b), 11(a) and 11(b), respectively.

Referring to figures 9(b), 11(a) and 11(b), it is observed that the power generated through the engine of the RH-SPT drive in single-motor mode increases by 21% (figure 9(b)) at the cost of 10% increase in fuel consumption (figure 11(a)). The increase in power output of the RH-SPT drive (figure 11(b)) used for electrical energy generation is found to be approximately 70% of the increase in engine power generated (figure 9(b)). This is due to various losses of the hydraulic components.

7 Conclusion

The present article deals with minimization of the engine load torque variation and prevention of over-loading/under-loading operations, in particular. In this respect, the RH-SPT drive coupled with the fuel efficient controller was proposed; such that the controller prevents the hydraulic components from over-pressure by switching over between the two modes of operations and the engine from under-load condition by applying additional load on the DC generator. The simulation model of the system was made on MATLAB, where the parametric values of the major components were obtained through steady-state investigations. It is observed that a marginal increase in fuel consumption by 10% occurs due to application of additional load on the generator per cycle, of which approximately 70% of the power from engine is used for electrical energy regeneration. It is also observed that the RH-SPT drive facilitates comparatively lesser fluctuations of pressure, engine torque and significant reduction of bsfc value. This assists in efficient operation of the hydraulic components as well as the engine and leads to modification of the engine load cycle without effecting the duty cycle of the transmission.

From the study, it is also observed that bsfc for RH-SPT system is lower than that of CH-SPT system which is desirable. The proposed innovative idea may be helpful to the engineers to design the construction equipment subjected to fluctuated load profile to operate the major components and the engine, in particular to operate in the efficient zone during the maximum span of the duty cycle. This will also assist in complete combustion of the fuel and may lead to reduction in CO in the exhaust gas. This additional advantage can be verified through experiments in near future. The energy stored in the battery bank can be used for lighting purpose and air-conditioning of the operator’s cabin.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>bsfc</td>
<td>Brake Specific Fuel Consumption of the Engine</td>
<td>g/kWh</td>
</tr>
</tbody>
</table>
### References


Investigation of the Potential of Different Cooling System Structures for Machine Tools

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In the current cooling system structure of machine tools a central fixed pump provides a constant cooling volume flow to cool all the components of the machine tool. The provided cooling volume flow does not match the temperature development of each component. This may lead to some of the components heating up while the other components are simultaneously being cooled. Due to these temperature differences, a thermo-elastic deformation of the machine structure occurs. This deformation is responsible for the displacement of the Tool Centre Point (TCP) of the machine tools. Consequently, the machine’s accuracy during the production process is reduced.

The main goal of this paper is to analyse the thermal behaviour of the current cooling system structure of two demonstration machines and to present a simulative study of new cooling system structures under consideration. The investigation of this research will examine the effectiveness as well as the temperature characteristics of the components of the new structures under consideration comparing them to the current cooling system structure in order to ensure a uniform temperature distribution of the machine tool at minimal energy consumption.

The results show that the new concepts have great potential in respect to better thermal behaviour and lower hydraulic power compared to the current cooling system structure. The simulation results show a more stable temperature profile of the components as well as a lower energy consumption of the cooling system.

Keywords: machine tool, thermo-elastic deformation, cooling system, energy consumption, decentralized system

Target audience: Stationary Hydraulics, Manufacturing Industry, Machine Tool Producer

1 Introduction

Power losses of the machine tool caused by the manufacturing process are converted into thermal energy. Due to the temperature fluctuation, a thermo-elastic deformation of the machine structure occurs. This deformation is responsible for the displacement of the TCP of machine tools. Consequently, the accuracy of the machine during the production process is reduced. The warmed-up components such as the rotary table, tool holder, linear guide rails etc. need to be cooled. Therefore, cooling system is installed to reduce the temperature fluctuation of the components. In order to reduce the thermo-elastic deformations that occur and to enhance the production quality it is necessary to minimize the heat input. Previous research projects in this area mainly focused on reducing the energy demand of the machine tool and its main drives, reducing the energy consumption by developing more efficient components, and control strategies [1, 2, 3]. However, the analysis of the cooling systems and the investigation of their thermal behaviour has not yet been carried out in detail. Therefore, a detailed analysis of the existing cooling system structures, their thermal behaviour and their influence on the deformation of the machine structure is necessary in order to ensure a uniform temperature distribution of the machine tools at minimal energy consumption. Previous research activities of this project, which were carried out by the authors and focused on two demonstration machines, showed that sufficient cooling capacity in the cooling system is available but that the cooling is insufficiently adjusted to the process and to the individual demand of the machine components [4, 5, 6]. In order to address this deficit, it is necessary to consider and analyse the potential of new structures for cooling systems.

The main target of this study is to highlight the latest project activity regarding the investigation of the thermal behaviour of the current cooling system structure and to present a simulative study of new cooling system structures based on the validated simulation models of the current cooling system structures of two demonstration machines. The research will help obtain information concerning the effectiveness of the new cooling system structures under consideration as well as the temperature characteristics of the components compared to the current cooling system structure.

2 Design of the cooling system structures of the demonstration machines

The main function of a cooling system is to provide the cooling media for the components or spots of the machine to dissipate the heat energy and to avoid high temperature fluctuations within the machine structure. This helps to reduce the thermo-elastic deformation and finally increases accuracy in the production process. Figure 1 illustrates the cooling system circuit of the demonstration machine type Scharmann DBF630 investigated first (machine 1). The cooling circuit consists of three components to be cooled, electrical cabinet 6, rotary table 4, and main spindle 7. Furthermore, a central fixed pump 14 provides the cooling medium (40 l/min at 5.5 bar) to the three components. The calculated hydraulic power (Q · Δp) of the pump is about 370W. Usually, the cooling medium used in cooling systems of machine tools is a mixture of water (60 % - 80 %) and Antifrygen® (40 % - 20 %). Moreover, the cooling unit 13 is placed directly into the return flow side and cools down the heated fluid to a set temperature. The function principle of the cooling unit (Counter-Clockwise Carnot Cycle Process or Clausius-Rankine-Process) is a two-point temperature-controlled refrigerator. It is turned on once the temperature overruns the upper threshold of the set temperature and is turned off when the cooling medium is cooled to the lower temperature set. The cooling of the electrical cabinet is carried out by an air heat exchanger 12. The main spindle 7 and the rotary table 4 are cooled directly by the cooling medium that flows through integrated cooling channels.

In contrast to machine 1, the cooling system of the second investigated demonstration machine type DU/M80 eVo linear (machine 2) shown in Figure 2 simultaneously cools 13 components. Here, a fixed displacement pump 1 supplies the cooling medium (45 l/min at 4.5 bar) to the motor spindle 2, all the axis drives (3, 5, 7, 8), the housing of the B and C axes 4 as well as the rails of X, Y, and Z (6, 9-12). The calculated hydraulic power of the pump is approximately 340 W. The electrical cabinet is not cooled directly by the cooling system but by a separate cooling unit. Moreover, the cooling unit 13 of machine 2’s cooling system is not integrated in the return flow as in machine 1; it is mounted directly to the tank. Furthermore, a three-way-valve 14 is placed into the return flow side.

Figure 1: Cooling system of DBF630 (machine 1) [4]
This valve is used as a diverting valve, so, with a defined setting, a part of the heated backflow is introduced directly to the inlet side of the pump, and the remaining fluid flows back to the tank /7/. The controller of the three-way valve adjusts the flow to the tank or to the inlet side of the pump so that the temperature on the pump inlet side always stays at 25°C.

Additionally, the convective heat transport through the cooling medium in the hydraulic hoses caused by the forced convection is taken into account. The corresponding heat transfer coefficient in Figure 3 is determined by the following equations /9/:

\[
Re = \frac{D_{h} \cdot V}{\nu} \quad \text{(5)}
\]

\[
Pr = \frac{c_{\text{fluid}} \cdot \rho \cdot V}{\lambda_{\text{fluid}}} \quad \text{(6)}
\]

\[
Nu = \frac{0.0235 \cdot (Re^{0.8} - 230) \cdot (1 + Pr^{0.5} - 0.8) \cdot [1 + \left( \frac{d_{o}}{D_{h}} \right)^{2}] \cdot \left( \frac{h}{\lambda_{\text{fluid}}} \right)^{0.34}} \quad \text{(7)}
\]

\[
\alpha_{\text{inside}} = \frac{h_{\text{fluid}} \cdot Nu}{L} \quad \text{(8)}
\]

Applying exemplarily equations (5) to (8) to the hydraulic hose from the flow valve to the motor spindle \((l = 5 \text{ m}; d_{i} = 9 \text{ mm}; d_{o} = 12 \text{ mm}, \text{volume flow} = 9.4 \text{ l/min}, \lambda_{\text{fluid}} = 0.443 \text{ W/m K}) of machine 2, a heat transfer number of \(\alpha_{\text{inside}} = 8.1 \text{ W/m K} \) is calculated. Furthermore, the free convection heat transfer at the outer surface (Figure 3) is considered. The related heat transfer coefficient is calculated by following equations /10/:

\[
\beta = \frac{1}{l} \quad \text{(9)}
\]

\[
Gr = \frac{g \cdot D^{3} \cdot (T_{w} - T_{a})}{\nu^{2}} \quad \text{(10)}
\]

\[
Ra = Gr \cdot Pr \quad \text{(11)}
\]

\[
Nu = \left[ 0.6 + \frac{0.387 \cdotRal^{1/6}}{(1 + (0.559/Pr)^{0.6})^{1/6}} \right] f r o m 10^{-5} < Ra < 10^{12} \text{ and } 0 < Pr < \infty \quad \text{(12)}
\]

\[
\alpha_{\text{outside}} = \frac{\lambda_{\text{outside}} \cdot Nu}{L} \quad \text{(13)}
\]

Regarding to equations (9) to (13) to the mentioned hose at an ambient temperature of 25°C, a heat transfer number of \(\alpha_{\text{outside}} = 4.7 \text{ (W/m² K)} \) is obtained. Moreover, the heat transfer through the heat conduction in the hose is determined by equation (14), for the exemplarily hose \(\lambda_{\text{con}} = 0.42 \text{ W/m K} \) amounts \(\alpha_{\text{con}} = 243 \text{ (W/m² K)} \). For other hoses the same method is used to calculate the inner, outer, and conductive heat transfer numbers.

\[
\alpha_{\text{con}} = \frac{2 \cdot \lambda_{\text{con}}}{d_{o} \cdot \ln \left( \frac{d_{o}}{d_{i}} \right)} \quad \text{for cylinder shape} \quad \text{(14)}
\]

A simulation model of the current cooling system structure for the two machines is developed based on the modelling methods and machine documentations. The simulation model exemplified by machine 2 in Figure 4 consists mainly of a pump, flow valves, hydraulic hoses, a cooling unit, and the 13 components to be cooled as heat sources. The Table 1 gives an overview about the most important model parameters for the simulation model. Hydraulic connections or hoses are modelled by hydraulic volumes and hydraulic resistances. The geometrical parameters, such as length, inner and outer diameter are taken directly from the machine documentation.

**Figure 2: Cooling system of DMU80 eVo linear (machine 2)**

**Figure 3: Heat transfer mechanisms at the hose**

\[
Re = \frac{D_{h} \cdot V}{\nu} \quad \text{(5)}
\]

\[
Pr = \frac{c_{\text{fluid}} \cdot \rho \cdot V}{\lambda_{\text{fluid}}} \quad \text{(6)}
\]

\[
Nu = \frac{0.0235 \cdot (Re^{0.8} - 230) \cdot (1 + Pr^{0.5} - 0.8) \cdot [1 + \left( \frac{d_{o}}{D_{h}} \right)^{2}] \cdot \left( \frac{h}{\lambda_{\text{fluid}}} \right)^{0.34}} \quad \text{(7)}
\]

\[
\alpha_{\text{inside}} = \frac{h_{\text{fluid}} \cdot Nu}{L} \quad \text{(8)}
\]

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The simulation models developed for machines 1 and 2 show a high accuracy of the current structure of the cooling system as well as for the development of process- and demand-oriented control strategies. The considered heat transport in Figure 3 by enforced and free convection as well as by heat conduction of the total hydraulic hoses amount to 45 W as shown in Figure 7. Compared to the performance of cooling units 4.5 kW, the heat transfer through the hydraulic pipes is low, about 1%. It could be neglected in the future. The investigation of two demonstration machines shows that the cooling system of each machine in the idle process requires about 12 % (machine 1) and 26 % (machine 2) of the total energy consumption of the machine tool /4, 12/. The proportion of the calculated hydraulic energy of the pump according to equation (16) is 27.2 % of 12 % of machine 1 and 26 % of 26% (machine 2), the remaining energy 72.8 % of 12 % and 81.5 % of 26 % is consumed by cooling unit, electrical motor of the hydraulic pump and the pump efficiency. The investigation also depicts that these cooling systems as currently structured do not cool the warmed-up component based on their drive B axis and spindle nut Z axis is higher than the outlet temperature. Other components, such as the motor spindle or the secondary part X axis, are cooled during the test process. In contrast to machine 1, the temperature characteristics of the components in machine 2 are not influenced directly by the state of the cooling unit, in spite of the two-point temperature control of the cooling unit. This can be traced back to the three-way-valve (Figure 2, pos.14) that is placed into the return flow side of the cooling system. The valve controls the suction flow to the pump so that the cooling medium always has a constant temperature of 25 °C ±0.1 °C as shown in Figure 6 e.

Figure 6: Comparison of temperature development simulation and measurement in the idle process of machine 2

- a) Motor spindle
- b) Drive B axis
- c) Spindle nut Z axis
- d) Secondary part X axis
- e) Pump inlet temp.
- f) Tank

The simulation models developed for machines 1 and 2 show a high accuracy of the thermal and hydraulic quantities of the components /13/. The simulation model is validated and thus be used for the improvement of the current structure of the cooling system as well as for the development of process- and demand-oriented control strategies. The considered heat transport in Figure 3 by enforced and free convection as well as by heat conduction of the total hydraulic hoses amount to 45 W as shown in Figure 7. Compared to the performance of cooling units 4.5 kW, the heat transfer through the hydraulic pipes is low, about 1%. It could be neglected in the future. The investigation of two demonstration machines shows that the cooling system of each machine in the idle process requires about 12 % (machine 1) and 26 % (machine 2) of the total energy consumption of the machine tool /4, 12/. The proportion of the calculated hydraulic energy of the pump according to equation (16) is 27.2 % of 12 % (machine 1) and 18.5 % of 26% (machine 2), the remaining energy 72.8 % of 12 % and 81.5 % of 26 % is consumed by cooling unit, electrical motor of the hydraulic pump and the pump efficiency. The investigation also depicts that these cooling systems as currently structured do not cool the warmed-up component based on their

Figure 4: Model development of cooling system in the simulation software exemplified by machine 2

<table>
<thead>
<tr>
<th>No.</th>
<th>Element</th>
<th>Parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Thermal or hydraulic capacity</td>
<td>Geometry of the connections</td>
</tr>
<tr>
<td>2</td>
<td>Thermal resistance</td>
<td>$\delta_{\text{inside}} / \delta_{\text{outside}}, \delta_{\text{loss}}$, fluid properties, pipes properties, ambient properties</td>
</tr>
<tr>
<td>3</td>
<td>Hydraulic resistance</td>
<td>Description form in the simulation software, $\alpha(Re)$, $(Re)$, laminar resistance, $(\Delta p, Q)$ characteristic curve, reference measurement</td>
</tr>
<tr>
<td>4</td>
<td>Heat input</td>
<td>With aid of measurement is calculated by the equation (15)</td>
</tr>
<tr>
<td>5</td>
<td>Heat output</td>
<td>Calculated by forced convection, free convection and heat convection of the hoses, equations (5-14)</td>
</tr>
<tr>
<td>6</td>
<td>Flow valve</td>
<td>Characteristic curve of the valve</td>
</tr>
<tr>
<td>7</td>
<td>Pump</td>
<td>Flow rate and system pressure</td>
</tr>
<tr>
<td>8</td>
<td>Tank</td>
<td>Tank capacity e.g. 15 l</td>
</tr>
<tr>
<td>9</td>
<td>Cooling unit</td>
<td>Data sheets of the cooling unit, e.g. cooling capacity 4.5 kW</td>
</tr>
</tbody>
</table>

Table 1: Model parameter for the simulation model based on the model description in Figure 4

4 Model validation of the current cooling structures

For the experimental investigation of the cooling system of machines 1 and 2, several sensors, as shown in Figure 1 and Figure 2, are used to measure the temperature, the pressure, and the flow rate development. The measured process taken into consideration for the investigation is divided into four sub-processes: warm-up process, idle process (variation of spindle speed, axis position while cooling system is active), setup process (tool/part change
drive B axis and spindle nut Z axis is higher than the outlet temperature. Other components, such as the motor spindle or the secondary part X axis, are cooled during the test process. In contrast to machine 1, the temperature characteristics of the components in machine 2 are not influenced directly by the state of the cooling unit, in spite of the two-point temperature control of the cooling unit. This can be traced back to the three-way-valve (Figure 2, pos.14) that is placed into the return flow side of the cooling system. The valve controls the suction flow to the pump so that the cooling medium always has a constant temperature of 25 °C ±0.1 °C as shown in Figure 6 e.

The simulation models developed for machine s 1 and 2 show a high accuracy of the cooling systems as currently structured do not cool the warmed-up component based on their thermal and hydraulic quantities of the components /13/. The simulation model is validated and thus be used for the improvement of the current structure of the cooling system as well as for the development of process- and demand-oriented control strategies. The considered heat transport in Figure 3 by enforced and free convection as well as by heat conduction of the total hydraulic hoses amount to 45 W as shown in Figure 7. Compared to the performance of cooling units 4.5 kW, the heat transfer through the hydraulic pipes is low, about 1%. It could be neglected in the future. The investigation of two demonstration machines shows that the cooling system of each machine in the idle process requires about 12 % (machine 1) and 26 % (machine 2) of the total energy consumption of the machine tool /4, 12/. The proportion of the calculated hydraulic energy of the pump according to equation (16) is 27.2 % of 12% (machine 1) and 18.5 % of 26% (machine 2), the remaining energy 72.8 % of 12 % and 81.5 % of 26 % is consumed by cooling unit, electrical motor of the hydraulic pump and the pump efficiency. The investigation also depicts that these cooling systems as currently structured do not cool the warmed-up component based on their temperature development (heat input), rather, some components are warmed-up (T_{outlet} < T_{inlet}) and other components are cooled (T_{tank} > T_{inlet}).

\[
E_{phy} = \int_0^t P_{phy} \, dt
\]

(16)

Figure 7: Total heat transport through the hydraulic hoses of the machine 2

5 Potential analysis of different cooling system structures

In relation to the shown deficits of the current cooling system structures of machines 1 and 2, the focus is developing new structures for the cooling system to optimize its thermal behaviour and its effectivity according the goal of a uniform temperature distribution at minimal energy consumption. Figure 8 shows three new structures of a cooling system that can be applied for a demand-oriented supply, in the subchapter the new structures under consideration are described in detail. The effectivity of the new structures will be evaluated firstly in regard to: a constant temperature behaviour at the components, a minimal pressure loss, and a minimal hydraulic power of the pumps. The new structure show also the possibility of the degree of the individualization for new developed cooling system structure.

5.1 Structure 1: Central, variable speed drive unit with proportional valves

The cooling concept considered with a central variable speed drive unit with proportional valves (structure 1) is presented in Figure 9 by three components, electrical cabinet, rotary table, and motor spindle (main drive). The components are cooled individually so that the system control of the cooling system compares the actual and set temperature of the individual components, and, on this basis, adjusts the proportional valves as well as the central variable speed drive unit. The temperature detection is carried out via sensors in the components. For this purpose,
a suitable sensor concept is necessary. If the temperature development of a component does not exceed a predefined threshold, the associated proportional valve will stay closed. With regard to the control strategy, Figure 9 shows that the cooling structure under consideration has three control variables (component’s temperatures) and four control elements (three proportional valves and a variable pump). This makes the system with actual concept over-determined. To solve this problem, three approaches can be taken into account [13]:

- definition of a constraint
- removal of a control element from the active control loop and
- definition of an additional control variable.

For the potential analysis of new cooling structures as well as current cooling structure, three set temperatures are defined for the components: 26 °C for the electrical cabinet, 27 °C for the rotary table and 28 °C for the motor spindle. Based on the measurement, an average equivalent heat flow in the idle process for each component is calculated with equation (15). It is 150 W for the electrical cabinet and rotary table and 1500 W for the motor spindle. Only static operating points of the cooling system are considered in the simulation so that the thermal capacity of the components is not required. Furthermore, the system inlet temperature on the suction side of the pump is considered in the simulation at 25 °C. The cooling unit stays in the two-point temperature control as a bypass cooling and refers to the mixing temperature of the tank.

In Figure 10, the simulation results of the components’ temperature development and the needed cooling medium volume flow of structure 1 in comparison to the current structure of machine 1 are represented. The diagrams (a-c) show the temperature profile of the component and diagram (d) depicts the volume flow development. From the comparison of the simulation results is possible to derive that the thermal behaviour of the component in structure 1 is more stable than in the current structure of machine 1. The components’ temperature in structure 1 keeps constant at the set temperature (26 °C, 27 °C and 28 °C) despite the increase of heat input. In the current structure, the component temperature rises with an increase in the heat input. For this reason, the actual temperature of the components is dependent on the heat input to the component. Looking more closely at the volume flow profiles of the proportional valve of each component, it should be noticed that the cooling volume flow in structure 1 increases based on the temperature development (rising heat input) of the components. In contrast, in the current cooling structure the supplied volume flow to the component is constant. Finally, it is possible to achieve a demand-oriented supply with the considered structure and its controlling. Dependent on the heat input and the temperature development of the component, the proportional valve regulates the required volume flow. It can be ascertained that the volume flow control based on the temperature development is a means for designing the system in a more energy-efficient way based on each individual component’s demand.

The total hydraulic power of the pump in the current structure and structure 1 for different heat input is pointed out in Figure 11. With the variable central displacement pump (structure 1), the total hydraulic power is approximately 160 W at a maximal heat input. Compared to this, the hydraulic power of the fixed displacement pump (current structure) of machine 1 amounts to 370 W (40 l/min at 5.5 bar) and of machine 2 to 340 W (45 l/min at 4.5 bar). A significant energy savings of 56.7 % to 53 % in contrast to the current structures of machines 1 and 2 are possible. The new cooling structure under consideration is significantly more energy-efficient compared to the current structures of machines 1 and 2 with a continuous cooling volume flow.

Figure 10: Simulation results of cooling structure 1 under consideration in comparison to the current cooling structure a) electrical cabinet (EC) b) rotary table (RT) c) motor spindle (MS) d) volume flow profile

Figure 11: Comparison of the pump performance in the current system and in system structure 1

5.2 Structure 2: Decentralized, variable speed drive units without flow control valves

Figure 12 shows the second optimization structure of the cooling system structures under consideration. In structure 2, the components are cooled with individual variable speed pumps that are connected to a common tank and a cooling unit. This kind of cooling structure design does not require flow control valves to distribute the volume flow to the components. Additionally, the system control of the cooling system compares the actual and the set temperature of the individual components and on this basis adjusts the variable speed drive units. So, each pump supplies a different demand-oriented cooling volume flow. If the temperature development of a component
does not exceed predefined threshold, the pump remains inactive. As well as in the structure 1, the cooling unit stays in the two-point temperature control as a bypass cooling and refers to the mixing temperature of the tank. The system boundary condition of structure 1, in regard to the components’ set temperature, heat input, system inlet temperature and static operating points of the system, applies also to structure 2.

Similar to structure 1, Figure 13 shows the simulation results of the components’ temperature development and the required cooling volume flow of structure 2 in comparison to the current cooling system structure of machine 1. Since the components’ temperature remains constant at the set temperature despite an increase in heat input, structure 2 also evidences very good thermal behaviour of the components. A demand-oriented supply of the cooling medium to the components can be realized with this structure. In Figure 14, the total hydraulic power of the variable speed displacement pumps is about 120 W at maximal heat input. An essential energy savings of 67% to 64.7% in contrast to the current structures of machines 1 (370 W) and 2 (340 W) can be shown. So the new cooling structure under consideration is more energy-efficient compared to the current cooling structures.

Figure 14: Comparison of the pump performance in the current system and in system structure 2

6 Summary and outlook

The TCP-displacement of machine tools influences the accuracy in the production process. In order to minimize the deformation of machine tools structure, it is necessary to reduce the thermo-elastic deformation of the machine components with the aid of fluidic systems such as a cooling system. Therefore the requirements on cooling systems in a machine tool with regard to their effectiveness (targeted cooling) and efficiency (lower energy consumption), is too high.

The experimental investigation of a cooling system for two different machine tools has been instrumental in determining the effectiveness as well as the energy consumption of the cooling system /4, 6/. It could be shown that the machine components are not cooled specifically with the current cooling structures, and that the cooling is adjusted insufficiently in reference to the component demand and process requirement. Therefore, the investigation and evaluation of new cooling concepts, both simulatively (network models) and experimentally (test rig), is of great importance.

The examined new cooling structures in this paper, central, variable speed drive unit with proportional valves (structure 1) and decentralized, variable speed drive units without flow control valves (structure 2), show high accuracy with respect to the temperature control of the components compared to the current cooling structure. Apart from that, the hydraulic pump performance of the new structures is about 53% to 67% lower than the hydraulic pump of the current cooling structures.

The focus of further research of the projects will address, firstly, the evaluation of decentralized, variable speed drive units, tanks and cooling units (structure 3), compared to the current structure as well as to structure 1 and 2. Secondly, an energetic analysis is to be made of the overall system for each structure considered, the energy consumption of the electrical motor, frequency converter etc. Lastly, the new structures under consideration shall demonstrate their benefit practice and not only in simulation. To this end, a test rig is being developed, which will allow an experimentally sound statement about the structures regarding their effectiveness and efficiency.

Acknowledgements

The research activities presented are part of the project “Thermo-energetic description of fluid systems” (Ref. No. CRC/TR 96, A04). The authors would like to thank the German Research Foundation (DFG) for financial support.

Funded by

Deutsche Forschungsgemeinschaft

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
</table>
\(A\) Area of the hydraulic connection \([\text{m}]\)

\(c_{\text{fluid}}\) Specific heat capacity at constant pressure \([\text{J/kg} \cdot \text{K}]\)

\(C_{\text{th}}\) Thermal capacity \([\text{J/K}]\)

\(C_{\text{hy}}\) Hydraulic capacity \([\text{m}^3/\text{Pa}]\)

\(D_H\) Hydraulic diameter \([\text{m}]\)

\(D\) Outside diameter of the hydraulic connection \([\text{m}]\)

\(d_{\text{o}}\) Outer diameter of the hydraulic connection \([\text{m}]\)

\(d_{\text{i}}\) Inner diameter of the hydraulic connection \([\text{m}]\)

\(g\) Gravity \([\text{m/s}^2]\)

\(Gr\) Grashof number \([-]\)

\(l\) Length \([\text{m}]\)

\(L\) Characteristic length \([\text{m}]\)

\(N_u\) Nusselt number \([-]\)

\(R_a\) Rayleigh number \([-]\)

\(Re\) Reynolds number \([-]\)

\(P_{\text{outlet}}\) Component outlet pressure \([\text{Pa}]\)

\(P_{\text{inlet}}\) Component inlet pressure \([\text{Pa}]\)

\(\Delta p\) Pressure difference \([\text{Pa}]\)

\(Pr\) Prandtl number \([-]\)

\(\dot{Q}, \dot{Q}_{\text{heat}}\) Heat flow \([\text{W}]\)

\(V\) Volume flow \([\text{m}^3/\text{s}]\)

\(t\) Time \([\text{s}]\)

\(a_{\text{in,dem}}\) convective heat transfer coefficient inside the hydraulic connection \([\text{W/m}^2\cdot\text{K}]\)

\(a_{\text{out,dem}}\) convective heat transfer coefficient outside the hydraulic connection \([\text{W/m}^2\cdot\text{K}]\)

\(a_{\text{con}}\) convective heat transfer coefficient in the hydraulic connection \([\text{W/m}^2\cdot\text{K}]\)

\(\beta\) Coefficient of thermal expansion \([1/\text{K}]\)

\(T_{\text{outlet}}\) Component outlet temperature \([\text{K}]\)

\(T_{\text{inlet}}\) Component inlet temperature \([\text{K}]\)

\(T_w\) Wall temperature of hydraulic pipe \([\text{K}]\)

\(T_a\) Ambient temperature \([\text{K}]\)

\(\Delta T\) Temperature difference \([\text{K}]\)

\(\lambda_{\text{fluid}}\) Thermal conductivity of the cooling medium \([\text{W/m} \cdot \text{K}]\)

\(\lambda_{\text{air}}\) Thermal conductivity of the air \([\text{W/m} \cdot \text{K}]\)

\(\lambda_{\text{con}}\) Thermal conductivity of the hydraulic connection \([\text{W/m} \cdot \text{K}]\)

\(v\) Kinematic viscosity \([\text{m}^2/\text{s}]\)

\(\rho\) Density \([\text{kg/m}^3]\)

References


An energy efficiency evaluation method based on least squares combination weight in refrigeration system

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A new energy efficiency evaluation method, based on least squares combination weight (LSCW), is proposed in this paper. Furthermore, the method is based on the thorough analysis of Fuzzy Analytic Hierarchy Process (FAHP) and Information Entropy (IE). Because of the multi-parameter characteristic of the ammonia refrigeration system, some critical parameters are firstly selected with the help of detailed simulation. Subsequently, a new two-dimension matrix constructed by these parameters is designed. According to the actual working system, compared with the FAHP and IE, results show that the new method has better precision, smaller relative error and greater consistence with actual energy efficiency change.

**Keywords:** Energy efficiency evaluation, Two-dimension matrix, Combination weight, Relative error

**Target audience:** Design Process, Energy Management, Industrial Application

1 Introduction

Ammonia refrigeration system (ARS) is widely used in a large field of application like chemical, food industries and much more. Besides the better refrigeration effect, it is also natural environment friendly, for example, the Ozone Depression Potential (ODP) and Global Warming Potential (GWP) are zero /1/. Now, the mainly problem of ARS is the lower energy efficiency which is caused by the worse management of complex multi-parameter system.

Nowadays, many experts are carrying out the research in two aspects. On one hand, here is the optimization of ARS’s components by pro-simulation /2/, /3/. On the other hand, the developments of fault diagnosis method and energy consumption forecasting algorithm are performed /4/, /5/. However, there is still no effective method which was widely accepted.

In this paper a new two-dimension matrix based on the critical parameters, providing the basic energy efficiency evaluation model, is designed. Through comparing the influence degree to the ARS’s Coefficient of Performance (COP) with the help of AMESim and MATLAB/Simulink, some critical parameters are selected. Combined with the advantages of two common evaluation methods, Fuzzy Analytic Hierarchy Process (FAHP) and Information Entropy (IE), a new evaluation method based on the least squares combination weight (LSCW), is proposed. Finally, the new method is validated and compared with FAHP and IE in actual ARS.

2 Two-dimension Matrix

2.1 Modelling of Ammonia Refrigeration System

Ammonia refrigeration system (ARS) generally consists of compressor, condenser, expansion valve/throttle, evaporator and accumulator. With the help of software AMESim, the basic model is firstly constructed, which is shown in Figure 1.

Refrigerant [ammonia, R717 is refrigerant grade high purity ammonia (NH₃). The product typically is 99.98% pure with minimal levels of moisture (<200 ppm) and other impurities (<5 ppm oil), making it ideal for use in all types of refrigeration systems] flows through the compressor firstly, which raises the pressure of ammonia. Subsequently, ammonia flows through the condenser, where it condenses from vapour form to liquid form, giving off heat in the process. After the condenser, it goes through the expansion valve, where it experiences a pressure drop. Finally, the ammonia goes to the evaporator. It draws heat from the evaporator which causes ammonia to vaporize. The evaporator draws heat from the region that is to be cooled. The vaporized ammonia goes back to the compressor to restart the cycle.

2.2 Selection of Critical Parameters

ARS is a multi-parameter system, including more than 30 parameters. Generally, the system’s energy efficiency will be changed following with the change of each parameter. It is certainly that different parameter have different influence degree. The evaluation matrix will be so big that the calculation process is much slowly, if each parameter is considered /6/. So some critical parameters are usually selected. The select standard is the influence degree to the Coefficient of Performance (COP) of the system.

In refrigeration system, COP is a basic standard of the energy efficiency, as shown in Equation (1).

\[
\text{COP} = \frac{Q_{\text{out}}}{W_{\text{com}} + W_{\text{others}}} = \frac{W_{\text{c}}}{W_{\text{com}} + W_{\text{others}}}
\]

where \(Q_{\text{out}}\) is the exchanged energy in evaporator, \(C_{\text{c}}\) is the Specific Heat Capacity of refrigerant, \(m\) is the mass flow, \(W_{\text{com}}\), \(W_{\text{others}}\) are the energy consumption of compressor and other components.

The influence degree is the ratio of the change of COP to the size of COP’s range. It reflects the influence of the parameter to the system’s energy efficiency. Each parameter has its own working range. It means that the energy efficiency evaluation of the system is based on the healthy working state, without fault. According to the actual statistic and monitor of the testing ammonia refrigeration system, the detailed parameters’ working ranges of the system are shown in Table 1.

![Figure 1: Simulation-model of ammonia refrigeration system](image)
With the help of standardized method, a new matrix $R$ can be obtained.

$$ R = \{r_{ij}\}_{mn} $$

Assumed that the evaluation weight is set to be $w$. $W$ is set as the value of each energy efficiency evaluation.

$$ W = [W_{s1}, W_{s2}, \ldots, W_{sn}] = R \cdot w $$

where $W_{sn}$ is the evaluation of working status $s^n$.

### 3 Energy efficiency evaluation method of Least Squares Combination Weight

#### 3.1 FAHP and IE

Nowadays, there are two methods, Fuzzy analytic hierarchy process (FAHP) and Information Entropy (IE), by which widely used in some state-evaluation systems /7/ /8/ /9/.

Fuzzy Analytic Hierarchy Process (FAHP) is developed from Analytic Hierarchy Process (AHP) which was introduced by Thomas L. Saaty in year 1971 to meet the needs of resource allocation for the military planning /10/. FAHP is used determining the weights of the criteria by decision makers and ranking the methods by AHP subsequently. The main steps of the FAHP are shown in Figure 3. As we know from the working process, the experiences and recommended value of experts are strongly affecting the final results. Therefore, it cannot match the precise evaluation requirement of the ARS’s energy efficiency.

![Figure 3: The working steps of FAHP](image)

Entropy is a concept of thermodynamics, statistical mechanics and information theory. Information Entropy (IE) is occasionally called Shannon’s entropy in honour of Claude E. Shannon. It was introduced by Shannon in 1948 /12/. It tells how much information there is in an event. In general, the more uncertain or random the event is, the more information it will contain. More clearly stated information is a decrease in uncertainty or entropy. The working steps of IE are shown in Figure 4. To some extent IE can reduce the calculation-time of the evaluation; additionally, the critical parameters will be optimized furtherly. But meanwhile, the accuracy is also decreased.

![Figure 4: The working steps of IE](image)
\[ \partial \sum_{j=1}^{m} w_j = 0 \]  

(12)

Then the application matrix is defined as:

\[ \begin{bmatrix} A \ e^T \\ e^T \end{bmatrix} \begin{bmatrix} w \\ \lambda \end{bmatrix} = B \]

(13)

where \( A \) is a diagonal matrix; \( e, w \) and \( B \) are vectors.

\[ A = \begin{bmatrix} \sum r_{11} & \sum r_{12} & \cdots & \sum r_{1m} \\ \sum r_{21} & \sum r_{22} & \cdots & \sum r_{2m} \\ \vdots & \vdots & \ddots & \vdots \\ \sum r_{m1} & \sum r_{m2} & \cdots & \sum r_{mm} \end{bmatrix} \]

(14)

\[ e = [1,1,\ldots,1]^T \]

(15)

\[ B = \begin{bmatrix} \sum (xw_{11} + yw_{12}) r_{11}^2 \\ \sum (xw_{12} + yw_{12}) r_{12}^2 \\ \vdots \\ \sum (xw_{m1} + yw_{m2}) r_{mm}^2 \end{bmatrix} \]

(16)

Solving the equation (13), the combination weight can be obtained.

\[ w = A^{-1} \cdot \left[ B + \frac{1-e^T A^{-1} B}{e^T A^{-1} e} \cdot e \right] \]

(17)

### 4 Comparison of different evaluation methods

The comparison of evaluation effect among different methods is based on an actual ammonia refrigeration system, which belongs to a resin factory in Sichuan Province in China. All of the parameters come from the online monitoring system. According to the selected fifteen parameters, eight online working states are selected.

The values of the parameters are shown in Table 2.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Evaluated working states</th>
<th>No. 1</th>
<th>No. 2</th>
<th>No. 3</th>
<th>No. 4</th>
<th>No. 5</th>
<th>No. 6</th>
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<td>( P_{\text{in}} ) (MPa)</td>
<td></td>
<td>0.24</td>
<td>0.28</td>
<td>0.20</td>
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<td>0.20</td>
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<td>( P_{\text{out}} ) (MPa)</td>
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<td>1.28</td>
<td>1.27</td>
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<td>1.26</td>
<td>1.22</td>
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<td>66.2</td>
<td>85.6</td>
<td>80.0</td>
<td>78.5</td>
<td>82.6</td>
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<td>( K_{\text{in}} ) (%)</td>
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<td>60</td>
<td>65</td>
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<td>( T_{\text{c},\text{in}} ) (°C)</td>
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<td>( T_{\text{c},\text{out}} ) (°C)</td>
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<td>35</td>
<td>36</td>
<td>33</td>
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<td></td>
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<tr>
<td>( T_{\text{e},\text{in}} ) (°C)</td>
<td></td>
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<td>40</td>
<td>38</td>
<td>36</td>
<td>38</td>
<td>38</td>
<td>34</td>
<td>32</td>
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<tr>
<td>( T_{\text{e},\text{out}} ) (°C)</td>
<td></td>
<td>32</td>
<td>32</td>
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<td>33</td>
<td>33</td>
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<tr>
<td>( T_{\text{e},\text{c}} ) (°C)</td>
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<td>4.8</td>
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<tr>
<td>( K_{\text{in}} ) (%)</td>
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<td>85</td>
<td>80</td>
<td>85</td>
<td>75</td>
<td>75</td>
</tr>
</tbody>
</table>

Table 2: Values of the selected states' parameters
4.1 Verification of FAHP and IE

4.1.1 Verification of FAHP

The set of evaluated object is shown in Equation (18).

\[ P = [p_1, p_2, \ldots, p_9]^T = [S_1, S_2, \ldots, S_9] \]  

(18)

According to the working steps of FAHP, the decision matrix of four mainly components (compressor, condenser, evaporator, expansion valve) can be firstly obtained.

\[ Q = \begin{bmatrix} 1 & 9 & 9 & 9 & 1 \\ 9 & 1 & 1 & 1 & 9 \\ 9 & 1 & 1 & 1 & 9 \\ 1 & 9 & 9 & 9 & 1 \\ 9 & 1 & 1 & 1 & 9 \\ 9 & 1 & 1 & 1 & 9 \\ 1 & 9 & 9 & 9 & 1 \\ 9 & 1 & 1 & 1 & 9 \\ 9 & 1 & 1 & 1 & 9 \end{bmatrix} \]  

(19)

Taking advantage of the maximum eigenvalue, the corresponding feature vector can be derived, and then normalized, which is the final weight of four major components, as given by the Equation (20).

\[ w = [w_c, w_e, w_e, w_p] = [0.32, 0.25, 0.25, 0.18] \]  

(20)

Similarly, the inner weight of every parameter can also be calculated.

\[ w_c = [p_{11}, p_{12}, T_{11}, T_{12}, \ldots, p_{91}, p_{92}, T_{91}, T_{92}]^T = [0.257, 0.250, 0.200, 0.186] \]  

(21)

\[ w_e = [T_{e1}, T_{e2}, T_{e3}, T_{e4}, \ldots, T_{e9}]^T = [0.263, 0.263, 0.474] \]  

(22)

\[ w_v = [T_{v1}, T_{v2}, T_{v3}, T_{v4}, \ldots, T_{v9}]^T = [0.230, 0.230, 0.310, 0.230] \]  

(23)

\[ w_p = [p_{p1}, p_{p2}, p_{p3}] = [0.294, 0.294, 0.412] \]  

(24)

The set of evaluated object is shown in Equation (18), the standard matrix can be constructed, as shown in equation (31).

\[ S = [S_1, S_2, \ldots, S_9]^T \]  

(25)

\[ W_{FAHP} = V \cdot S = [2.369, 2.561, 2.364, 2.331, 2.396, 2.345, 2.509, 2.674] \]  

(27)

4.1.2 Verification of IE

Due to the characteristic of big calculation set in IE, in order to reduce the calculation period, combining with the different influence degree, 9 parameters are further selected in the verification of IE, as given by Matrix (28).

\[ \begin{bmatrix} p_{11}, p_{12}, T_{11}, T_{12}, S_1, S_2, \ldots, S_9 \end{bmatrix} \]  

(28)

Construct the evaluation matrix:

\[ U = \begin{bmatrix} u_{11} & u_{12} & \ldots & u_{19} \\ u_{21} & u_{22} & \ldots & u_{29} \\ \vdots & \vdots & \ddots & \vdots \\ u_{91} & u_{92} & \ldots & u_{99} \end{bmatrix} \]  

(29)

As the parameters have different units, the index normalization method is used to map the values to the interval (0, 1). Because the scope of each parameter differs the interval indicators standardization method is selected, as shown in the following equation (30) /15/.

\[ r_i = \begin{cases} 1 - \frac{u_i - u_{\min}}{u_{\max} - u_{\min}} & u_i \in [c_1, c_2] \\ \frac{1}{1 + \frac{u_{\max} - u_i}{u_{\max}} } & u_i \in [c_2, u_{\max}] \end{cases} \]  

(30)

where

- \( r_i \) is the normalized value of \( i \)th parameter;
- \( u_{\min}, u_{\max} \) are the minimum and maximum value of parameter;
- \([c_1, \ c_2]\) is the fixed working range of parameter.

The standard matrix \( R \) can be obtained and then normalized as

\[ R = \begin{bmatrix} 1 & 1 & 1 & 1 & 1 & 1 & 0.83 & 0.62 & 1 \\ 1 & 1 & 1 & 1 & 1 & 1 & 1 & 0.77 & 1 \\ 1 & 0.97 & 0.83 & 0.94 & 1 & 0.08 & 0.71 & 0.39 & 1 \\ 0.98 & 0.87 & 1 & 1 & 0.93 & 0.77 & 0.46 & 1 \\ 1 & 0.99 & 0.92 & 1 & 1 & 0.98 & 0.83 & 0.54 & 1 \\ 1 & 0.99 & 0.83 & 0.97 & 0.09 & 0.76 & 0.39 & 1 \\ 0.99 & 1 & 1 & 1 & 1 & 1 & 0.62 & 1 \\ 1 & 1 & 1 & 1 & 1 & 1 & 0.77 & 1 \\ 1 & 1 & 1 & 1 & 1 & 1 & 0.77 & 1 \end{bmatrix} \]  

(31)

\[ E = \begin{bmatrix} E_1, E_2, \ldots, E_n \end{bmatrix} \]  

(32)

\[ E_i = \frac{1}{\ln n} \sum_{j=1}^{n} E_{ij} \ln E_{ij} \quad j = 1, 2, \ldots, n \]  

(33)

Following the equation (35), the weight of output attribute can be got, as shown in (36).

\[ w_j = \begin{cases} 1 & j = 1, 2, \ldots, n \\ \frac{1 - E_j}{\max_{j=1}^{n} (1 - E_j)} & \text{other cases} \end{cases} \]  

(35)

\[ w_{IE} = [0.042, 0.125, 0.042, 0, 0.167, 0.625, 0] \]  

(36)

The comprehensive evaluation value of every state can be obtained,

\[ W_{IE} = R \cdot w = [0.735, 0.864, 0.532, 0.608, 0.675, 0.557, 0.763, 0.864] \]  

(37)

4.2 LSCW

According to the description of LSCW’s working principle in above 3.2, it is also that nine parameters are selected as the selection in IE. The standard matrix can be constructed, as shown in equation (31).

The attribute weight from IE is the same as the equation (36).

\[ w_{IE} = [0.042, 0.125, 0.042, 0, 0.167, 0.625, 0] \]  

In accordance with the principle of FAHP, the attribute weight from FAHP can be obtained,

\[ w_{FAHP} = [0.153, 0.153, 0.119, 0.119, 0.051, 0.119, 0.119, 0.003, 0.003] \]  

(38)

In accordance with Equations (13), (14), (15) and (17), the attribute weight \( w_{LSCW} \) are calculated with different reliabilities which are set in proportion. Subsequently, the corresponding comprehensive evaluation values \( W_{LSCW} \) are obtained by calculation in Equation (5), as shown in Table 3 /16/.
Eight selected states’ value of COP are shown in the following, $S_{\text{COP}} = [3.6, 4.0, 3.2, 3.3, 3.4, 3.3, 3.8, 4.1]$.

Normalize equation (27), (37), (41) and (42), the results are shown in Table 4. The detailed comparison between three evaluation methods and real COP can be shown in Figure 6.

<table>
<thead>
<tr>
<th>Methods</th>
<th>Total points</th>
<th>Deviation points</th>
<th>Relative error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>FAHP</td>
<td>1440</td>
<td>217</td>
<td>15.07</td>
</tr>
<tr>
<td>IE</td>
<td>1493</td>
<td>193</td>
<td>13.40</td>
</tr>
<tr>
<td>LSCW</td>
<td>1493</td>
<td>49</td>
<td>3.34</td>
</tr>
</tbody>
</table>

Table 5: Comparison in deviation points and relative error
5 Summary and conclusion

The paper deals with the research and the development of a new energy efficiency evaluation method in refrigeration system. Considering the characteristic of complex multi-parameter, the major evaluation parameters (15 parameters) are firstly selected by simulation. Then a two-dimension evaluation matrix is designed. Based on the analysis of two general evaluation methods (FAHP and IE), a new energy efficiency evaluation method, Combination Weight based on Least Squares (LSCW) is proposed. Combined with an actual project, the evaluation effect of different methods are tested and compared. The results show that, the proposed method has better precision and greater consistence with the actual energy efficiency change. Furthermore, it has less derivation points with relative error at 3.34%, much better than the given two general evaluation methods. It can be developed to be a module in energy management system, by which the energy efficiency of refrigeration system can be real managed and optimized.

6 Acknowledgements

The work described in this article is based on the real energy-saving project of Beijing ECOSO Co., Ltd. The authors would like to thank ECOSO for its support.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{suc}$</td>
<td>Suction Pressure of Compressor</td>
<td>[MPa]</td>
</tr>
<tr>
<td>$P_{dis}$</td>
<td>Discharge Pressure of Compressor</td>
<td>[MPa]</td>
</tr>
<tr>
<td>$T_{suc}$</td>
<td>Suction temperature of Compressor</td>
<td>[°C]</td>
</tr>
<tr>
<td>$T_{dis}$</td>
<td>Discharge temperature of Compressor</td>
<td>[°C]</td>
</tr>
<tr>
<td>$K_c$</td>
<td>Opening of Guiding Valve</td>
<td>[%]</td>
</tr>
<tr>
<td>$T_{oil}$</td>
<td>Temperature of oil</td>
<td>[°C]</td>
</tr>
<tr>
<td>$p_{oil}$</td>
<td>Pressure of Compressor oil</td>
<td>[MPa]</td>
</tr>
<tr>
<td>$T_{en,in}$</td>
<td>Inlet Temperature of Ammonia in Condenser</td>
<td>[°C]</td>
</tr>
<tr>
<td>$T_{en,out}$</td>
<td>Outlet Temperature of Ammonia in Condenser</td>
<td>[°C]</td>
</tr>
<tr>
<td>$T_{atm}$</td>
<td>Temperature of Atmosphere</td>
<td>[°C]</td>
</tr>
<tr>
<td>$T_{cw,in}$</td>
<td>Inlet Temperature of Water in Condenser</td>
<td>[°C]</td>
</tr>
<tr>
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<td>Outlet Temperature of Water in Condenser</td>
<td>[°C]</td>
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<td>$T_{cm}$</td>
<td>Condensing Temperature</td>
<td>[°C]</td>
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<td>$T_{ew,in}$</td>
<td>Inlet Temperature of Water in Evaporator</td>
<td>[°C]</td>
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<tr>
<td>$T_{ew,out}$</td>
<td>Outlet Temperature of Water in Evaporator</td>
<td>[°C]</td>
</tr>
<tr>
<td>$T_{tn,in}$</td>
<td>Inlet Temperature of Ammonia in Throttle</td>
<td>[°C]</td>
</tr>
<tr>
<td>$T_{tn,out}$</td>
<td>Outlet Temperature of Ammonia in Throttle</td>
<td>[°C]</td>
</tr>
<tr>
<td>$\theta_{sh}$</td>
<td>Degree of Superheat</td>
<td>[°C]</td>
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<tr>
<td>$T_{en,in}$</td>
<td>Inlet Temperature of Ammonia in Evaporator</td>
<td>[°C]</td>
</tr>
<tr>
<td>$T_{en,out}$</td>
<td>Outlet Temperature of Ammonia in Evaporator</td>
<td>[°C]</td>
</tr>
<tr>
<td>$p_{1,in}$</td>
<td>Inlet Pressure of Throttle/Expansion Valve</td>
<td>[MPa]</td>
</tr>
<tr>
<td>$p_{1,out}$</td>
<td>Outlet Pressure of Throttle/Expansion Valve</td>
<td>[MPa]</td>
</tr>
<tr>
<td>$T_{eva}$</td>
<td>Evaporating Temperature</td>
<td>[°C]</td>
</tr>
<tr>
<td>$K_l$</td>
<td>Opening of Throttle</td>
<td>[%]</td>
</tr>
<tr>
<td>$m_w$</td>
<td>Mass Flow of Water</td>
<td>[m³/min]</td>
</tr>
<tr>
<td>$m_a$</td>
<td>Mass Flow of Ammonia</td>
<td>[m³/min]</td>
</tr>
</tbody>
</table>

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Adaptive Control for Direct-Driven Hydraulic Drive

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Energy efficient and environment conscious solutions are currently in high demand. This paper illustrates the potential of pump-controlled actuators such as directly driven hydraulic drives (DDH) for various zonal hydraulics applications. A novel pump-controlled actuator, powered directly by a servo motor is considered for industrial and mobile applications replacing conventional valve-controlled hydraulics. This solution is targeting improvements in energy efficiency, especially for continuous operation systems, however, due to the nature of the solution, system response has high dependency on electric motor dynamics. Therefore, adaptive controller is designed to realise benefits of the DDH. Study presents results of performance by simulation of this new concept.

Keywords: Fluid power networks, electro-hydrostatic drive, adaptive controller, zonal hydraulics (provide 4 to 5 keywords)

Target audience: Mobile Hydraulics, Control

1 Introduction

The rapid rise of oil prices and government-enforced CO2 regulations /1/ leads to increased demand for energy efficient construction /2/, stationary /3/, and mobile machinery /4/. Electric and hybrid drives are potential solutions to meet these requirements. In particular, non-road mobile machinery (NRMM) is a challenging field of electrification and hybridization applications due to typical duty cycles, which may include high, brief power peaks and extreme working conditions. In various publications, zonal or decentralized solutions are proposed for the hybridization of NRMM. In /5/, an overview of various levels of individualization and integration were investigated. Proposed technological solutions offered a great potential for further development and optimization of hydraulic drive technology. Examples of pump controlled solutions are displacement controls /6-8/ and decentralized hydraulics /9-12/. Figure 1 illustrates examples of pump-controlled actuators.

![Figure 1: Examples of the pump-controlled actuators a) /13/; b) /14/; c) /15/; d) /16/](image)

According to Figure 1 and /17/, a pump-controlled system can be utilized as a combination of a variable displacement pump with a fixed-speed motor, or fixed displacement pump with variable speed motor or a combination of these two.

The authors in /18/ provide an overview of pump controlled differential cylinder systems. In general, pump-controlled actuators provide the most benefits for applications where infrequent use is required.

At the same time, valve-less approaches demand highly dynamic performance of these types of systems. Recent research in pump-controlled systems has focused on symmetrical cylinders. The thermal behavior of pump-controlled actuators is a significant concern and it has been investigated /19, 20/. On the other hand, topics related to performance and energy consumption are also of great interest but not investigated widely. Authors in /18/ claim that 80% of all cylinder applications utilize the differential cylinder due to its lower space requirement.

In previous research /3/, application of Electro-hydrostatic actuator (EHA) as pump-controlled actuator for industry application showed high requirements for response and dependency of electric motor dynamics.

The current study will continue with modeling work and experimental testing to validate proposed technical solution. The direct-driven hydraulic drive (DDH) was utilized as an example of a pump-controlled actuator. In DDH, a hydraulic cylinder is directly actuated using two pump/motors, a Synchronous Torque Motor drive and pump/motor units which are coupled in one axle. Therefore, in the DDH setup, the velocity and position control of a cylinder are implemented in an open-loop control system without conventional valves and defined with the speed of a motor drive and displacement of the pump/motor units. Setup benefits of the compact high power of the hydraulic system and flexible control of the electric motor. However, the dynamic processes in hydraulics are naturally nonlinear, and the system parameters vary widely. Therefore, a conventional PID controller cannot provide equally results for various operations of this pump-controlled system. Considering the control options for the speed and position control of this DDH system, various alternative methods are available. Therefore, goal of this paper is investigating control challenges and design an adaptive controller.

First in this paper, a DDH setup as a pump-controlled system is introduced. This paper introduces in short the Matlab/Simulink model of the direct-driven hydraulic setup, highlighting effects of the dynamics of the electric motor. Secondly, to achieve excellent control capabilities, an adaptive controller is designed to control the cylinder position. The system is investigated by means of simulations and compared with classic PID controller. As a result, conclusion are drawn considering designed controller and its benefits for the hydraulic system.

2 Case study

DDH topology was selected as a case study. The working principle of the proposed structures can be described as follows: A speed-controlled electric servo motor drive rotates two hydraulic pumps. This way input and output flow delivered to the double-acting hydraulic cylinder is controlled directly by motor speed. This study will investigate performance of the selected topology for lifting-lowering cycle.

Figure 2 shows a schematic of the DDH setup. A double-acting asymmetrical cylinder was applied. Hydraulic motors with displacements of 22.8·10^{-4} m^3/rev and 14·10^{-4} m^3/rev, respectively were utilized.

In Figure 2, an additional hydraulic accumulator B (Figure 2, g) was utilized to balance flow from cylinder chambers, and hydraulic accumulator A (Figure 2, l) was applied as a conventional tank replacement. The following section introduces equations for modelling the proposed DDH setup and controller design.
3 Model and Controller

In order to acquire the dynamic response of the DDH unit, this study constructed a model in Matlab/Simulink, which integrated with hydraulics, electric drive and control systems of the DDH. The DDH system consists of an electric drive, cylinder, two fixed displacement pumps/motors, and hydraulic accumulators. Section 3.1 introduces modelling of electric drive and hydraulic system of DDH. Section 3.2 presents developed adaptive controller.

3.1 Model

Electric drive modelled with following Equations (1), (2) and (3) based on /11/ in Matlab/Simulink:

\[ T_{em} = \frac{\pi}{4} P_d q_d i_q. \]  

(1)

where \( T_{em} \) is electromagnetic torque, \( p_d \) is pole pairs, \( q_d \) is the flux linkage of the stator d winding that produced by the rotor magnets, \( i_q \) is stator current in q axis.

\[ v_d = R_i i_d + \omega_m l_m i_d, \]  

(2)

\[ v_q = R_i i_q + \omega_m l_m i_q + \omega_m l_d. \]  

(3)

where \( v_i \) is stator voltages in dq transformation, \( L_i \) is stator inductance, \( R_i \) is stator resistance, \( i_d \) and \( i_q \) are stator current in d axis and \( \omega_m \) is speed. The parameter of the motor is shown in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated Speed</td>
<td>6000</td>
<td>(rpm)</td>
</tr>
<tr>
<td>Rated torque</td>
<td>8.1</td>
<td>[N·m]</td>
</tr>
<tr>
<td>Stall current</td>
<td>3.9</td>
<td>[A]</td>
</tr>
<tr>
<td>Rated power</td>
<td>2.54</td>
<td>[kW]</td>
</tr>
<tr>
<td>( R_\ell ) - Resistance(phase)</td>
<td>0.416</td>
<td>[Ω]</td>
</tr>
<tr>
<td>( L_\ell ) - Inductance(phase)</td>
<td>1.36×10⁻⁴</td>
<td>[H]</td>
</tr>
<tr>
<td>inertia</td>
<td>3.4×10⁻⁴</td>
<td>[kg·m²]</td>
</tr>
</tbody>
</table>

Table 1: Utilised motor parameters.

The mathematical models of the double-acting asymmetrical cylinder, pump, and pipes of DDH are built also in Matlab/Simulink. Vivoti gear motors with displacements of 22.8×10⁻⁴ and 14.4×10⁻⁴ m³/rev are utilized with leakage submodels obtained by measurements in /21/. For detailed explanations regarding hydraulics, refer to work in /21/.

The pressure in a double-acting asymmetrical cylinder is described in Equations (4) and (5):

\[ \begin{align*}
\dot{p}_A &= \frac{a}{V_0} \left[ \frac{\dot{i}_A}{K} - K_i (p_A - p_B) - A_\lambda \dot{x} \right], \\
\dot{p}_B &= \frac{a}{V_0} \left[ \frac{\dot{i}_B}{K} + K_i (p_A - p_B) + A_\lambda \dot{x} \right].
\end{align*} \]  

(4)

(5)

where \( B(p) \) is the bulk modulus for oil varying with pressure, \( V_0 \) is the dead volume of the cylinder’s piston side and piston rod side chambers, \( K_i \) is the leakage constant between the cylinder chambers, \( q_A \) and \( q_B \) are the flows into the cylinder’s side chambers, respectively. Cylinders’ force \( F \) was calculated as:

\[ F = (p_A A_k - p_B A_k) - F_{end}. \]  

(6)

where \( A_k \) and \( A_{end} \) are cylinder’s piston side and piston rod side areas, \( p_A \) and \( p_B \) are corresponding pressures in cylinder chambers, \( F_{end} \) is the friction force of the cylinder calculated by utilizing the LuGre model /22/. \( F_{end} \) is the end force.

3.2 Adaptive controller

Aim of this study is to develop a controller that will reduce negative effect mass on system dynamics and oscillating processes in cylinder position. Therefore, the basic structure of the developed control system is illustrated in Figure 3, where model consists of electric motor, pumps, cylinder, mass and reference model and its adaptive algorithm. Utilised principle of adaptive controller is based on adding a corrective control signal, which forces the difference of these signals is fed to the adaptive algorithm, which corrects input to the plant.

![Figure 3: Control system diagram](image)

For electric motor control, inner current loop is realized with \( i_d = 0 \) control and with PI control. Speed control loop and position control loop are realized as in a classic PID control system. Cylinder position is compared to the reference model. The difference of these signals is fed to the adaptive algorithm, which corrects input to the plant. Reference dynamics are realized with help of a transfer function of first order as shown in Equation (7) and another system of second order shown in Figure 4. System of second order realised with limitation of output velocity.

\[ W(s) = \frac{1}{0.0035s + 1} \]  

(7)

Full reference model demonstrated in Figure 4.
Comparison of the Figures 6 and 7 demonstrates that adaptive algorithm reduces delay between reference model and acting position.

Additional adaptive control signal was formed as a weighted sum of differences between output of the plant and reference model and its derivative.

In Figure 5, \( x \) is the actual cylinder position, \( \dot{x} \) is the first derivative of position i.e. velocity. Also, \( x_m \) is reference models cylinder position and \( \dot{x_m} \) is the reference model velocity. Error between velocities is multiplied by gain \( K_1 \). And result is a sum of error between model and actual cylinder position and multiplied with gain \( K_2 \). Output is limited and fed to the output of motor current PI controller in q axes.

Due to system limitation, the selected coefficients are limited by amplitude of corrective control signal.

Due to the dynamics of adaptive controller supposed to be fast (speed of adaptation process need to be faster than response of real system), therefore, gain \( K_2 \) (Figure 5) should be huge. Thus, limitation of system was applied. Magnitude of adaptive signal is not more than maximum input control signal of the motor controller.

Gains for adaptive algorithm (\( K_1=0.0048 \) and \( K_2=6858 \)) were selected by using a genetic algorithm. The utilised fitness function based on mean square deviation between output of reference model and output real system. The genetic algorithm finalized its work by reaching best result on the last 10 generations.

4 Results.

This section contains simulation results with the proposed controller. Performance of classic system with cascade PID regulator was compared to the proposed adaptive algorithm. Figures 6 and 8 illustrate the cylinder position response with 60 kg payload and 120 kg for cascade PID regulator, respectively. In Figures 6 and 8 dotted line correspond to reference signal, grey line is response of PID controller, and black line is reference model.

Figures 7 and 9 illustrate the cylinder position response with 60 kg and 120kg payload with proposed adaptive algorithm, respectively. In Figure 7 dotted line is reference, grey line is response of adaptive controller, and black line is response of reference model.
The aim of this paper was to create a universal controller which will reduce the negative effect of oscillations during lowering in the steady state region.

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The research was enabled by the financial support of Academy of Finland (project IZIF), internal funding at the Department of Automatic Control Systems, Faculty of Electrical Engineering and Automatics, St. Petersburg Electrotechnical University “LETI”, Department of Mechanical Engineering at Aalto University, Finland and Department of Automatic Control Systems, Faculty of Electrical Engineering and Automatics, St. Petersburg Electrotechnical University “LETI”, Department of Mechanical Engineering at Aalto University, Finland.

6 Acknowledgements
The research was enabled by the financial support of Academy of Finland (project IZIF), internal funding at the Department of Automatic Control Systems, Faculty of Electrical Engineering and Automatics, St. Petersburg Electrotechnical University “LETI”, Russia.

Comparison of Figures 8 and 9 demonstrates faster response of the system with adaptive controller and reduced oscillations during lowering steady state region.

The aim of this paper was to create a universal controller which will reduce the negative effect of mass on system dynamics and oscillating processes in cylinder position. This target became challenging. Analysis of simulation results demonstrated that better results for DDH could be achieved by splitting and developing independent controller for lifting and lowering part of cycle. This work utilised a genetic algorithm which helped to find gains suitable for both parts of the cycle. However, the utilised adaptive controller did not demonstrate significant improvement compared to cascaded PID controller. In addition, simulation study showed high sensitivity of system response to the tuning of electric motors’ inner PID current loop.

For future investigation, it is suggested to switch from simplified (proposed) adaptive algorithm to combination with fuzzy logic and separate lifting and lowering controllers or increase complexity of adaptive algorithm.

5 Summary and Conclusion
This research analysed the modern electric drive in controlling the position of a cylinder rod in a direct driven electro-hydraulic system. The developed adaptive controller allows on-line tuning depending on the direction of motion. Simulation results demonstrated improvement in system response with proposed simplified adaptive algorithm. This makes the controller more adaptive and effective in the proposed application compared to cascade PID regulator. However, utilised adaptive controller did not demonstrate significant improvement compared to cascade PID controller. In addition, simulation study showed high sensitivity of system response to the tuning of electric motors’ inner PID current loop. Improvements for further study were suggested.

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Nomenclature

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<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$A_s$</td>
<td>Cylinder’s Piston Side Area</td>
<td>[m$^2$]</td>
</tr>
<tr>
<td>$A_n$</td>
<td>Cylinder’s Piston Side Area</td>
<td>[m$^2$]</td>
</tr>
<tr>
<td>$B$</td>
<td>Bulk Modulus for Oil Varying with Pressure</td>
<td>[-]</td>
</tr>
</tbody>
</table>

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Identification and synthesis of linear-quadratic regulator for digital control of electrohydraulic steering system

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The paper presents an optimal reference tracking algorithm for electrohydraulic steering systems which is based on multivariable system identification, linear quadratic control and Kalman filtering for state estimation. A laboratory test-bench composed of electrohydraulic-steering unit (EHSU), steering cylinder, 32-bit microcontroller, steering wheel and joystick supports experimental work. Traditional approach for reference tracking in steering usually is based on classical control algorithms such digital PI regulator or non-digital hydromechanical feedback. In contrast the control theory suggests advanced control techniques, which can take into account multivariable nature of the process. In this way a higher closed-loop performance can be achieved.

Keywords: Multivariable identification, linear-quadratic regulator, Kalman filter, electrohydraulic steering system

Target audience: Systems, Mobile Hydraulics

1 Introduction

In modern mobile machines, the proportional electrical control of the steering system is used due to the need for remote control via GPS. In addition, the mechanical steering with variable steering ratio from the steering wheel to the machine's steer axle is often a sought after function to improve driver productivity and comfort [1,2]. This leads to the need for an effective integrated control system which should ensure the quality behavior of the entire electrohydraulic system [3].

In the case of remote control via controller, the behavior of the machine depends heavily on embedded software regulator. Quality of control for this case can be improved if more accurate plant model is used. Model-based approach is widely applied in modern control system design. This approach is also useful in software engineering for design of real-time signal processing systems. In mathematical modeling always there is a trade-off between complexity and accuracy of the model [4]. Usually complex models are appropriate for plant behavior analysis but they are inconvenient to design control algorithm.

Steering control systems are necessary because loading torques acting upon steering axle may disturb steering performance. Also the mathematical model cannot take into account all physical details because it would become impractically complex [5]. In industry there are two common control strategies - feedback interconnection and hierarchical (or cascade) subordination of feedback loops. A wide known fact is that more than 90 % of all the industrial feedback control loops are derivatives of PID algorithm [6,7]. The reason is not only the intuitive idea behind PID but also because it has proved robust to small model uncertainties. However PID controller tuning becomes a difficult tasks in case of many inputs many outputs (MIMO) plant. For MIMO case the control theory suggests many advanced control techniques, which can take into account multivariable nature of the process. Such practical approved control technique is linear quadratic Gaussian regulator (LQG) that involves linear quadratic regulator (LQR) and uses state estimates obtained by Kalman filter [8,4]. The LQG algorithm takes into account not only multivariable process nature but also the influence of noises to the plant dynamics.

The main objective of this work is to present the designed system for control of electrohydraulic steering system that is implemented in low speed mobile machines. The goal of control algorithm is to achieve fast transient response without overshooting and static error in whole working range. To achieve this aim first a multivariable dynamical plant model is estimated by identification procedure. The model obtained is validated by various statistical tests. The multivariable LQR regulator with integral action and Kalman filter are designed. Appropriate software which is implemented in 32-bit microcontroller is developed. Experimental results are presented which confirm that the control system achieves the prescribed performance.

The paper is organized as follows: section 2 presents designed experimental setup, in section 3 the results from is multivariable system identification are presented, section 4 shows design of linear-quadratic controller with Kalman filter and in section 5 some experimental results are given.

2 Experimental system layout

Authors have developed a laboratory hydraulic test equipment for EHSU type OSPEC200 LSRM, taking into account current technical specifications from the manufacturer [3]. Figure 1 shows hydraulic schematics of the test bench system, described in detail in [9].

![Figure 1: Hydraulic schematic of EHSU test-bench with pressure loading subsystem.](image-url)
A digital control system has been developed in which an electronic joystick is configured for an input device. It consists of an electronic joystick, a digital LQG regulator supplying control voltage to the PVE module, which in turn control the PVE built-in electrohydrostatic proportional valve. The electrohydraulic proportional valve (6) determines the direction of movement of the executive hydraulic cylinder (12), by feeding a working fluid to one of both chambers. The plant output is a cylinder piston position that is measured through a sensor. The measured sensor signal is used in feedback and is compared with the reference signal.

Digital control systems are based on a controller type MC012-022 and electronic joystick type JS6000 (Danfoss platform for mobile applications), by which operated module with two-way, two-position valves connected in parallel - PVE.

In the programming environment (PLUS + 1 Guide) [10] controller MC012-022 is created and a program that calculates a parameters of control signal. The experimental system (Fig.1) consists of EHSU type OSPE 200 and symmetric servo-cylinder connected by pipelines through L and R.

The developed test bench system is in accordance with the current requirements for the testing of electro-hydraulic steering devices with different pressure loads. Pressure loading system is composed of a hydraulic block with over-center valves (pos. 11, Fig. 1), which are connected to the both chambers of the servo-cylinder.

3 Multivariable system identification

To determine the mathematical model of electrohydraulic steering system one may apply physical modeling or identification [5,11]. Physical modeling requires profound of knowledge about physics of the plant and a lot of a priori information such as various characteristics, values of specific constants and hydraulic resistances. Due to the lack of a priori information, in this study a numerical model obtained by identification procedure is used. Another reason to use this approach is that in addition to the description of plant dynamics the noise model is obtained. This model can be used to design an appropriate optimal filter such as Kalman filter. Thus the goal of identification is to obtain a linear black box model which sufficiently well describes electrohydraulic steering system dynamics and noises in wide working range. To obtain such model first the open loop identification experiments, according to scheme shown in Figure 2, is designed. The sample time of $T_p = 0.05s$ is chosen, that is sufficiently small. To provide persistent excitation to the plant input a random binary signal (RBS) is applied. It is obtained from filtered through relay white Gaussian noise. The amplitude of RBS is chosen to be ±2500. In this manner the whole working range of input signal is used. The plant dynamics can be represented by single input two outputs model in which the input is control signal, the first output is the difference between measured pressures in right and left chambers and the second output is the measured position of cylinder.

![Figure 2: Scheme of open loop identification experiment](image)

Before use the input-output data for identification the constant values should be removed. From the centered measured data two data sets are formed. First of them is used for model estimation and second - for model validation. They are depicted in Figures 3-4. The excitation level of identification input signal is 500. This means that up to 500 parameters can be estimated from estimation data set. The identification procedure starts with estimation of state space model with free parameterization

$$x(k+1) = Ax(k) + Bu(k) + K_vv(k)$$
$$y(k) = Cx(k) + Du(k) + v(k)$$

(1)

where $x(k) = [x_1, x_2, ..., x_n]^T$ is a state vector, $u(k)$ is the input signal, $y(k) = [y_{pres}, y_{pos}]^T$ is the output vector, $v(k)$ is a model disturbance (residual) and $A, B, C, D, K_v$ are the matrices with appropriate dimensions.

$A = \begin{bmatrix} 0.8769 & -0.3987 & 0.3986 & 0.0043 & 0.1112 & 0.0621 \\ 0 & 0 & 1 & 0.0043 & 0.0011 & 0.0221 \\ -0.1666 & -0.5099 & 1.509 & 0.1112 & -0.62614 \end{bmatrix}$

(2)

$B = \begin{bmatrix} 0.5099 \\ 0.5099 \\ 1.509 \end{bmatrix}$

$C = \begin{bmatrix} 1 \\ 0 \\ 0 \end{bmatrix}$

$D = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}$

$K_v = \begin{bmatrix} 0.1112 & -0.62614 \\ -0.09525 & 1.55 \\ -0.2003 & 1.897 \end{bmatrix}$

The comparison between measured pressure and cylinder position (according validation data set) and model outputs is presented in Figure 5. The value of $F$ of FIT between measured pressure and model pressure is 56.12% and the one between the measured position and model position is 76.5%. These results mean that estimated model captures sufficiently well plant dynamics. The results from whitening and independence tests of residuals are shown in Figure 6. The frequency response of estimated high order finite impulse response (FIR) model between control signal and residuals along with 99% confidence region is depicted in Figure 7. As can be seen from Figure 6 the noise model is adequate and there is no significant correlation between input and residuals. This result is confirmed again from the test presented in Figure 7 (it shows that there is not significant dynamics between input signal and residuals in the whole interested frequency range).

![Figure 3: Estimation data set](image)

![Figure 4: Validation data set](image)

![Figure 5: Model outputs and measured outputs](image)

![Figure 6: Residual test of estimated model](image)
The optimal control law is obtained in the form
\[ u(k) = -\mathbf{K}_i x(k), \quad \mathbf{K}_i = \left[ K_x - K_c \right] \]
(5)
where \( K_x \) is the proportional term matrix gain and \( K_c \) is the integral term gain. The controller matrix \( \mathbf{K} \) is obtained from minimization of quadratic performance index
\[ J(u) = \sum_{k=0}^{\infty} T \mathbf{x}^T(k) \mathbf{Q} \mathbf{x}(k) + u^T(k) \mathbf{R} u(k), \]
(6)
where \( \mathbf{Q} \) and \( \mathbf{R} \) are positive definite matrices chosen to ensure acceptable transient response of the closed-loop system. The optimal feedback matrix \( \mathbf{K} \) is determined by
\[ \mathbf{K} = \left( \mathbf{R} + \mathbf{B}^T \mathbf{P} \mathbf{B} \right)^{-1} \mathbf{B}^T \mathbf{P}, \]
(7)
Where \( \mathbf{P} \) is the positive definite solution of the discrete-time matrix algebraic Riccati equation
\[ \mathbf{A}^T \mathbf{P} \mathbf{A} - \mathbf{A}^T \mathbf{P} \mathbf{B} \left( \mathbf{R} + \mathbf{B}^T \mathbf{P} \mathbf{B} \right)^{-1} \mathbf{B}^T \mathbf{P} \mathbf{A} + \mathbf{Q} = \mathbf{0}. \]
(8)

The matrix \( \mathbf{Q} = \begin{bmatrix} 10^4 & 0 & 0 & 0 \\ 0 & 10^4 & 0 & 0 \\ 0 & 0 & 10^4 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \) and \( \mathbf{R} = 5000 \).

Since the state \( x(k) \) of system (2) is not accessible, the optimal control law (6) is implemented as
\[ u(k) = -K_x \hat{x}(k) + K_c x(k), \]
(9)
where \( \hat{x}(k) \) is estimate of \( x(k) \). It is obtained by discrete time Kalman filter
\[ \hat{x}(k+1) = A \hat{x}(k) + B u(k) + K_f (y(k+1) - C \hat{x}(k) - D v(k)), \]
(10)
The filter matrix \( \mathbf{K}_f \) is determined as
\[ \mathbf{K}_f = D_f \mathbf{C}^T \left( \mathbf{C} \mathbf{C}^T + 10^{-4} \mathbf{I}_2 \right)^{-1}, \]
(11)
where \( \mathbf{I}_2 \) is second order unit matrix and matrix \( D_f \) is the positive semi-definite solution of Riccati equation
\[ A \mathbf{C} - A \mathbf{C} \mathbf{C}^T \left( \mathbf{C} \mathbf{C}^T + 10^{-4} \mathbf{I}_2 \right)^{-1} \mathbf{C} A + \mathbf{K}_f \mathbf{K}_f^T = 0. \]
(12)
The matrix \( D_f = \begin{bmatrix} 108.97 & 0 \\ 0 & 27.44 \end{bmatrix} \) is the variance of noise \( v(k) \).

### 4 Design of linear-quadratic regulator

The structure scheme of control system with LQR controller and Kalman filter is shown in Figure 8. To ensure sufficiently well reference tracking an LQR controller with integral action is designed. The design is done on the basis of deterministic part of estimated model (2) which is extended with an extra state \( x_i \). This extra state is discrete time integral of position error

\[ x_i(k+1) = x_i(k) + T_f \left( y_{\text{ref}}(k) - y_{\text{pred}}(k) \right), \]
(3)
where \( y_{\text{ref}}(k) \) is the reference. Thus, combining the deterministic part of equation (2) and equation (3) one obtains the augmented system
\[ \mathbf{x}(k+1) = \mathbf{A} \mathbf{x}(k) + \mathbf{B} u(k) + \mathbf{C} y_{\text{ref}}(k), \]
\[ y(k) = \mathbf{C} \mathbf{x}(k), \]
(4)

The optimal controller matrix is obtained for
\[ \mathbf{Q} = \begin{bmatrix} 10^4 & 0 & 0 & 0 \\ 0 & 10^4 & 0 & 0 \\ 0 & 0 & 10^4 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \]
and \( \mathbf{R} = 5000 \).

Since the state \( x(k) \) of system (2) is not accessible, the optimal control law (6) is implemented as
\[ u(k) = -K_x \hat{x}(k) + K_c x(k), \]
(9)
where \( \hat{x}(k) \) is estimate of \( x(k) \). It is obtained by discrete time Kalman filter
\[ \hat{x}(k+1) = A \hat{x}(k) + B u(k) + K_f (y(k+1) - C \hat{x}(k) - D v(k)), \]
(10)
The filter matrix \( \mathbf{K}_f \) is determined as
\[ \mathbf{K}_f = D_f \mathbf{C}^T \left( \mathbf{C} \mathbf{C}^T + 10^{-4} \mathbf{I}_2 \right)^{-1}, \]
(11)
where \( \mathbf{I}_2 \) is second order unit matrix and matrix \( D_f \) is the positive semi-definite solution of Riccati equation
\[ A \mathbf{C} - A \mathbf{C} \mathbf{C}^T \left( \mathbf{C} \mathbf{C}^T + 10^{-4} \mathbf{I}_2 \right)^{-1} \mathbf{C} A + \mathbf{K}_f \mathbf{K}_f^T = 0. \]
(12)
The matrix \( D_f = \begin{bmatrix} 108.97 & 0 \\ 0 & 27.44 \end{bmatrix} \) is the variance of noise \( v(k) \).

### 5 Experimental results

Experiments with the designed LQR regulator require specific implementation technique in target microcontroller MC012-022. The controller interconnection can be represented in equivalent single matrix vector formulas.
There is zero error in steady state. Transitional processes are of an aperiodic nature, without overshoot. The quality of the transition processes is maintained when cylinder piston moving in both directions.

\[
\begin{align*}
\begin{bmatrix}
\dot{x}(k+1) \\
x(k+1) \\
u(k+1)
\end{bmatrix} &=
\begin{bmatrix}
A - CA & 0 & B - CB \\
-Tx & 1 & 0 \\
K_c & -K_f & 0
\end{bmatrix}
\begin{bmatrix}
x(k) \\
y(k)
\end{bmatrix} \\
y(k) &= \begin{bmatrix} y_{ref}(k) & f_{pvea}(k) & y_{pos}(k) \end{bmatrix}^T
\end{align*}
\]

(13)

(14)

Danfoss programming environment supports visual dataflow programming (like in Simulink or LabView) and textual programming options. Our choice for the present project is visual programming style. Hence Figure 9 shows implementation of vector matrix multiplication in PLUS 1 IDE with fixed-point arithmetic. The blocks on the figure relate to one of two functions – resource and control.

Figure 9: LQR with Kalman filtering controller implementation

For example multiplication and summation blocks have resource function while index counter and register reset Boolean switch block have resource function within the system. A start signal is triggered in the beginning of the sample period to execute the calculation in the next step. The index counter increments through indices of the matrix which is represented as one dimensional row-stacked array. Based on the current index corresponding elements from the matrix and the vector are selected to be multiplied. Since they are represented as fixed point numbers with scaling 1000 after multiplication the result is divided by 1000. Multiplication results corresponding to the given matrix row are accumulated within register formed by sumator element and positive feedback. Accumulator reset is triggered by the increment of the current row index. Rest of the blocks controls formation of the input vector array.

Figure 10 shows experimental response of the cylinder piston during periodic step reference trajectory in both directions relative to the central position. Recorded reaction of the cylinder piston is aperiodic with setting time around 35 seconds for 1/3 of the piston stroke. This is acceptable for low speed steering of heavy duty machines. During the motion to the final position the piston stops for a certain amount of time in some intermediate positions. The reason is relatively large dead zone of EHSU with PVE module due to its constructive specifics.

There is zero error in steady state. Transitional processes are of an aperiodic nature, without overshoot. The quality of the transition processes is maintained when cylinder piston moving in both directions.

Figure 10: Step response of closed-loop system

Figure 11 shows the measured control signal supplied by the controller to the PVE module during the experiment. The control signal approaches its maximum value during the transition process, indicating that the output reacts with the maximum possible performance boost [14]. The noise level in the control signal is low, indicating a high accuracy of the measuring sensor [15] for the cylinder position. This in turn translates as a quality of the closed-loop system.

Figure 11: Control signal to PVE block

Figure 12 shows the dynamic pressure variation in the two chambers of the executive servo-cylinder, and Figure 13 shows the difference from them. Experimental studies were performed at a fixed setting of the loading pressure (0.5MPa) set by the load system based on a hydraulic block with over-center valves (pos.11, Fig.1). This system makes it possible to realize different pressure loads in the two chambers of the servo-cylinder.

Figure 12: Cylinder chamber pressure
The variable pressure load affects the closed system as a low-frequency output disturbance. The results show the low sensitivity of the system to it. It’s an important quality because there is no need to re-set the system at different loads. This insensitivity occurs at the expense of the increased power of the control signal.

6 Conclusion

The main result of the paper is a developed system for LQG control of electrohydraulic steering system that is implemented in low speed mobile machines. A two output one input discrete time stochastic plant model is obtained by identification procedure. This model is validated by statistical tests and is used to design of LQR controller and Kalman filter. The multivariable LQR regulator with integral action and Kalman filter are designed. Appropriate software which is implemented in 32-bit microcontroller is developed. The results from experiment with developed by authors laboratory setup confirm control system performance. The embedded control system achieves the prescribed requirements: fast transient response without overshooting and static error in whole working range.

7 Acknowledgements

Current research was supported by the funding contract № DM07/7 with National Science Fund Bulgaria. Also the authors are thankful to conference organization committee for their invitation.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
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<th>Unit</th>
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<tr>
<td>$x$</td>
<td>State vector</td>
<td>[-]</td>
</tr>
<tr>
<td>$y$</td>
<td>Output vector</td>
<td>[-]</td>
</tr>
<tr>
<td>$v$</td>
<td>Model disturbance (residual)</td>
<td>[-]</td>
</tr>
<tr>
<td>$K_p$</td>
<td>Proportional term matrix gain</td>
<td>[-]</td>
</tr>
<tr>
<td>$K_i$</td>
<td>Integral term matrix gain</td>
<td>[-]</td>
</tr>
<tr>
<td>$K_f$</td>
<td>Filter matrix</td>
<td>[-]</td>
</tr>
<tr>
<td>$T_s$</td>
<td>Sample time</td>
<td>[s]</td>
</tr>
<tr>
<td>$u$</td>
<td>Input signal</td>
<td>[-]</td>
</tr>
<tr>
<td>$A, B, C, D$</td>
<td>Matrices with appropriate dimensions</td>
<td>[-]</td>
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<tr>
<td>$Q, R$</td>
<td>Matrices chosen to ensure acceptable transient response of the closed-loop system</td>
<td>[-]</td>
</tr>
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Development and implementation of a control concept for a hydraulic load unit


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Functionality and performance of novel hydraulic systems under real life stress can usually be examined in field tests only. In order to gather information about the behavior under stress during the development process as soon as possible, system components as well as systems get tested on test rigs. In hydraulics, applying passive loads e.g. to linear actuators can easily be done by throttling the outflow. For rotary units, loads can either be applied with ropes and masses or other rotary units. Especially applying active loads i.e. loads with the same orientation as the motion of the cylinder, is difficult and usually connected to a high complexity. At Karlsruhe Institute of Technology (KIT), a hydraulic load unit for hydraulic cylinders was developed to be used at various test rigs. The load units’ controller design allows for the application of either active or passive loads in variable directions and intensities. The following paper introduces the load unit, its open- and closed-loop control concept and the verification results from simulation, which show the potential of the load unit. 

Keywords: hydraulic load unit, simulation of active and passive loads, open- and closed-loop control concept

Target audience: Mobile Hydraulics, Mobile Working Machines, Test Bench Simulation

1 Introduction

Improving the efficiency of hydraulic systems is a very important task of modern system and component engineering, motivated by reduction of mobile machine emissions and protection of natural resources. /1/ Thus, various research projects focus on developing new concepts, systems and components for optimized mobile machines (e.g. /2/, /3/, /4/, /5/, /6/). Not every project leads to a fully grown mobile machine prototype and therefore test rigs are often used for testing. Benefits of test rigs are e.g. a higher reproducibility of conducted experiments and easier data acquisition compared to a field test with a mobile machine. /7/ Nevertheless, comprehensive and significant test results of a system and its behavior under real life stress can usually be conducted only during field operation tests of the machine. /8/

A load duty cycle usually consists of two types of loads: active and passive loads. In this paper, the terms active and passive loads will be used according to /9/. A load is called active (or overrunning) load, whenever the force applied to an object and the objects’ direction of motion have the same orientation (Figure 1, left). For a passive load, force and motion have the opposite direction (see Figure 1, right). For hydraulic systems, active loads need to be considered especially during the design process, because hydraulic oil can only transmit compressive loads, but no tensile loads, which can lead to unsafe or dangerous situations in operation.

Since extensive field tests usually cannot be conducted in the early stages of development, different ways of applying loads and stresses on test rigs are available.

Figure 1: Visualization of Active (left handed) and Passive (right handed) Loads, according to /10/

A very common method to apply passive loads on hydraulic cylinders is a down-stream throttling valve, e.g. a proportional pressure relief valve (PPRV). The increased outlet pressure caused by the PPRV results in a force acting on the piston, which has to be compensated by a force applied by the inlet pressure. For example, this effect is also used by lowering break valves, which help to avoid cavitation during lowering, /11/.

The advantages of this method are an easy handling and implementation. By using a bypass valve and a PPRV in both cylinder ports, passive loads can be applied in both directions. As a disadvantage, no active loads can be applied this way.

Active loads can be applied on cylinders e.g. by using test masses /12/. With masses, however, active loads can only be applied in one movement direction – since the force of the test mass is caused by gravity, the load in the opposing direction is passive. Applying exchangeable active and passive loads in both movement direction without changing the setup of the test rig is often not easy and very complex /9/.

Another, more flexible method of applying loads is using a load cylinder and a test cylinder. In this configuration, the test cylinder simulates the motion while the load cylinder applies the intended load, preferably independent of the direction of motion. /8/

In order to improve the test capabilities, the Chair of Mobile Machines (Mobima) of the Karlsruhe Institute of Technology (KIT) developed a linear hydraulic load unit which can apply a load duty cycle, both variable in force direction and magnitude and thus interchangeable between active and passive loads. The main focus of the development was on the design of the load unit and its control concept, which is required to make possible a force application independent of the motion direction of the cylinders. Also, the control concept should prevent cavitation due to load magnitude and direction changes and allow for a high reproducibility and accuracy of applied load cycles. More detailed information about the control concept can also be found in /13/.

In this paper, the developed load unit and the implemented control concept will be introduced and discussed. The results of functionality verification by simulation will be shown and discussed. At the end of paper completion, the load unit was set up on the test rig, but was not put into operation yet.

2 Functional Description

Figure 2 shows the scheme of the developed load unit and its components. It consists of two collinear hydraulic differential cylinders, henceforth called load cylinders, which can apply stress on the test cylinder. Both load cylinders and the test cylinder are connected to a mechanical carriage, see Figure 2 and Figure 3. The mechanical carriage can be moved horizontally and loaded with additional weights, e.g. metal plates. By this, additional inertia can be applied to the load unit. The flange of the mechanical carriage is exchangeable to fit multiple test cylinders. /14/ Both load cylinders can be controlled by a 4/3 proportional directional valve (4/3 PDV) with upstream pressure compensators. The load can be adjusted by two PPRVs in closed-loop pressure control. /14/ Since active loads can be applied, the load unit needs to be powered by an additional power supply during operation. At Mobima, it will be powered by a central pressure supply with closed-loop pressure control (CPS), q.v. /15/. /14/
Figure 2: Scheme of Load Unit, according to /13/, /14/

To avoid cavitation, the entire load unit system is preloaded with a pressure of 20 bar. Furthermore, the load cylinders can draw oil via a bypass feeding valve (0V2), as soon as the pressure drops below the preload pressure. This happens if the load cylinders are residing in an active load situation. In case of failure or emergency, the load unit can be disconnected from the power supply by an additional security valve (0V0) immediately. During normal operation, the security valve (0V0) is fully opened. /13/, /14/

In Figure 2, several pressure transducers (0S1, 0S3) and flow rate transducers (0S2, 0S4) are depicted. They measure the quantities necessary for closed-loop control and status monitoring. 0S1 detects the pressure in the piston chamber, 0S3 the pressure in the rod chamber of the load cylinders. 0S2 measures the flow rate from the piston chamber, 0S4 the flow rate from the rod chamber of the load cylinders.

In the current configuration of the load and test cylinders, the maximum load cylinders extension velocity is approximately \( v_{\text{ext}} = 0.109 \) m/s, the maximum retraction speed is \( v_{\text{ret}} = 0.184 \) m/s. Both maximum speed levels can be increased by changing either the installed 4/3 PDV (0V1), by using smaller load cylinders or by increasing the maximum flow rate of the CPS.

Figure 3 shows the hydraulic load unit (red), an exemplary test system (blue) and part of the central pressure supply (yellow) of Mobima / KIT. The test system consists of an electric asynchronous machine driving a hydraulic load sensing pump with variable displacement, conductive components like valves, pipes and hoses and a test cylinder. The test cylinder is connected to the mechanical carriage, which is also connected to the load cylinders of the load unit.

Since mobile machines and their hydraulic systems come in various sizes, setting up an equally scaled test system is not always possible and reasonable. Thus, scaling of system and load parameters to a viable magnitude is necessary. By respecting principals of dimensional analysis (e.g. Buckingham \( \pi \) theorem, see /16/), gathered test results of scaled models still can be deemed as valid.

Figure 3: Set-up of the Hydraulic Load Unit

3 Control Concept

Figure 4 shows a flow chart of the load unit control concept.

Previous to testing, the load has to be specified, e.g. by using a time series of the load force applied to the test cylinder. Also, various parameters such as cylinder diameters, supply pressure and maximum pressure level have to be specified, since they have an influence on the Load Calculation module.

The control concept consists of three major components: the Load Calculation module, the Closed-Loop Pressure Control (CLPC) and the Closed-Loop Flow Rate Control (CLFRC) module.
The Load Calculation module processes the load specification data from the dataset into two separate control signals—a pressure signal and a flow rate signal. The maximum resulting load force \( F_{\text{res}} \) depends on the direction of movement and thus on the pressurized piston surface (piston or ring). If necessary, the intended load force \( F \) gets limited and scaled within the permitted operation range in accordance with the specified parameters, cf. Equation (1).

\[
F_{\text{res,max/min}} = k \cdot (F_{\text{pA,max}} - F_{\text{pB,min}})
\]

\[
F_{\text{pA,max}} = \pi \frac{d_1^2}{4} \cdot P_{\text{A,max/min}} \quad \text{with } d_1 = d_{\text{piston}}
\]

\[
F_{\text{pB,min}} = \pi \frac{d_2^2}{4} \cdot P_{\text{B,min/max}} \quad \text{with } d_2 = d_{\text{rod}}
\]

\[
F \in [F_{\text{res,min}} - F_{\text{res,max}}]
\]

The resulting load force \( F_{\text{res}} \) is limited by the maximum pressure either of the loading or test cylinder. If it is assumed that for all configurations the load cylinders are always stronger than the test cylinder, the saturation depends on the test cylinder parameter \( d_{\text{piston}} \) and \( P_{\text{A,max/min}} \) and \( P_{\text{B,min/max}} \). Figure 1. To satisfy that the test cylinder movement will not be corrupted, the test cylinder force has to be larger than the resulting load force \( F_{\text{res}} \). Therefore, the factor \( k \) reduces \( F_{\text{res,max/ min}} \) to include mechanical resistance and other losses, which can reduce the test cylinder force.

Then, both necessary chamber pressures of the load cylinders are calculated by using the transformed intended load \( F \) and the pressurized piston surfaces \( A_1 = \pi \frac{d_1^2}{4} \) and \( A_2 = \pi \frac{d_2^2}{4} - d_{\text{rod}} \) with respect to the minimum load unit pressure of \( p_{\text{min}} = 20 \) bar, cf. Equation (5) and Figure 5. The calculated pressure signals \( p_{\text{pA,B,max/min}} \) then are forwarded to the CLPC module.

\[
P_{\text{pA,B,max}} = \frac{F - 2 \cdot (P_{\text{pA,min}} - P_{\text{pB,max}})}{2 \cdot A_{\text{A/B}}} + P_{\text{min}}
\]

Additionally, flow rate signals are calculated. Depending on predefined load cases, cf. Figure 5, the chamber is chosen, which has to be supplied with oil and the corresponding valve signal is forwarded to the 4/3 PDV of the load unit. The threshold between both load cases depends on the load cylinder specifications as well as on the minimum system pressure of 20 bar. In the current configuration, the threshold force is approximately 20 kN.

Subsystem CLPC compares the current values of the pressure transducers (0S1, 0S3) with the calculated pressure values from the Load Calculation module. CLPC also uses the load case signal \( L_{\text{CC1/2}}(t) \) to compensate effects in active load situations. The deviation of both comparisons is then forwarded to the implemented control system. The controller adjusts the current pressure levels of the load cylinders to the nominal pressure levels and thus adjusts the intended force magnitude and direction.

Subsystem CLFRC adjusts the oil supply of both load cylinders by using the previously specified load cases and parameters. Furthermore, the CLFRC reduces the oil requirement of the system to a minimum. This feature is necessary for the load unit as long as it is powered by a power supply with closed-loop pressure control. Due to the feeding valve (0V2) the load unit will act like an open centre system. Thus, depending on the pressure difference between CPS and load unit pressure, an unnecessary flow rate would produce energy losses as well as unnecessary heating and stress for the hydraulic fluid. The flow rate is adjusted by a closed-loop control of the 4/3 PDV (0V1) to a nominal value of at least 20 l/min, in order to avoid cavitation and to prevent a pressure collapse caused by load or movement changes.

Figure 5: Nominal Pressure Signals and Spool Position of the 4/3 PDV (0V1) Depending on Load Case, adapted from [13/]

The upper left diagram in Figure 5 shows an exemplary load progression (curve Load Cycle) and the corresponding nominal signals \( y_A(t) \) (curve Control Signal PPRV 0V3) and \( y_B(t) \) (curve Control Signal PPRV 0V4). The spool position of the 4/3 PDV (0V1) is depicted in the lower left diagram of Figure 5.

While in load case 2, the resulting load force \( F_{\text{res}} \) is oriented in extending direction caused by the pressure adjusted by PPRV 0V3. PPRV 0V4 provides the system minimum pressure of 20 bar. While the load magnitude decreases, the nominal signal of PPRV 0V3 also decreases. When the load drops below 20 kN, load case 1 is active. Thus, the nominal pressure of PPRV 0V3 is set to 20 bar, while the nominal pressure of PPRV 0V4 increases to the specified level. In this case, \( F_{\text{res}} \) acts in retracting direction. For example, in this case an extending test cylinder would be applied with a passive load (load case 1) and an active load (load case 2) during its movement.

In the upcoming sections, the modules Closed-Loop Pressure Control and Closed-Loop Flow Rate Control will be described in more detail.

3.1 Closed-Loop Pressure Control (CLPC)

The module CLPC, see Figure 6, calculates the deviation \( \Delta P_{\text{A/B}} \) from the measured pressure \( P_{\text{pA/B,current}} \) and the nominal pressure \( P_{\text{pA,B,max/ min}} \). Calculated by the module Load Calculation. Then \( \Delta P_{\text{A/B}} \) is forwarded to the appropriate PID-controller to adjust the pressure in the piston or rod chamber. The PID-controller’s output \( u_{\text{A/B}}(t) \) added to the nominal value \( y_{\text{A/B,max/ min}}(t) \) results in the control signal \( y_{\text{A/B},(t)} \), which operates the PPRVs 0V3 and 0V4. Thus, each PPRV is operated by its own closed-loop control system (CLCS), cf. Figure 6.

The example in Figure 6 shows a load cylinder with a resulting load force \( F_{\text{res}} \) in extending direction as a result of forces \( F_{\text{pA}} \) and \( F_{\text{pB}} \). If the load cylinder gets moved against the direction of \( F_{\text{res}} \), here in retracting direction, the current pressure in the rod chamber (green) \( P_{\text{pB/current}} \) can drop below 20 bar due to an unsatisfactory supply with oil caused by friction losses in components, piping and others. This affects the counter force \( F_{\text{pA}} \) and thus changes the resulting load force \( F_{\text{res}} \). Since the pressure control loop of PPRV 0V4 is not active at this moment, the rod chamber pressure cannot be raised by PPRV 0V4. In order to simulate the intended load \( F \) exactly, in this case, the deviation of the passive CLCS \( \Delta P_h \) is added to the deviation \( \Delta P_{\text{A/B}} \) via switch \( L_{\text{C1}}(t) \) (depicted as closed) right before the PID-controller, see Figure 6, thus altering the pressure in the piston chamber in an appropriate way to fit \( F_{\text{res}} \) to the intended load \( F \).

Thus, changes of load force caused e.g. by changes of movement, can be compensated by the reciprocal interaction of the passive and active CLCS, depending on load cases \( L_{\text{C1}} \) and \( L_{\text{C2}} \).
The current flow rate value \( Q_i \) and the nominal minimum flow rate are compared. The flow rate deviation \( e_{Q_i}(t) \) then is fed forward into a PID controller. Next, the output signal from the controller \( u_{Q_i}(t) \) gets converted and limited in accordance with the predefined specifications. If necessary and to fit different valve characteristics, an additional correction value can be added to the control signal \( y_{Q_i}(t) \) of the valve. In the current configuration, a linear valve characteristic was modelled and adjusted appropriately. Depending on the load case, see Chapter 3, the control signal \( y_{Q_i}(t) \) is fed to the 4/3 PDV (0V1) which then moves into the intended position 1 or 2. In case of emergency, both control signals are interrupted so that the valve returns to idle position 0, which interrupts the oil supply of the load cylinders.

A safety rule is implemented to avoid cavitation in the load cylinders caused by an active load from the test cylinder. If the PPRV flow rate drops below a value of 5 l/min (critical zone, see Figure 7), the PID controller signal gets overruled by an additional signal, causing the 4/3 PDV (0V1) to open to maximum. This results in an immediate increase of the supply flow rate supported by the power supply of the load unit. Thus, pressure breakdown in the cylinders and the accompanied problems like cavitation and sealing damage can be avoided effectively.

4 Simulation Results

The functionality of the load unit and its control concept was verified by simulation. The hydraulic system, the load unit itself and an exemplary test cylinder system were simulated by using DSHplus, by Fluidon. The control concept, see Figure 4, was modelled in Matlab / Simulink, by Mathworks. The verification was conducted by using coupled simulation between DSHplus and Matlab / Simulink, see Figure 9.
The results of the simulation using the previously described model are depicted in Figure 12 and Figure 13.

Figure 12: Nominal Load, Position and Load Deviation Trajectory, according to /13/

Figure 13: Load and Position in Detail

The cylinder position trajectories of reference and simulation have minor deviations of less than 3 % absolute, see curve $\Delta x$, Figure 12. Also changes in the movement direction can be compensated. The control concept thus allows for a high accuracy of position and velocity.

The load trajectories also have minor deviations of less than approximately 8 % absolute, which are mainly caused by changes in load and motion (curve $\Delta F_{\text{load}}$, Figure 12). The deviation peak at the beginning of the cycle is caused by initializing effects of the simulation.

Figure 14 shows the pressure curve progression in the piston and rod chamber of the load cylinders.

Figure 10: Adapting a Real Working Machine to a Test Rig Model

In Figure 11, the cylinder movement as well as the load on the boom cylinder caused by the working cycle can be seen. During retraction, the position trajectory has a negative gradient, during extension it has a positive gradient. Since the load always acts in the retracting direction, the cylinder is affected by a passive load during lifting and an active load during lowering the arm. The load trajectory shows the highest absolute loads with a magnitude of approximately 275 kN in the time ranges from 5 s to 10 s and from 16 s to 22 s. The high loads were caused by lifting the log, refer to /17/. For the rest of the time, the nominal load was caused by the kinematics of the machine itself and by dynamics.

Figure 11: Reference Cycle Data – Position Trajectory and Nominal Load, according to /17/

For verification, the boom cylinder model of a forestry crane from /17/ was used, see Figure 10. The boom cylinder was equipped with an electro-hydraulic flow-on-demand system, consisting of a variable displacement pump, an 8/3 PDV with downstream pressure compensators and an electronic control unit. In the simulation, the crane system was operated by a virtual user according to /18/. The virtual user is a closed-loop position control with weighted predictive behavior. Load and positioning trajectory were based on measurement data from /17/. The dataset was taken from the working process of the forestry crane while positioning a log and feeding it to a hydraulic debarker. The cycle had a duration of 30 s.
summing up, the simulation results show, that the hydraulic load unit, developed by Mobima / KIT, can apply a dynamic and variable load progression to a test cylinder. Passive and active loads can be applied independently of the direction of movement. By using the hydraulic load unit, more thorough and reliable assessments can be conducted on a system test rig. Also aspects of system safety and safety critical scenarios can be examined and analyzed on the test rig in an early state of a project.

5 Summary and Conclusion

Test rig experiments can only help to assess a limited number of parameters. Especially in relation to real life stress, the efforts are very high. For hydraulic actuators, active and passive loads should always be taken into account, since they usually occur during operation. In this paper, a hydraulic load unit was introduced, which allows for the application of dynamic and direction variable active and passive loads on a test cylinder with a high position and load accuracy.

The hydraulic load unit was designed with a high grade of flexibility and modularity and thus can be used for a vast range of experiments. By changing cylinders and / or valve components, the power range of the load unit can be adapted easily. In the current configuration, the load unit has a maximum extension velocity \( v_{\text{ext}} = 0.184 \) m/s, a maximum retraction velocity \( v_{\text{ret}} = 0.109 \) m/s and can apply loads in extension direction of up to \( F_{\text{ext}} = 235 \) kN and in retraction direction of up to \( F_{\text{ret}} = 400 \) kN.

A simulation of the hydraulic load unit stressing a test system showed, that the developed control concept is able to reproduce position and load trajectories at the test cylinder with an accuracy of nearly 92%. Furthermore, the load unit system can be operated without cavitation and with reduced energy losses.

The hydraulic load unit can be used to examine hydraulic systems with linear motors in a more realistic test scenario. Apart from static and dynamic analysis, also safety aspects resulting of changes in loads can be addressed on the test rig, without having a full scale prototype available. Thus, valuable and important information can be gathered about the test system and its components in a very early state of the product development process.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>A</td>
<td>Cylinder Port A</td>
<td></td>
</tr>
<tr>
<td>B</td>
<td>Cylinder Port B</td>
<td></td>
</tr>
<tr>
<td>CLCS</td>
<td>Closed-Loop Control System</td>
<td></td>
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<tr>
<td>CLPC</td>
<td>Closed-Loop Pressure Control</td>
<td></td>
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<tr>
<td>CLFRC</td>
<td>Closed-Loop Flow Rate Control</td>
<td></td>
</tr>
<tr>
<td>CPS</td>
<td>Central Pressure Supply</td>
<td></td>
</tr>
<tr>
<td>ECU</td>
<td>Electronical Control Unit</td>
<td></td>
</tr>
<tr>
<td>KIT</td>
<td>Karlsruhe Institute of Technology</td>
<td></td>
</tr>
<tr>
<td>LC</td>
<td>Load Case</td>
<td></td>
</tr>
<tr>
<td>Mobima</td>
<td>Chair of Mobile Machines</td>
<td></td>
</tr>
<tr>
<td>PDV</td>
<td>Proportional Directional Valve</td>
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</table>

Undersupply situations need to be dealt with before the system can be examined on a test rig. If not, the test rig or components could be damaged.

It can be seen, that the applied load is caused by the rod chamber pressure \( p_B \), since the piston chamber pressure \( p_A \) remains at approximately 20 bar during the entire cycle. The stable chamber pressure also shows, that the load units flow rate supply is satisfactory for the flow rate demand resulting from the load cycle and the movement of the test cylinder. A closer look at Figure 14 furthermore shows, that in the time ranges 6 s to 9 s, 11 s to 12 s, 17 s to 21 s and 28 s to 30 s, the piston chamber pressure drops below the minimum pressure of 20 bar. In those time ranges, the test cylinder always extends, thus the load cylinders require additional oil from the power supply of the load unit. Nevertheless, the pressure deviations of the rod chamber are always smaller than ± 5 bar and thus no cavitation occurs.

Figure 15 depicts the pressure and position progression of the test cylinder. Durations of active and passive loads are highlighted: white background color indicates passive, yellow background color indicates active loads.

Like on the real machine, active loads only occur during retraction of the (boom) test cylinder. The simulation also shows, that at some moments during the work cycle, the rod chamber pressure drops to values near or approximately 0 bars, which means cavitation in the cylinder or at least undersupply by the system. Those
PPRV  Proportional Pressure Relieve Valve  

\( A_{\text{cyl}} \)  Pressurized Area of the Cylinder  

\( e(t) \)  Control Deviation  

\( F_{\text{load}} \)  Load Specification  

\( F_p \)  Force at Cylinder Port A/B  

\( F_{\text{res}} \)  Resulting Load Force  

\( \Delta F_{\text{max}} \)  Maximum Force Deviation  

\( \Delta F_{\text{avg}} \)  Average Force Deviation  

\( k \)  Safety Factor  

\( L_C(t) \)  Load Case Signal  

\( p \)  Pressure  

\( p_{\text{meas}}(t) \)  Measured Pressure  

\( p_{\text{set}}(t) \)  Predicted Pressure Signal  

\( \Delta p \)  Pressure Deviation  

\( Q \)  Flow Rate  

\( u(t) \)  Controller Output Value  

\( v_{\text{max}} \)  Maximum Velocity of the Cylinder’s Piston  

\( x \)  Movement Direction  

\( y(t) \)  Control Signal  

\( y_{\text{set}}(t) \)  Nominal Value  

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Fault-Tolerant Control of a Multi-Outlet Digital Hydraulic Pump-Motor

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Fault tolerance is the most important feature in safety-critical applications, including aircraft flight controls, nuclear systems, and medical devices, but it is a desirable property of any mechatronic system. In this paper, the fault tolerance of a multi-outlet digital hydraulic pump-motor is studied. This machine has actively controlled on/off valves to independently connect each piston to the tank or one of its outlets. Furthermore, the pump-motor can control an actuator directly without having directional control valves in the system; thus, the on/off control valves of the machine are the most vulnerable components of failure. A valve can either become jammed on (not able to close) or off (not able to open), whether the fault is electrical or mechanical. The effect of a defective valve is studied through simulations, and a method for fault compensation is proposed with a control algorithm adapted for each fault case. The simulations and experimental results show that the valve faults can be effectively compensated for by reconfiguring the software. Only slight degradation in the control performance can be expected.

Keywords: Digital hydraulic power management system, energy efficiency, fault tolerance, motor, pump, transformer

Target audience: Digital Hydraulics, Mobile Hydraulics

1 Introduction

1.1 Fault-tolerant hydraulic systems

Safety-critical applications, such as aircrafts and nuclear power plants, have redundant hydraulic systems. For example, Boeing airplanes have three independent hydraulic supply systems for their flight actuators: left, right, and centre /1/. Either the left or the right system can be replaced by the centre system if hydraulic power is lost. Even if all three systems fail to produce hydraulic pressure, a ram air turbine-driven pump can ensure a safe landing.

Fault-tolerant remote handling operations at ITER (an international nuclear fusion research and engineering megaproject) were studied in /2/. The water hydraulic manipulator under investigation has a redundant servo valve system. A faulty valve can be isolated from the hydraulic circuit, and the operation is continued using the redundant valve.

Usually, the hydraulic systems of mobile working machines are considered less safety-critical. However, component failures may cause economic losses for the owner if the fault prevents the machine from operating. Digital hydraulics is a solution toward more reliable machines, as it utilizes simple on/off components. Moreover, the parallel connected components make digital hydraulics configurable. Hence, a single component failure can be compensated for with an intelligent control algorithm. The fault tolerance of a digital flow control unit (DFCU) was studied in /3/. A basic principle of compensating for a single valve fault is to reconfigure the controller. In case of an off-jammed valve, the controller excludes all DFCU states related to the faulty valve. On the other hand, the distributed valve system allows also compensation for the effect of an on-jammed valve: the additional volume flow can be cancelled by using another flow path.

In digital displacement machines, the control valves are switched on and off at high frequency; thus, it is probable that the valves fail in process of time /4/. Therefore, fault tolerance of a digital hydraulic power management system (DHPMS) is worth studying. This research focuses on compensation of a single valve fault in a multi-outlet pump-motor. Methods for identification of the valve faults are not considered in this study. However, similar approaches (based on electrical and pressure measurements) that are presented in /5, 6/ can also be applied for the digital pump-motors.

1.2 Digital power management

Digital displacement machines have increasingly been studied in recent years. Artemis Intelligent Power Ltd is a pioneer of the digital pumping technology /7, 8/ and they have interested other institutes to study and apply the technique as well /9, 10, 11, 12/. A multi-outlet digital hydraulic pump-motor, the DHPMS, has actively controlled on/off valves to connect the pumping/motoring pistons independently to the tank or one of the outlets /13/. Thus, the volume flows at the outlets are controlled with the required number of pistons while the remainder are left to idle. This piston-by-piston control of digital machines also enables good efficiency at partial displacement; thus, the overall efficiency of a digital pump-motor can reach over 0.9 for the entire operating range /14/. Additionally, the multi-outlet solution enables the hydraulic energy to be transferred from one outlet to another without reducing the efficiency, as verified by the first prototype /15/. As both the pressure and flow can be altered at the outlets, the DHPMS can operate as a hydraulic transformer.

The DHPMS can be used for controlling the system pressures, whereas the actuators are controlled by directional valves. The ability to control several outlet pressures is profitable, especially in multi-actuator systems, because the supply pressure of each actuator can be set according to the load pressures independently of one another. Thus, the pressure-matching losses that are typical for traditional load-sensing (LS) hydraulics can be avoided /16/. Alternatively, the DHPMS can directly actuate a hydraulic cylinder by controlling the volume flows at the outlets. This approach comes with small hydraulic losses and the system is capable of full-scale energy recovery /17/. Moreover, a hybrid system can be implemented by attaching a hydraulic accumulator to one outlet of the DHPMS. The accumulator can be used as an energy source/sink, and allows the prime mover to be a smaller size /18/.

However, the good performance and efficiency of the DHPMS places high demands on the on/off control valves, which must have 1) high durability, 2) leakage-free construction, 3) fast response time with low variance, 4) high flow capacity, and 5) low electric energy consumption. In addition, the geometrical displacement of each individual pumping/motoring piston must be small to avoid excessive flow and torque oscillations. Therefore, a large number of pistons is needed or a high rotational speed must be used to achieve good controllability and sufficient volume flow /19/.
Displacement control was used to control the lift cylinder of a small excavator boom directly using a 6-piston DHPMS in [17]. Pumping and motoring modes (T, A or B) are selected for each DHPMS piston once per revolution. Hence, six mode-selection instants occur during one revolution, as the pumping and motoring modes are selected simultaneously for a pair of pistons. The selected modes realize when the corresponding valve (T, A or B) for each piston is opened in the beginning of the pumping/motoring stroke. Figure 1 shows the connection between the pumping/motoring pistons and the actuator via on/off control valves.

A control block diagram for the mode selection is shown in Figure 2. Targets for the cylinder fluid volumes A and B are calculated according to the velocity reference of the actuator (\(v_{\text{ref}}\)) and effective piston areas (\(A_A\) and \(A_B\)). The cumulative volume for the piston side is

\[
V_{\text{vol}A} = \int v_{\text{vol}A} \cdot A_A
\]

whereas the equation for the rod side volume is

\[
V_{\text{vol}B} = \int -v_{\text{vol}B} \cdot A_B
\]

Additionally, extrapolation is used to estimate the target volumes at the end of the pumping/motoring cycle. Thus, a faster response can be achieved by assuming that the velocity reference and rotational speed are unchangeable between the sequential mode-selection instants. Fluid volume errors at the outlets (\(V_{\text{err}A}\) and \(V_{\text{err}B}\)) are further determined as differences between the target volumes and volume estimates (\(V_{\text{vol}A}\) and \(V_{\text{vol}B}\)).

The resulting errors in fluid volumes are calculated for every mode combination, as stipulated in Table 1 where \(\text{disp}\) is determined hydraulic capacitance for the pumping cylinder. When calculating the change in cumulative fluid volume estimates (\(V_{\text{vol}A}\) and \(V_{\text{vol}B}\)) according to the decided modes. For example, if the piston is filled from the tank and it is decided to pump to outlet A, the new fluid volume estimate for that outlet is calculated as

\[
V_{\text{vol}A}(k) = V_{\text{vol}A}(k-1) + V_{\text{disp}} - C_h \cdot (p_A(k) - p_T(k))
\]

where \(C_h\) is determined hydraulic capacitance for the pumping cylinder.

The final pumping and motoring modes, \(M = [M_0, M_4]\), are decided after a back-pressure check. A non-load chamber of the actuator is also kept pressurized for the system stiffness and to avoid cavitation. Based on the former research, the changes in actuator-loading and the leakages through the DHPMS can also be effectively compensated for by controlling the back-pressure [19].

The actuator pressures have user-defined limits and the pressure is raised or lowered according to the rules defined in Table 2. In general, the pressure can be raised by pumping to the expanding chamber or by restricting flow from the contracting chamber. For example, if the cylinder (actuator) pressure goes below the user-defined minimum pressure while the cylinder is extending, \(p_{\text{back}} = p_B\), motoring from outlet B is restricted by selecting \(M_0 = \text{B}\). The rules for lowering the pressure are the opposite. For example, if the cylinder pressure exceeds the user-defined maximum pressure while the cylinder is stationary or extending, \(p_{\text{back}} = p_A\), motoring is enforced from outlet B by selecting \(M_0 = \text{B}\).

In this study, the controller is implemented using MATLAB/Simulink, whereas the control algorithm is processed by dSPACE DS1105 PPC board. The sampling rate of 20 kHz is used to ensure accurate valve timing. A DS4001 digital I/O board controls the valves.
2 Proposed fault-tolerant control method

2.1 Effect of an uncompensated valve fault

Two kinds of control-valve faults are considered: a valve is jammed fully in either the on-position (not able to close) or the off-position (not able to open). Additionally, it is assumed that only one defect can occur at a time. Hence, there are six different fault cases for the studied DHPMS:

- Inlet T-valve jammed off
- Outlet A-valve jammed off
- Outlet B-valve jammed off
- Inlet T-valve jammed on
- Outlet A-valve jammed on
- Outlet B-valve jammed on

An off-jammed valve causes the DHPMS chamber pressure to rise excessively during the pumping stroke. A pressure-relief valve should therefore be installed in each pumping piston of the DHPMS. During the motoring stroke, an off-jammed valve prevents the chamber from being filled with oil and causing cavitation. If not compensated, an off-jammed outlet valve also affects the actuator behaviour due to incomplete pumping or motoring strokes. Jamming on again causes energy-wasting cross-flow through the valves and can make the system uncontrollable.

2.2 Fault compensation

Generally, an off-jammed valve means that the inlet or outlet related to the valve can no longer be used. On the other hand, an on-jammed valve means that the inlet or outlet related to the valve must be used constantly. Hence, compensation of an off-jammed valve rules out the port of the defective valve, whereas compensation of an on-jammed valve closes the remaining ports in the related piston. The valve faults also decrease the effective geometrical displacement of the DHPMS.

The studied valve faults and actions for fault compensation are shown in Figure 3. A cross (X) on top of a control valve indicates the flow path that cannot be used, whereas a circle (O) indicates the forced valve position after fault compensation. Figure 3a shows an off-jammed T-valve. Thus, outlets A and B can still be used for that piston. Similarly, inlet T and outlet B can be used in the case of an off-jammed A-valve (Figure 3b), and an off-jammed B-valve prevents operation of that outlet (Figure 3c).

Table 3. Volume error calculation for the mode combinations in case of an off-jammed A-valve in a pumping piston.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Mode</th>
<th>Total fluid volume error</th>
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<tbody>
<tr>
<td>T T</td>
<td></td>
<td>$V_{err} =</td>
</tr>
<tr>
<td>A T</td>
<td></td>
<td>$V_{err} = \infty$</td>
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<tr>
<td>T B</td>
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<td>$V_{err} =</td>
</tr>
<tr>
<td>A B</td>
<td></td>
<td>$V_{err} = \infty$</td>
</tr>
<tr>
<td>T A</td>
<td></td>
<td>$V_{err} =</td>
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<tr>
<td>B A</td>
<td></td>
<td>$V_{err} =</td>
</tr>
<tr>
<td>A A</td>
<td></td>
<td>$V_{err} = \infty$</td>
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<tr>
<td>B B</td>
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<td>$V_{err} = \infty$</td>
</tr>
</tbody>
</table>

Table 4. Volume error calculation for the mode combinations in case of an on-jammed A-valve in a pumping piston.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Mode</th>
<th>Total fluid volume error</th>
</tr>
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<tbody>
<tr>
<td>T T</td>
<td></td>
<td>$V_{err} = \infty$</td>
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<tr>
<td>A T</td>
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<td>$V_{err} =</td>
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<td>B B</td>
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<td>$V_{err} = \infty$</td>
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Figure 3d shows an on-jammed tank valve. In this case, A- and B-valves must be kept closed to prevent cross-flow; hence, the piston is continuously connected to the tank. Correspondingly, the piston is constantly pumping to and motoring from outlet A when that valve is jammed on (Figure 3e). On the other hand, T- and A-valves are always kept closed in case of an on-jammed B-valve (Figure 3f).

2.3 Controller adaptation

In order to minimize the influence of a valve fault on system performance, the control algorithm must adapt to the limited controllability. A certain fault has an effect on the optimal mode selection, as well as on the back-pressure control. The rules for the fluid volume error calculation in case of an off-jammed A-valve in a pumping piston are shown in Table 3. As the fluid cannot be pumped to outlet A, each related mode combination results in “infinite” volume error. Additionally, simultaneous pumping and motoring for outlet B is prohibited, as the inlet can be used for idling. Hence, the best mode combination is selected from among the remaining five choices.

Correspondingly, mode combinations [T, A], [B, A], [A, A], and [B, B] would be banned if a motoring stroke was considered. The rules for compensate an off-jammed valve are also similar in the case of valves T and B. However, mode combinations [A, A] and [B, B] are exploitable when the T-valve is jammed off. Between these two combinations, the one connected to the back-pressure is chosen when idling.
4 Simulation results

The system is modelled using MATLAB/Simulink and the SimMechanics Toolbox, based on the principles presented in /16, 17/. The lift cylinder is controlled by the DHPMS without using position feedback. Figure 5 shows the system behaviour in normal operation; thus, no faults exist. The boom is first lifted by completing a retracting movement of 0.2 m. At the end of the trajectory, the boom returns to its original orientation. Figure 5a shows that the position tracking of the piston is good, as is the accuracy of the velocity tracking (Figure 5b); only slight oscillation can be seen in piston velocity. The cylinder-side pressure (back-pressure) is approximately 3 MPa, while the pressure in the rod-side (load pressure) is approximately 12.5 MPa as shown in Figure 5c.

The rotational speed of the electric motor is set to 850 r/min to produce sufficient flow even if only five out of six DHPMS pistons can be used. During the movement, the rotational speed has a slight ripple due to torque variation in spite of the attached flywheel (Figure 5d). Figure 5e shows the input and output powers of the system: the cylinder power is approximately 2 kW during boom lifting, and approximately −1.5 kW when the boom is lowered. The power output by the electric motor has quite a large ripple, but the average is close to the cylinder power. However, the model does not consider mechanical friction losses in the DHPMS. The losses caused only by compressibility of the fluid and the pressure drop in the on/off control valves have been taken into account on the DHPMS model. The cumulative energy outputs by the electric motor, DHPMS, and lift cylinder are shown in Figure 5f. At the end of the trajectory, the work done by the cylinder is 0.4 kJ, whereas this value is 0.5 kJ for the DHPMS and 1.3 kJ for the electric motor. Thus, the total energy loss of the system is 0.9 kJ for the studied trajectory, and the corresponding hydraulic loss is 0.1 kJ. The utilization rate of DHPMS outlets is shown in Figure 5g. The rate is expressed as selected pumping/motoring modes per revolution, so the rate has six levels. It can be seen that the maximum rate is ±0.83, which means that full pumping/motoring is not required for the trajectory.
Figure 6 shows the simulated position and velocity tracking without valve fault compensation. An off-jammed T-valve causes a shorter piston movement of the cylinder as the controller calculates the fluid volumes, assuming that each pumping and motoring stroke is completed (Figure 6a). The effect of an off-jammed A-valve (Figure 6e) or B-valve (Figure 6i) is similar; the piston position falls behind its reference value. The missed pumping and motoring strokes obviously also influence the velocity tracking, as shown in Figures 6b, 6f, and 6j.

An on-jammed valve even has a drastic impact on the cylinder piston velocity. Due to cross-flow between the outlets and the tank, an on-jammed T-valve makes the velocity uncontrollable, as shown in Figure 6d. The cylinder piston also reaches its front end as the load pressure drops (Figure 6c). An on-jammed outlet valve has similar impact to the velocity, as shown in Figures 6h and 6l. However, oscillation is more intensive because the cross-flow occurs more often than in cases of on-jammed inlet valves. However, the influence on the cylinder piston position is the same, and eventually the front end is reached.

Figure 7 shows the trajectories after fault compensation. The simulations show that, regardless of a valve fault, the position tracking performance is as good as it is during normal operation (Figures 7a, 7c, 7e, 7g, 7i, and 7k). In the case of off-jammed valves, the velocity curves are also smooth, as shown in Figures 7b, 7f, and 7j. However, jamming on causes a slightly larger ripple to the velocity (Figures 7d, 7h, and 7l). The ripple is at its largest in cases where an outlet valve is jammed on, as the fluid is forced to pump to and motor from that outlet, independent of the movement. Nevertheless, the velocity is zero during idling, as the volume flow resulting from the on-jammed outlet valve is compensated for using the reverse DHPMS piston.

5 Experimental results

The system was also studied experimentally. Figure 8 shows the boom lifting/lowering cycle in normal operation, with no active faults. Position tracking is good, considering that the position is controlled open-loop, as shown in Figure 8a. Slight oscillation can be seen in velocity, but the tracking is satisfactory (Figure 8b).
Figure 8c shows the cylinder pressures during the trajectory. It is notable that the pressures decrease between the movements due to leakage through the DHPMS control valves. The rotational speed of the electric motor is set to 850 r/min, as shown in Figure 8d. At the end of boom lowering, the rotational speed drops due to back-pressure control. Figure 8e shows that the cylinder power is approximately 2 kW during boom lifting, but −1.5 kW when the boom is lowered. Correspondingly, the electric motor outputs 5.5 kW at maximum, while the minimum is approximately −1 kW. At the end of the measurement, the work done by the cylinder is 1.1 kJ, whereas this value is 1.7 kJ for the DHPMS and 0.8 kJ for the electric motor, as shown in Figure 8f. The utilization rate of the outlets remains below 1; hence, full pumping/motoring is not required to produce sufficient volume flows. The black (+) signs show the mode-choosing instant, when the back-pressure control interferes due to the leakages.

The fault compensation is tested such that the valve faults are created with software and the adapted controller is utilized during the experiments. Figure 9 shows the piston movement and velocity for the fault cases: an off-jammed T-valve is shown in the first row, an on-jammed T-valve in the second row, an off-jammed A-valve in the third row, an on-jammed A-valve in the fourth row, an off-jammed B-valve in the fifth row, and an on-jammed B-valve in the sixth row. Position tracking is good in Figures 9e, 9f, and 9g. In cases of an off-jammed T-valve (Figure 9a) and an on-jammed A-valve (Figure 9g), the positioning at the end of the trajectory is less accurate, as the back-pressure controller cannot compensate for the leakage from the rod side to the cylinder side. The velocity curves hardly differ from one another. However, the on-jammed valves (Figures 9d, 9h, and 9i) cause somewhat coarser velocity tracking than do the off-jammed valves (Figures 9b, 9f, and 9g).

6 Analysis of the results

The simulations show that an uncompensated off-jammed valve impairs the position and velocity tracking performance of the studied system, as the unbalanced pumping/motoring induces a jerky motion of the HF cylinder piston. An on-jammed control valve has an even more drastic impact on the boom, as the controllability can be completely lost. The cross-flow between the actuator and the tank line causes the load pressure to drop rapidly, thus, the cylinder piston or the boom moves towards its end position.

The impact of faults can be effectively compensated for by the controller adaptation: the position and velocity tracking remains good despite a faulty control valve. The simulation results showcase leakage-free control valves; hence, the positioning accuracy is as good as in normal operation. However, compensation of on-jammed valves causes a slighter lower accuracy to the actuator velocity than in the normal operation.

The experimental tests verify the simulation results, showing that the valve faults can be effectively compensated for by using the controller adaptation. However, the leakages of the test system affect the position tracking accuracy. Figure 10 shows the relative root mean square error (RMSE) for the piston position in each test. An absolute RMSER for the normal operation is 0.1 mm, which is used as a reference value. Clearly, the error becomes large in cases where the DHPMS piston is constantly connected to port A. The leakage from the rod side to the cylinder side reduces the back-pressure, therefore, the leakage cannot be compensated for by the back-pressure control during the active fault.

The relative errors for the system are shown in Figure 10b. The errors increase 30% when the faulty valve connects the DHPMS piston to the load pressure, as the fault is compensated for by running the source piston to that outlet as well, both the mechanical-hydraulic and volumetric efficiency of the DHPMS are decreased. In the remainder of cases, however, the increased overall errors are within 4%.

7 Conclusion and future work

Digital hydraulic systems possess natural redundancy, as this technique uses parallel connected components. Thus, the systems are fault-tolerant as long as the fault can be detected and identified. The key is to incorporate the controller to minimize the effects of the existing fault. Normally, the operation can be continued despite slightly degraded performance.
The fault-tolerant controller was validated using both simulations and experimental tests. The results showed that all valve faults could be effectively compensated for by using the adapted controller. Thus, the system is naturally fault-tolerant with no changes to the hardware needed. Moreover, the fault compensation requires no extra computing power compared with the normal operation. The leakage of the control valves, however, degraded the position tracking accuracy in some fault cases. The simulation model, which assumed leakage-free control valves, gave a better result: neither off-jammed nor on-jammed valves decreased the control performance, provided that the reconfigured controller was used.

In most cases, a defective valve in the multi-outlet digital hydraulic pump-motor decreases the potential volume flow at the outlets. The studied trajectory for the cylinder piston was selected such that the required maximum volume flow was about 80% of the maximum capacity. Thus, the produced flow remained adequate even though only five pumping/motoring pistons out of six could be used. Otherwise, the rotational speed of the electric motor should have been controlled as well. Hence, control of the rotational speed should be considered in future studies. Furthermore, applicability of existing fault detection methods should be verified.

In summary, displacement control using a multi-outlet digital pump-motor has shown potential for the development of more energy-efficient machinery. Despite challenges related to the new technology, these digital machines are worth studying as their programmability offers new capabilities, including features related to fault tolerance.

**Nomenclature**

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_{h}$</td>
<td>Cylinder blind end area</td>
<td>[m$^2$]</td>
</tr>
<tr>
<td>$A_{e}$</td>
<td>Cylinder rod end area</td>
<td>[m$^2$]</td>
</tr>
<tr>
<td>$C_{h}$</td>
<td>Hydraulic capacitance</td>
<td>[m$^3$/Pa]</td>
</tr>
<tr>
<td>$M$</td>
<td>Final control mode vector for a pair of pumping/motoring pistons</td>
<td>[-]</td>
</tr>
<tr>
<td>$M_{ref}$</td>
<td>Preselected control mode vector for a pair of pumping/motoring pistons</td>
<td>[-]</td>
</tr>
<tr>
<td>$n_{DHPMS}$</td>
<td>Rotational speed of the DHPMS axis</td>
<td>[r/s]</td>
</tr>
<tr>
<td>$p_{A}$</td>
<td>Cylinder A-chamber pressure</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$p_{B}$</td>
<td>Cylinder B-chamber pressure</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$p_{back}$</td>
<td>Cylinder back (non-load) pressure</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$p_{T}$</td>
<td>Tank (DHPMS inlet) pressure</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$V_{Cyl,A}$</td>
<td>Cylinder A-chamber fluid volume</td>
<td>[m$^3$]</td>
</tr>
<tr>
<td>$V_{Cyl,B}$</td>
<td>Cylinder B-chamber fluid volume</td>
<td>[m$^3$]</td>
</tr>
<tr>
<td>$V_{disp}$</td>
<td>Geometrical displacement of pumping/motoring piston</td>
<td>[m$^3$]</td>
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<tr>
<td>$V_{err,A}$</td>
<td>Cylinder A-chamber fluid volume error</td>
<td>[m$^3$]</td>
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<tr>
<td>$V_{err,B}$</td>
<td>Cylinder B-chamber fluid volume error</td>
<td>[m$^3$]</td>
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<tr>
<td>$V_{chf,A}$</td>
<td>Cylinder A-chamber target fluid volume</td>
<td>[m$^3$]</td>
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<tr>
<td>$V_{chf,B}$</td>
<td>Cylinder B-chamber target fluid volume</td>
<td>[m$^3$]</td>
</tr>
<tr>
<td>$v_{ref}$</td>
<td>Cylinder piston target velocity</td>
<td>[m/s]</td>
</tr>
</tbody>
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**References**


Design of Control System for Independent Metering Valve

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An independent metering valve control system (IMVCS) controls the meter-in and meter-out orifices of a valve independently. This innovative structure achieves a better energy saving performance, but also requires a more complex control algorithm. A flow and pressure coupling control system is proposed to control both the flow rate of the load and the pressure in each chamber. A DSP controller with TI-RTOS real-time operating system and complex control algorithm. A flow and pressure coupling control system is proposed to control both the flow rate and pressure coupling control.

Keywords: Independent Metering, Control System, Two Level Fuzzy PID, Coupling Control
Target audience: Independent Metering Control, Control System

1 Introduction

Hydraulic systems are widely applied in many industrial applications because of their high power and force to weight ratio. Usually, proportional directional valves are used in conventional hydraulic systems to realize the desired flow direction and flow rate control. With this kind of hydraulic system, the meter-in and meter-out orifices are mechanically connected, which makes the actuator easier to control. However, the conventional valve control system brings high energy consumption and throttling temperature increasing. In order to overcome these shortcomings, a new hydraulic technology named independent metering valve control system (IMVCS) is proposed, as shown in Figure 1. This innovation breaks the mechanical linkage of the meter-in and meter-out orifices, so that more additional degrees of freedom can be achieved. Therefore, the pressure and flow can be coupling controlled at desired values to realize a better energy saving and control performance of hydraulic systems.

Figure 1: Independent metering valve control system.

Jan Ove Palmberg from Linköping University first proposed the concept of independent metering control /2/ according to the cartridge valve control theory presented by professor Backé /3/. Aardema used two conventional four way proportional valves to control the meter-in and meter-out orifices respectively /4/, which was an expensive method since it replaced one by two. Solutions for the function of IMVCS devices were also patented with four poppet valves by Caterpillar Inc /5/, Moog Inc /6/ and Husco International /7/. Yao, B focused on the energy saving performance of IMVCS with five programmable valves, four of which were used for the independent metering function and the last for flow regeneration, and realized a good performance in trajectory following precision /8/, /9/, /10/. Dresden University of Technology studied the structure of IMVCS, and parallel and series arrangements were discussed /11/, /12/. Linjama M used digital flow control units (DFCU) to replace the traditional spool valves, and the structure of DFCU had the function of independent metering control /13/, /14/. Zhejiang University investigated the multi-mode control method of IMVCS to improve the precision and efficiency of the system /15/. Industrial applications developed by Eaton /16/ and Danfoss /17/ were used in mobile machinery. Clearly, the IMVCS has existed for a long time, but it still hasn’t been widely used in the practical hydraulic systems. The main reason is the increased degrees of freedom require more complex control algorithms and more expensive equipment /18/, which limits its real application.

This research focuses on the programmable control system of IMVCS. An embedded controller is designed, and a real-time multitasking control software is also developed. Built on the TI-RTOS embedded operating system and toolset, a lookup table algorithm is developed for flow control and a two-level fuzzy PID control algorithm is applied to modify pressure in real-time, so that the function of coupling control and fast response can be achieved.

2 Programmable Control System Design

2.1 Design of IMVCS controller

The controller of an IMVCS requires more powerful hardware performance and more optimized software algorithms than a conventional valve controller, since it needs to control two valves at the same time. A third generation of IMVCS controller is developed to give full play to the advantages of IMVCS, as shown in Figure 2. The controller includes five main parts: the DSP minimum system, power management module, signal processing module, CAN communication module and digital drive module of a voice coil motor (VCM).

Figure 2: IMVCS controller.

A TMS320F28335 DSP is selected as the core MCU, and its clock frequency reaches 150MHz with the ability of Floating Point Unit (FPU). Benefiting from the TI-RTOS embedded operating system, the controller can realize multi-thread real time operation with a closed-loop control of 0.5ms. The power management module provides different voltage values to make sure every part of controller operates effectively: 32V for VCMs, 24V for sensors, and both 3.3V and 1.9V for the DSP minimum system. The signal processing module amplifies the differential signal from the sensors, and transfers to DSP through sampling ports. The digital drive module is developed for VCM, which is a fast response proportional actuator of the pilot stage of the valve. In order to control the output force and moving direction of the VCM accurately, a bipolar "H" type PWM changing circuit is designed, as shown in Figure 3. A logic chip is used to reverse the PWM signal from DSP. The bipolar "H" type drive circuit receives two adverse PWM signals. When their duty ratio is 50%, the effects of two driving PWM signals are cancelled out, the output force of the VCM is zero, and the pilot spool stays at zero position. When the duty ratio is not equal to 50%, the two high frequency adverse PWM signal can be regarded as an effective analog power source, and the average voltage determines the direction and values of the VCM output force.
2.2 Design of host control system

A control program for the host computer is provided by LabVIEW, the interface for which is shown in Figure 4. Through CAN bus, the host computer is able to communicate with IMVCS controller. With the help of a Wifi module, this control program can achieve remote control and monitoring of the IMVCS. With the function of selectable operation mode, the user can easily change the operation mode between displacement control, pressure control and flow rate control, so that a most suitable operation mode will be applied under any working condition to achieve a better performance. In the interface, target value and control parameters can be set, a tracking signal can be produced, and dynamic data of the IMVCS can be saved for analysis.

3 Control Algorithm Analysis

Based on TI-RTOS embedded operation system, a multi-threaded 1ms closed-loop control algorithm is programmed. Each valve has a dedicated spool displacement control thread, chamber pressure control thread, and flow rate control thread. In order to verify basic control performance of this control system, an optimized fuzzy PID strategy is applied in main spool displacement control, as illustrated in Figure 5. The output PWM is an adjustable duty ratio square wave of high frequency of 10KHZ. VCM is a fast pilot actuator with very low inductance, so a quick response will be ensured.

![Figure 5: Main spool displacement control structure.](image)

The equation of displacement fuzzy PID control algorithm can be written as

\[ u(k) = \alpha_{d_{out}} k_1 e_1(k) + k_2 \sum_{i=1}^{n} \alpha_{d_{in}} e_{i-1}(k) + \alpha_{d_{dd}} e_{d_{dd}}(k) \]  

where \( u(k) \) is the output duty ratio, \( e_1(k) \) is the error to the target displacement, \( k_1, k_2 \) and \( k_3 \) are parameters of PID algorithm, \( \alpha_{d_{out}}, \alpha_{d_{in}} \) and \( \alpha_{d_{dd}} \) are outputs of displacement fuzzy PID controller, and their rules are shown in Table 1.

<table>
<thead>
<tr>
<th>( e ) (μm)</th>
<th>( \alpha_{d_{out}} )</th>
<th>( \alpha_{d_{in}} )</th>
<th>( \alpha_{d_{dd}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-100</td>
<td>1</td>
<td>1</td>
<td>5</td>
</tr>
<tr>
<td>100-200</td>
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<td>(200-e)/100</td>
<td>(200-e)/20</td>
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<tr>
<td>&gt;200</td>
<td>0.25</td>
<td>0</td>
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</tr>
</tbody>
</table>

**Table 1:** Parameters of displacement fuzzy PID control.

Based on the operation principle of a bipolar "H" type drive circuit, the average voltage applied on the VCM can be written as

\[ U = U_{\text{sup}} \left(2u(k) - 1\right) \]  

(2)

where \( U \) is effective average voltage value, and \( U_{\text{sup}} \) is supply voltage value of the bipolar "H" type drive circuit.

In the pressure control thread, a two level closed-loop control algorithm is applied, as shown in Figure 6. A fuzzy PID control program developed especially for the pressure dynamics and stabilities is applied in the outer loop. The output results from the pressure PID algorithm module are the input target displacements of the inner displacement closed-loop control thread. Every cycle, the pressure sensor detects the chamber pressure and feeds back to the DSP system. Then, the pressure control module gives a new output for the inner displacement closed-loop control thread, and the spool moves to the target position. The above process repeats until the pressure reaches the desired value. At last, the chamber pressure can be controlled with the method of adjusting pressure difference through the valve by moving the spool.

![Figure 6: Pressure control structure.](image)

The equation of pressure fuzzy PID control algorithm can be written as

\[ Di_{out}(k) = \alpha_{p_{out}} k_1 e_1(k) + k_2 \sum_{i=1}^{n} \alpha_{p_{in}} e_{i-1}(k) + \alpha_{p_{dd}} e_{p_{dd}}(k) \]  

(3)

where \( Di_{out}(k) \) is the output displacement value to the inner displacement closed-loop, \( e_1(k) \) is the error to the target pressure value, \( \alpha_{p_{out}}, \alpha_{p_{in}} \) and \( \alpha_{p_{dd}} \) are output of pressure fuzzy PID controller, and their rules are shown in Table 2.

<table>
<thead>
<tr>
<th>( e ) (bar)</th>
<th>( \alpha_{p_{out}} )</th>
<th>( \alpha_{p_{in}} )</th>
<th>( \alpha_{p_{dd}} )</th>
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<td>0</td>
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<tr>
<td>10-20</td>
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</tr>
<tr>
<td>&gt;20</td>
<td>0.25</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

**Table 2:** Parameters of pressure fuzzy PID control.
In the flow rate control thread, a lookup table algorithm is applied. A small database of flow rates with different differential pressures, different spool displacements and different temperatures is built. As Figure 7 shows, the sensors feedback current temperature and differential pressure to the DSP, which calculates the displacement position according to the target flow rate value and feedback values from sensors. Then, the displacement control thread moves the spool to the target. The above process will be repeated until output flow rate matches the target flow rate.

The principle of the flow rate database can be described as

\[ Q = f(T, x, \Delta P) \]  

(4)

where \( Q \) is the flow rate, \( x \) is the spool displacement, \( T \) is the temperature, and \( \Delta P \) differential pressure.

Therefore, the flow rate lookup table algorithm can be improved as

\[ x = g(Q, T, \Delta P) \]  

(5)

4 Experiments and Results

An experimental IMVCS is established, and its schematic is illustrated in Figure 8. The IMVCS mainly consists of a programmable controller, a pump, a hydraulic bridge circuit which is used as a load, and two two-stage proportional valves. Experiments about spool displacement control characteristics, chamber pressure control performance and flow rate control performance have been carried out in this section.

IMVCS performance, especially where pressure control and flow rate control are concerned, is heavily dependent on spool displacement performance. Therefore, experiments are performed to determine the displacement dynamics of the developed control system. Step response of the two valves switching their operation conditions at the same time are illustrated in Figure 9. Positive displacement means the operation port is connected with the tank, while, negative displacement means the operation port is connected with the supply pressure. Response times of steps from -3000\( \mu \)m to 1500\( \mu \)m and 3000\( \mu \)m to -1500\( \mu \)m are 58.5ms and 59.3ms respectively with an allowable error and an overshoot of 1%. From the curves, it is easily observable that the two valves transition from one displacement level to the other at nearly equivalent rates, which will ensure a better moving performance of the actuator.

Experiments of using the control system to follow a displacement sine signal are also carried out to verify the advantages of the programmable control system. Figure 10 shows the results when tracking a sine signal of 10Hz with an amplitude of 0.2mm. It can be seen that the spool closely tracks the target and there is almost no reduction of amplitude, though there is an average delay time of about 6ms. The successful results of the basic displacement control experiments validate that the developed control system has excellent displacement control, which will enhance the pressure and flow control performance.
control of outlet chamber, it is usually used to improve the controllability of the actuator. During the outlet chamber pressure control experiments, the inlet spool is also controlled with an opening orifice of 3mm, and the results are illustrated in Figure 12. When tracking a step signal from 5bar to 60bar, pressure can be established within 130ms, and remains stable within 250ms. Unfortunately, the closer to the critical closing position of the main spool, the more sensitive the back pressure of the chamber will be. Therefore, an overshoot of about 16% is caused by only a very small excess movement of the spool. When it comes to tracking a drop step signal, the system gives a better performance with an adjusting time of about 150ms, and no overshoot. Both the inlet and outlet chamber pressure control experiments show that the developed control system is effective in pressure control.

Flow and pressure coupling control is one of the most important functions of IMVCS, so experiments to test the system’s coupling control performance are carried out. The main purpose of this experiment is to keep the inlet pressure constant while controlling the flow rate of the outlet. The results are shown in Figure 13 and Figure 14. Because conventional flow meters cannot adequately detect transient flow, the flow data in the figures are calculated by Equation (4). Inlet port of load is controlled at 60bar, and the flow control algorithm is applied to the outlet port. Figure 13 shows the performance when tracking a flow rising step from 20L/min to 40L/min. The flow response time is about 150ms and pressure adjusting time is about 100ms. Figure 14 shows the performance when tracking a flow drop step from 40L/min to 20L/min. The flow response time is about 150ms and pressure adjusting time extends to about 300ms, which is caused by the decreasing of the opening area of the outlet valve, and makes both outlet and inlet chamber pressure increase greatly, and cause a longer time to adjust the pressure. However, in general this programmable control system has excellent performance in flow and pressure coupling control, which has huge potential for energy saving in hydraulic systems.

5 Summary and Conclusion

Although a large amount of experimental research has been done about independent metering control based on the developed control system, the theoretical analysis and real application of such systems still needs effort. From hardware to software, a programmable control system including embedded lower controller and host control system are developed for IMVCS, and a multitasking software is programed based on TI-RTOS real time operation system. A series of experiments of IMVCS are carried out, and the results show the superiority of the fuzzy PID control algorithm and lookup table algorithm. With the help of the programmable control system, IMVCS can easily achieve good performance in displacement, pressure and flow control, which can expand its application area, and provide a huge potential for energy-efficient control. Further study will concentrate on energy saving performance and controllability of the actuator.

6 Acknowledgements

This research was supported by the National Key Technology Support Program of China (Grant No. 2014BAF02B00), National Science and Technology Major Project of China (Grant No. 2012ZX04004021).
Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>$u(k)$</td>
<td>Output Duty Ratio of PWM</td>
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<tr>
<td>$e_d(k)$</td>
<td>Error to the Target Displacement</td>
<td>[μm]</td>
</tr>
<tr>
<td>$k_p$</td>
<td>Proportional Gain of PID Algorithm</td>
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<tr>
<td>$k_i$</td>
<td>Integral Gain of PID Algorithm</td>
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</tr>
<tr>
<td>$k_d$</td>
<td>Differential Gain of PID Algorithm</td>
<td>[-]</td>
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<tr>
<td>$\alpha_{d1}$</td>
<td>Output of Displacement Fuzzy PID controller</td>
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</tr>
<tr>
<td>$\alpha_{d2}$</td>
<td>Output of Displacement Fuzzy PID controller</td>
<td>[-]</td>
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<tr>
<td>$\alpha_{d3}$</td>
<td>Output of Displacement Fuzzy PID controller</td>
<td>[-]</td>
</tr>
<tr>
<td>$U$</td>
<td>Effective Average Voltage Value</td>
<td>V</td>
</tr>
<tr>
<td>$U_{sup}$</td>
<td>Supply Voltage Value of the Bipolar &quot;H&quot; Type Drive Circuit</td>
<td>V</td>
</tr>
<tr>
<td>$D_{out}(k)$</td>
<td>Output Displacement Value to the Inner Displacement Closed-loop</td>
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<td>$e_p$</td>
<td>Error to the Target Pressure Value</td>
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<tr>
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<td>Output of Pressure Fuzzy PID Controller</td>
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<td>Output of Pressure Fuzzy PID Controller</td>
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<td>$\alpha_{p3}$</td>
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<tr>
<td>$Q$</td>
<td>Flow Rate</td>
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<td>Spool Displacement</td>
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</tr>
<tr>
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<td>Temperature</td>
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<td>$\Delta P$</td>
<td>Differential Pressure</td>
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</table>

References

A Semi-Empirical Lumped Parameter Model of a Pressure Compensated Vane Pump

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This paper presents the results of a research study investigating the causes of instability in a pressure compensated vane pump system and verified it with measurements conducted on a specially built test rig. Analysis of the complete model reveals the performance limitations imposed by the control system valves in terms of system stability and achievable controller bandwidth are the most restrictive.

**Keywords:** Analysis, Control, Simulation, Pressure Compensation, Vane Pump

**Target audience:** Design Process, Systems, Components

1 Introduction

The popularity of light on-highway vehicles with automatic transmissions in the US and the nearly constant push for higher system efficiencies has created a demand for research into methods for improving the transmission performance. Currently, these automatic transmission systems often use pressure compensated vane pump architectures for higher system efficiencies. The authors developed a semi-empirical lumped parameter model of the pressure compensated pump system and verified it with measurements conducted on a specially built test rig. Figure 1 illustrates the subsystems of the developed model.

The aim of this research is not to develop a model to support pump design, but to develop a model to investigate the pump control system’s instability. To that end, the variable displacement vane pump (VDVP) model (in the dashed red box in Figure 1) is a lumped parameter model including the pump kinematics, a model of the displacement chamber (DC) and control chamber pressures, a model of the forces acting on the pivoting cam, and a dynamic system model of the pump adjustment system. The model calculates the pressure in each of the seven differently sized DC based on precise pump geometry, oil properties, and their dependence on temperature, pressure, and entrapped air as in [1]. The cam dynamics model features a single damping coefficient representing all friction and viscous damping effects as well as a nonlinear spring rate based on measurements. The pump control system belongs to the class of pressure compensated pump controls and is modelled as a black box using a set of transfer functions based on measurements. Figure 2 gives a block diagram of the entire pressure compensated pump model. The resulting DC pressure profiles were compared with measurements as shown in Figure 5. The dynamic motion of the cam was compared with measurements as shown in Figure 8.

2 Displacement Chamber Module

The kernel of the VDVP subsystem is a lumped parameter model calculating the instantaneous pressure profile of each individual DC. As presented in [1], this is accomplished by solving the pressure build up equation for the ith DC, Equation (1), derived from the conservation of mass law at each time step using instantaneous flow rates exchanged with the ports via turbulent orifice equations, Equations (2) and (3).

\[
\begin{align*}
\rho_{DC} \frac{dV_{DCi}}{dt} &= Q_{in,i} + Q_{LPI,i} - Q_{DCi} - \rho_{DC} \frac{dV_{DCi}}{dt} \\
Q_{in,i} &= \frac{A_{i}}{\sqrt{\frac{2}{\rho_p}}} \Delta p_{in,i} \\
Q_{LPI,i} &= \frac{A_{i}}{\sqrt{\frac{2}{\rho_p}}} \Delta p_{LPI,i} 
\end{align*}
\]

The calculation of the instantaneous absolute pressure \( p_{DCi} \) in Equation (1) depends heavily on an accurate representation of the DC control volume size \( V_i \) and rate of change that is accomplished by a functional analysis of the DC’s angular span \( 2\alpha_i \), angular position \( \alpha_i \), and the eccentricity angle \( \beta \). The \( V_i \) function also depends on the radius of the rotor body \( r \) and the inner surface of the cam \( R \) along with the distances between the centres of these cylindrical surfaces and the pivot centreline, \( l \) and \( L \), respectively, as illustrated in Figure 3.

Equation (1) considers a lumped external leakage flow term \( Q_{LPI,i} \) which can be determined from steady state measurements or from a very complex pump model considering the fluid structure behaviour of the pump’s various tribological interfaces. The authors determined a value for this lumped external leakage from measurements for a...
few points of operation. As Equation (4) illustrates, this was accomplished by subtracting the compression losses and internal leakage due to cross-port flow and pump kinematics from the measured volumetric flow losses $Q_0$, in order to determine the portion of the $Q_0$ attributed to external leakages $Q_{ext}$. The terms $Q_{comp}$ and $Q_{leak}$, referring to the net compression losses and unmeasurable internal leakages respectively, in Equation (4) are accurately calculated using the model represented by Equations (1-3) when the $Q_{DC,DC}$ term is neglected. As Equation (5) indicates, $Q_0$ is the difference between the theoretical flow $Q_{th}$ and the effective flow measured $Q_{eff}$.

\[ Q_{ext} = Q_0 - Q_{comp} - Q_{leak} \]  
\[ Q_0 = Q_{th} - Q_{eff} \]  

Figure 4 shows the difference in the calculated instantaneous DC pressures when the external leakage term $Q_{DC,DC}$ is both considered and neglected in Equation (1). This term, $Q_{DC,DC}$, represents the portion of the net external leakage $Q_{ext}$ contributed by each of the seven DC while in the delivery stroke and is distributed over that portion of the shaft revolution. As can be seen in Figure 4, the impact of these external leakages is negligible for the purposes of this research in calculating representative internal forces for the determination of the system stability. In fact, $Q_{ext}$ was 27% of $Q_0$ (0.8[L/min] or 2% of $Q_{th}$) for the operating conditions depicted in Figure 4.

The instantaneous flow rates from the suction port into the $i^{th}$ DC, $Q_{s,DC}$, from the $i^{th}$ DC into the delivery port, $Q_{d,DC}$, are characterized by the instantaneous orifice areas $A_{s,DC}$ and $A_{d,DC}$. These represent the minimum cross-sectional area of the flow path perpendicular to the streamlines connecting the $i^{th}$ DC with the respective port at a given $\phi$ and $\beta$ (see /1/ for more details). Having a separate set of $A_{s,DC}$ and $A_{d,DC}$ for each uniquely sized DC builds into the model timing effects, cross-port leakage flow (similar to the work in /2/), and a more realistic frequency spread of the port pressure ripple and internal pressure forces. These areas, along with their alignment with the volume function, is what gives the characteristic DC pressure profile seen in /6/ and Figure 5.

\[ Q_{comp} = Q_{s,DC} - Q_{d,DC} \]

The DC module described by Equations (1-3) was validated using dynamic pressure transducers installed in the modified pump as described in /1/. Figure 5 shows two plots from this validation study to highlight the representative nature of the simulated pressure profile. Discrepancies between the simulated and measured profiles shown in Figure 5 are attributed primarily to differences between the pump geometry from 3D CAD data used in the model and the real geometry of the modified test pump used in the experimental investigation. These variations occur in both the area files and in the $V_i$ terms and reflect the impact of manufacturing tolerances on the profile.

\[ \tau_i(\beta, \phi) = \int_{A_{DC}} \rho_\text{oil} \frac{\partial P}{\partial \beta} \, dA \]  
\[ M_{DC} = \sum_{i=1}^{n} \tau_i P_i \]  

Realistic internal forces acting on the cam are calculated with Equations (6) and (7) using this validated module to generate the realistic DC pressure profiles. The first step in these calculations is the definition of a function giving the length $\beta$ depicted in Figure 3, or the distance from the rotor centre to any point $P$ on the cam inner surface. The second step involves using $\beta$ to define the vectors $\vec{P_1}$ and $\vec{P_2}$ in Figure 3. The third step requires the numerical integration of Equation (6) for a grid of $\beta$ that spans the accepted values at a sufficient level of discretization for small increments (e.g., 0.5°) of $\beta$ over a complete revolution of the shaft. This third step produces a matrix of influence factors $\tau_i$ for the $i^{th}$ DC. Internal pressure forces are then calculated at each time step using Equation (7) for the total DC pressure induced pivoting moment acting on the cam $M_{DC}$, which is the primary moment for the adjustment system to overcome. This approach differs from that of previous researchers /5/ and provides the cam dynamics subsystem in Figure 2 with more realistic internal forces.

\[ K_{int} = \frac{K_{oil}}{1 + \rho_\text{air} \frac{\rho_\text{air}}{\rho_\text{oil}}} \rho_\text{oil} \]  
\[ \rho_\text{oil} = \frac{1}{\gamma_p \rho_\text{air} \sqrt{1 + \frac{\rho_\text{air}}{\rho_\text{oil}}}} \]

Figure 5: Selected DC pressure profile validation figures comparing simulation model results and in-house measurements shown over a single rotation of the shaft.

Figure 3: Geometric parameters used in the calculation of the DC volume and pressure forces acting on the cam inner surface.

Figure 4: Comparison of the simulated instantaneous DC pressures with and without external leakages considered.
3 Semi-Empirical Cam Dynamics Model

A critical characteristic of any simulation tool designed to aid the engineer in an analysis of pump dynamics or in the evaluation and design of pump control systems is the ability to simulate a realistic motion of the pump adjustment system, in this case the pivoting cam of the VDVP. While in theory this is simple enough to accomplish by solving the equation of motion of the cam, implementation can be rather complicated. The engineer must decide what level of model complexity to adopt to achieve their research goals. Typically, model complexity is increased proportional to the amount and types of friction terms included (see /6/ for a representative list for a pivoting-cam VDVP). To simplify the model used in this study, a single speed-dependent damping term $C_s$ encompassing all friction effects was included in the equation of motion given by Equation (10) derived as illustrated in Figure 6.

$$l_6 = \beta_4 (l_f - (l_s \sin(\beta) + l_6)) - C_s \dot{\beta} - M_{DC} + \tau_{DCP} + \tau_{SCP} \sin \beta$$  \hspace{1cm} (10)

In the final two terms on the right-hand side of Equation (10), $\tau_{DC}$ and $\tau_{SCP}$ are influence factors similar to those calculated by Equation (6) that convert the control pressure $p_D$ and atmospheric pressure $p_{atm}$ into moments acting on the cam. These influence factors were generated using projected areas from a 3D CAD model of the reference pump. The resulting cam dynamics model is “semi-empirical” due to the definitions of the damping coefficient $C_s$ and spring rate $k$ from experimental results.

3.1 Damping Coefficient $C_s$

Analysing measured cam motion using the modified VDVP described in /1/ showed an apparently first-order response to a step command. The damping coefficient $C_s$ was then calculated using the measured rise time to result in an overdamped second-order dynamics for Equation (10) with an equivalent response time when considering a nominal bias spring rate. As an added verification that the selected value is reasonable, the power associated with the sum of the friction terms represented by $C_s$ was calculated using the root-mean-squared rotational velocity of the cam $\beta_{\text{max}}$ from the same measurement data using Equation (11).

$$P_{\text{friction}} = C_s \dot{\beta}_{\text{max}}$$  \hspace{1cm} (11)

$$P_{\text{friction}} = P_{\text{pump}} - P_{\text{out}}$$  \hspace{1cm} (12)

Given a measured shaft power $P_{\text{pump}}$ and the simulated shaft power $P_{\text{out}}$ for the same operating conditions, the total friction power $P_{\text{friction}}$ is found using Equation (12). Equation (12) is valid here because the lumped parameter DC module calculates the shaft torque neglecting friction while still accounting for the torque losses attributed to pump design features and oil compressibility. Comparing $P_{\text{p}}$ and $P_{\text{friction}}$ reveals that the friction power associated with the cam is about 20% of the total friction power, which matches results in /6/ for a similar oil temperature.

3.2 Nonlinear Spring Model

A common feature of every pressure compensated pump is the inclusion of a spring to bias the pump displacement to maximum /8/. This bias spring force, arising from the spring rate $k$ and the compression of the spring between a surface on the cam and a surface on the pump case, is one of the primary forces acting on the cam as illustrated in Figure 6. The first term on the right-hand side of the equality in Equation (10) highlights that the spring force in this example depends on the distance of the spring centreline from the pivot $L_6$, the free length $l_6$ of the spring, and the compressed length $l_6$ for an eccentricity angle of zero. Note that the compression of the spring at a maximum eccentricity angle gives the spring pre-load.

While measurements of the spring force as a function of the compression of the spring on a separate test stand indicate a constant linear spring rate (the nominal spring rate), various distinct values for $k$ were required for the simulated cam dynamics defined by Equation (10) to match measurements acquired using the modified VDVP. Two possible conclusions to explain this observation are that the description of other internal forces is inaccurate for some operating conditions or that the compression spring is behaving in a nonlinear fashion. Without changing modelling approaches for the VDVP’s displacement chambers to improve the estimation of the DC pressure induced pivot moment and using the measured control pressure $p_0$ in Equation (10), a closer look at the spring force is warranted.

Due to the pump geometry and the pivoting motion of the cam, the spring in Figure 6 does not experience compression between two parallel surfaces. Instead, the spring is subjected to both compression and bending as the angle between the surfaces varies linearly with the eccentricity. Under these conditions, the internal shear and torsional stresses in the spring would be different for a given distance between the surface centres than for the same distance between the centres of two parallel surfaces (see /9/ for the internal stresses in this case). Since it is common to model the spring force as acting at a point (in this case the centre of the cam surface in contact with the spring), the use of a constant spring rate may not be appropriate to capture the real response of the spring.

Simulation results improved, for each measurement in the study, when a progressive spring rate as a function of $\beta$ given by Equation (13) was used to calculate the value of $k$ in Equation (10). Equation (13) results in a spring rate that increases from the nominal value to a maximum almost sixty percent greater at the minimum compressed length. Because Equation (13) was derived in an iterative fashion comparing simulation results to various measurements, this $k$ completes the semi-empirical cam dynamics model given by Equation (10).

$$k = 21074\beta^2 - 2993.4\beta^2 - 80.447\beta + 47.104$$  \hspace{1cm} (13)

3.3 Experimental Validation

Figure 7 gives the hydraulic circuit for the experimental setup used to measure the cam motion for various operating conditions. As indicated in the figure, three pressures (inlet $p_i$, outlet $p_o$, and control $p_0$) were measured along with the pump speed $n$ and cam displacement via a LVDT as explained in /1/. Using this setup, the cam eccentricity model was validated by comparing simulation results with measurements at the same operating conditions. Two example comparisons are included in Figure 8.
Cam Eccentricity Dynamics Validation

Figure 8: Comparison between simulated VDVP cam eccentricity profile and measured data for two validation case studies conducted as part of this research.

Figure 8 shows that the semi-empirical cam dynamics model has good agreement with the measured results. Discrepancies can be attributed to variations in the angular spacing of the vanes and in the dimensions of various pump surfaces due to manufacturing tolerances about the nominal values assumed by the model as long as the assumptions about $C_s$ and $k$ hold.

While it is definitely possible to attain an even better agreement between simulation and measurements, the purpose of this model is not to recreate a perfect numerical representation of the real physical pump. Instead, the purpose of the model developed here is to provide representative pump dynamics and best-case performance for analysing potential limitations to achievable bandwidth as well as operating-condition dependent sensitivities in the development of improved pump control concepts and not the development or refinement of the pump design. The results depicted in Figure 5 and Figure 8 are sufficient to conclude that the lumped parameter VDVP model as illustrated in Figure 1 is representative of the real system and suitable for an analysis of the control system.

4 Load Simulation

As Figure 2 illustrates, the load simulator module makes use of the pump flow rate calculated by the DC module to calculate the port and line pressures. The solution of a pressure build-up equation and turbulent orifice equation for each of the ports as given by Equations (14-18) and discussed in /1/ accomplishes this important task.

\[
\begin{align*}
  p_{LP} &= \frac{A_{v}}{V_{LP}} \int (\sum_{i=1}^{n} Q_{f} - Q_{a}) \, dt \\
  Q_{a} &= \frac{A_{v} \alpha_{a}}{\sqrt{\pi \rho_{e}}} \frac{(2\pi a_{p} - p_{LP})}{m} \text{sgn}(p_{a} - p_{LP}) \\
  p_{a} &= R_{lp} Q_{a} \\
  p_{LP} &= \frac{A_{v}}{V_{LP}} \int (Q_{in} - \sum_{i=1}^{n} Q_{o,i}) \, dt \\
  Q_{in} &= \alpha_{r} A_{vin} \sqrt{\frac{2p_{in} - p_{LP}}{\rho_{e}}} 
\end{align*}
\]

Many of the variables in these equations are self-explanatory as corollaries to the variables for the DC control volumes in the lumped parameter model described in Section 2. The variables $A_{v}$, $R_{lp}$, and $A_{vin}$ were all determined to match the measurement setup and represent the area connecting the delivery port with the line, resistance of the experimental circuit, and resistance of the filter interpreted as an area, respectively. Using these definitions, representative line pressures, including pressure ripple, are achievable and can be fed into the pump control system as illustrated in Figure 2.

5 Pump Control System Model

One automotive application for a pressure compensated vane pump is the simultaneous lubrication, cooling, and supply of hydraulic power to an automatic transmission system. Figure 9 illustrates the baseline pressure compensated pump control system for this type of application studied in this research. In it, an electrical command to the solenoid operated pressure-reducing valve (V3) sets the desired outlet pressure of the pressure compensated vane pump. This desired pressure increases for each clutch-shifting event and returns to a lower value when no between events. In summary, two important requirements of the pump’s pressure compensated control system are to modulate the pump displacement effectively to maintain a constant low pressure regardless of input speed variations and to respond quickly to step commands when required to meet system performance specifications.

Instead of developing a more complex physically based model for the involved pump control system valves, the authors decided to characterize them in a black box approach through measurements of the valve system. This black box model was derived in three parts as illustrated in Figure 10 to be able to facilitate the derivation and to identify the contribution of each aspect of the pressure compensated control system to the overall system response.

As Figure 10 illustrates, a custom valve block containing actual control circuit valves from a transmission was instrumented with various dynamic pressure transducers and LVDT on the custom test rig presented in /1/. These transducers were sampled at a rate of 2 kHz to generate signals that for determining the transfer functions $G_{1}$, $G_{2}$, $G_{3}$, and the gain $K_{s}$. These transfer functions were each assumed to represent second-order dynamics with a steady-state gain. The natural frequency, damping ratio, and steady state gain $K$ of each transfer function.
was then tuned in an iterative fashion until a good agreement with the measured output was reached when the measured input was passed through the transfer function. Equations (19-23) give the resulting transfer functions.

\[ G_2 = \frac{k_1(2.22\times10^3)}{s^2+805.3+2.28\times10^3} \]  
\[ G_{3A} = \frac{k_{3a}(2.22\times10^3)}{s^2+48.2+2.22\times10^3} \]  
\[ G_{3B} = e^{-0.032s} \frac{k_{3g}(246.7)}{s^2+25.27+246.7} \]  
\[ G_4 = \frac{k_40.7}{s+0.25} \]  
\[ G_1 = \frac{(1.823\times10^8)}{s^2+806.5\times10^3+2.349\times10^7+1.823\times10^8} \]  

When the outlet pressure \( p_o \) was held constant at different levels, different steady state gains \( K_2, K_3A, K_3B, \) and \( K_4 \) were required for model agreement. Therefore, each of these are represented in the final model as look-up tables with linear interpolation between the values. As Equation (21) reveals, a pure time delay was added to the transfer function between the command signal \( CMD \) and the regulation setting pressure \( p_c \). Equation (22) shows that \( G_4 \) was reduced to a first-order low-pass filter while Equation (23) shows that an additional fast pole was added to the lightly damped poles of \( G_1 \). When these transfer functions and gains combine according to the block diagram given in Figure 10, the result is a two-input single-output linear model as represented by Equation (24) giving the control flow \( Q_b \). The control pressure \( p_B \) is then calculated in the adjustment system dynamics model according to Equation (25) where the partial derivative is the “control piston” area and \( V_R \) is the control chamber volume.

\[ Q_B = [G_{3A}, G_{3B}, G_{40.7}] \left( \frac{p_B}{CMD} \right) \]  
\[ p_B = \frac{1}{\rho s} \int \left( Q_B - \frac{dV}{dp} \right) dt \]  

One result of this block diagram reduction is that the new transfer function \( G_1 \) contains a non-minimum phase zero at lower pressures while \( G_{1340} \) contains the pure time delay from \( G_{134} \). Therefore, both components of the final transfer function matrix present limitations to the dynamic stability and achievable bandwidth of the system as a whole. The Bode diagram shown in Figure 11 illustrates these limitations.

**6 Analysis of the System Model**

As stated in Section 1, the model described in the previous sections is useful for realistically analysing the stability and performance limitations of a pressure compensated vane pump system, as shown in Figure 9, designed for use in an automatic transmission application. Upon inspection of Figure 1 and Figure 9, it is clear that the system performance depends on both the dynamics of the pump adjustment system (the pivoting cam and bias spring) and the pump control system (regulation and solenoid valves). While Figure 11 contains ample information regarding the dynamics of the pump control system, the dynamics of the pump adjustment system described in Section 3 may still be unclear and deserve additional discussion.

![Critical Frequencies](image)

**Figure 12:** A plot of the critical frequencies associated with the cam dynamics model.

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**Figure 12** also illustrates the critical frequencies \( f_{cr} \) of the bias spring model (solid red and blue lines) as calculated using Equation (26) taken from \( 9 \). For the spring parameters in Equation (26), \( d \) is the nominal wire diameter, \( D_1 \) is the mean diameter of the spring, \( N \) is the number of active coils, \( \rho \) is the density, and \( k \) is given by Equation (13). While \( 9 \) states that Equation (26) is valid for a helical compression spring between two flat parallel plates and the bias spring’s environment in the VDVP does not fit this description, the resulting \( f_{cr} \) can still give the approximate excitation frequencies that may induce other destabilizing effects such as spring surge. An evaluation, and subsequent simulation, of this kind of nonlinear behaviour would require additional research. Sufficient to say, un-modelled dynamics at higher frequencies may contribute to instabilities in the pump adjustment system.

\[ f_{cr} = \frac{1}{2\pi} \sqrt{\frac{k}{\rho d N N_a c}} \]

(26)

Nevertheless, the limiting factor to the system performance from a control perspective is definitely the control system valves as revealed by Equation (24) and Figure 11. Even when viewing the control system valves as a passive element that merely responds as a typical pressure regulator valve the limiting element is not the pump considering the natural frequency of the cam is roughly five times the bandwidth of \( G_s \) in Equation (24). Even operating at low oil temperatures with higher viscous damping forces would the pump response time approach the dynamics of the pump adjustment system, as shown in Figure 9, designed for use in an automatic transmission application. Upon inspection of Figure 1 and Figure 9, it is clear that the system performance depends on both the dynamics of the pump adjustment system (the pivoting cam and bias spring) and the pump control system (regulation and solenoid valves). While Figure 11 contains ample information regarding the dynamics of the pump control system, the dynamics of the pump adjustment system described in Section 3 may still be unclear and deserve additional discussion.

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7 Conclusion

In conclusion, a brief analysis of the semi-empirical lumped parameter model of a pressure compensated vane pump developed in this paper leads to the following key observations:

- Complex DC leakage path models are not required for the calculation of representative DC pressure profiles, and subsequent internal forces, to a sufficiently accurate level for analysing pump adjustment system dynamics provided an accurate representation of the DC geometry and orifice areas used in Equations (1-3) is available. This modelling approach has been validated with measurement data and provides a best-case scenario for the analysis of the control system performance. This strategy is equally valid for translating-cam VDVP considering their close resemblance to the pivoting-cam VDVP modelled here.

- Due to suboptimal pump geometry and low-cost helical compression springs, the VDVP bias spring may behave nonlinearly as shown in Section 3.3. Additional research of this observed behaviour would better answer whether or not the main issue is the compression between non-parallel surfaces or large variations in the spring characteristics due to manufacturing tolerances and variations in material properties. Additional research into the bias spring in this environment may also reveal that the observed nonlinear behaviour is a function of a variable not yet considered which may necessitate more complex simulation tools to characterize the various tribological interfaces.

- The primary source of performance limitations in a pressure compensated vane pump system is the control system design and control system performance. This strategy is equally valid for translating-cam VDVP considering their close resemblance to the pivoting-cam VDVP modelled here.

It is clear from the last of these observations that the most important contributor to the overall system performance of a pressure compensated vane pump is the control system performance. Engineers designing pressure compensated vane pump systems must pay closer attention to this aspect of the control system design and be willing to incur a higher cost to implement better architectures and achieve improved system performance. As illustrated in this paper, the models used in this analysis and design process need not be exhaustive to be useful.

Nomenclature

<table>
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<th>Variable</th>
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<td>A</td>
<td>Surface area of cam between two consecutive vanes</td>
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Speed encoder frequency signal in experimental study | [Hz] |

Bias spring critical frequency | [Hz] |

Primary transfer function of the regulator valve V1 in Figure 10 | [-] |

Transfer function representation of the pressure reducing valve V2 in Figure 10 | [-] |

Transfer function from p\text{in} to p\text{c} of the valve V3 in Figure 10 | [-] |

Transfer function from CMD to p\text{c} of the valve V3 in Figure 10 | [-] |

Resultant control system transfer function from p\text{in} to p\text{c} in Figure 10 | [-] |

Pre-filter transfer function of the regulator valve V1 in Figure 10 | [-] |

Vane height | [mm] |

Mass moment of inertia of the cam | [kgm²] |

Bias spring rate | [N/mm] |

Steady-state gain of the transfer function G_{f} | [-] |

Steady-state gain of the transfer function G_{2} | [-] |

Steady-state gain of the transfer function G_{3A} | [-] |

Steady-state gain of the transfer function G_{3B} | [bar/A] |

Pilot ratio gain of the regulator valve V1 in Figure 10 | [-] |

Effective bulk modulus of the fluid/air mixture | [Pa] |

Bulk modulus of the working fluid | [Pa] |

Distance from the pivot centre to the rotor centre | [mm] |

Compressed length of the bias spring at a zero eccentricity angle | [mm] |

Free ( uncompressed) length of the bias spring | [mm] |

Distance from the pivot centre to the centre of the cam inner surface | [mm] |

Distance from the pivot centre to the centreline of the bias spring | [mm] |

Total DC pressure induced pivoting moment acting on the cam | [Nm] |

Shaft rotational speed | [RPM] |

Number of active coils in the bias spring | [-] |

Atmospheric pressure | [bar] |

Pump delivery line absolute pressure | [bar] |

Intermediate control system supply absolute pressure | [bar] |

Regulation setting absolute pressure | [bar] |

Pump control (decrease) absolute pressure | [bar] |

Instantaneous absolute pressure of the i^th DC | [bar] |

Measured pump inlet pressure from the experimental study | [bar] |
\( p_{\text{P}} \) Instantaneous absolute pressure of the pump delivery port [bar]

\( p_{\text{LP}} \) Instantaneous absolute pressure of the pump suction port [bar]

\( p_{\text{tank}} \) Absolute pressure at the opening of the pump inlet [bar]

\( p_{\text{CF}} \) Mean power loss resulting from cam friction and viscous damping [W]

\( P_{\text{friction}} \) Total pump power loss resulting from all friction sources [W]

\( P_{\text{pump}} \) Measured pump input power [W]

\( P_{\text{Sim}} \) Simulated pump input power neglecting friction effects [W]

\( Q_{\text{Comp}} \) Net volumetric pump flow lost to oil compressibility effects [m³/s]

\( Q_{\text{I}} \) Volumetric flowrate through the regulation valve or the control flow [m³/s]

\( Q_{\text{II}} \) Volumetric flowrate through the pump inlet filter [m³/s]

\( Q_{\text{ID}} \) Volumetric flowrate through the load orifice [m³/s]

\( Q_{\text{III}} \) Volumetric flowrate from the i\(^{th}\) DC into the delivery port [m³/s]

\( Q_{\text{III}} \) Volumetric flowrate from the suction port into the i\(^{th}\) DC [m³/s]

\( Q_{\text{B}} \) Total measured flow losses of the pump under steady state conditions [m³/s]

\( Q_{\text{BE}} \) Net external leakage measured at the case drain for the pump [m³/s]

\( Q_{\text{BE,DCi}} \) External leakage corresponding to the i\(^{th}\) DC [m³/s]

\( Q_{\text{ILP}} \) Net internal leakage, unmeasurable, of the pump under steady state conditions [m³/s]

\( Q_{\text{DC}} \) Theoretical flow rate of the pump for the given displacement level and speed [m³/s]

\( r \) Radius of the rotor body [mm]

\( r_{\text{air}} \) Percent entrained air by volume in the fluid/air mixture [%]

\( R \) Radius of the inner surface of the cam [mm]

\( R_{\text{sim}} \) Equivalent resistance of the experimental test circuit load [bar/m²]

\( T \) Temperature of the fluid/air mixture measured at the pump inlet [°C]

\( u \) Transducer electrical signal [V]

\( V_{\text{I}} \) Volume of the pump control chamber [m³]

\( V_{\text{II}} \) Volume of the pump delivery port [m³]

\( V_{\text{III}} \) Volume of the i\(^{th}\) DC [m³]

\( V_{\text{LP}} \) Volume of the pump suction port [m³]

\( x \) Measured LVDT linear displacement in experimental study [mm]

\( a_{0} \) Turbulent orifice flow equation discharge coefficient [-]

\( a_{1} \) Half-sector angle giving the size of the i\(^{th}\) DC [rad]

\( \beta \) Cam eccentricity angle (in [rad] or [degrees]) or pump displacement (in [%]) [rad]

\( \beta_{\text{rms}} \) Root-mean-squared angular velocity of the cam from measurement data [rad/s]

\( \alpha_{yD} \) Constant describing the change in control chamber volume with eccentricity [m³/rad]

\( \gamma_{P} \) Vector from the pivot centre to a point P on the cam inner surface [mm]

\( \gamma_{SP} \) Vector from the centre of the cam inner surface to a point P on that surface [mm]

\( \phi \) Angular position of a DC as illustrated in Figure 3 [rad]

\( \rho \) Distance from the rotor centre to a point P on the cam inner surface [mm]

\( \rho_{\text{air}} \) Density of air at STP [kg/m³]

\( \rho_{i} \) Density of the fluid/air mixture [kg/m³]

\( \rho_{\text{oil}} \) Density of the pure working fluid [kg/m³]

\( \rho_{\text{bias}} \) Density of the bias spring material [kg/m³]

\( \tau_{1} \) Influence factor converting the i\(^{th}\) DC pressure to a pivoting moment [m³]

\( \tau_{\text{RC}} \) Influence factor converting the control pressure to a pivoting moment [m³]

\( \tau_{\text{SC}} \) Factor converting the pressure in the spring chamber to a pivoting moment [m³]

\( \omega_{n} \) Natural frequency of the cam [rad/s]

References


Process-driven component adjustment on variable speed pump drives – development of a strategy to increase the overall energy efficiency

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Regarding the trend of optimizing energy efficiency and meeting upcoming regulations of energy consumption there are many ways to refine existing hydraulic drive systems. To gain more knowledge about components, combinations of those components and their interaction with the overall process, a combination of measurement, simulation and calculation of energy consumption is used to build the foundation for finding optimization approaches regarding the efficiency of electro-hydraulic pump drives. This is a three-step process focusing on the following topics: increased component efficiency, matching pump drive components and adjusted process layouts. By utilizing this strategy, a manufacturer- and customer-dedicated optimization of pump drive systems can be realized.

Keywords: Variable Speed Pump Drives, Energy Efficiency, Design Strategy, Gear Pumps
Target audience: Design Process, Displacement Units, Systems

1 Introduction

Ever-expanding commodity prices and energy costs fan the flames of the trend towards energy efficient process solutions and increased efficiencies of drive systems. The emphatic demand for CO₂ savings and the optimal utilization of energy in technical applications are already part of legally required guidelines. For manufacturing companies, energy costs are a significant part of their operating costs. Customers of, e.g. machine tools, therefore show intensified interest in power-saving solutions for their machines. The manufacturers of hydraulic drive technologies are hereby obliged to provide drive technologies for energy efficient realization of functions like feed motion, clamping operation et cetera.

In this context, novel system architectures for hydraulic drives gain increasing significance. Valve-controlled systems, as a matter of principle lossy due to their energy losses at the cross sections of the valves, are replaced by variable speed pump drives to allow a needs-based use of the system /1/. Possible savings in energy consumption, compared to conventional hydraulic drives, are estimated to about 70 to 80%, what generates significant cost advantages for the user and simultaneously reduces the ecological damage /2/, /3/.

First systems were introduced in applications such as forming, cutting and pressing processes /4/ as well as in forging or injection molding machines /5/, /6/. In the meantime, almost every manufacturer of hydraulic systems offers variable speed pump drives.

Rapid progress in the electrical drive technology have therefore promoted the fast market presence of electro-hydraulic motor-pump-units (MPU). However, there is no continuous development but an assembly of existing components more or less suitable for the individual application. Usually, this “Bottom-Up” approach leads to oversized systems and therefore misses the target of an energetic and economic optimum.

The presented work addresses this issue by providing a process-driven „Top-Down“-approach with all tools necessary. It allows more efficient process solutions that are also quantifiable and therefore even permit economic and eco-friendly decisions.

There are two possible objectives in the optimization of the energy efficiency of variable speed pump drives. For one thing, especially for the developer of a pump drive system, an increased efficiency of that drive in general is needed. On the other hand, an individual optimization regarding a specific application can be the goal. In both cases, the optimization strategy, as shown in Figure 1, is based on the three complementary approaches:

Figure 1: The three steps of optimization

In general, the system efficiency can be optimized by using components with fewer losses. In addition, there is also interest in the optimization of existing components especially for their use in applications with variable speed pumps – in particular to further reduce their losses. All these measures are combined under the topic “Efficient components”.

Beyond this fundamental requirement to use components as efficient as possible, also the best possible interaction of all pump drive components has to be ensured. An unpropitious combination can shift the point of operation of one component to an unfavorable range, causing an unnecessary decrease of the system efficiency. “Matching pump drive components” therefore is the second approach of optimization.

Finally, no optimization is complete without the inclusion of the process – the systems requirements to the MPU define its points of operation. Vice versa, it is also wise to fit the systems requirements to the strengths and weaknesses of the pump drive. This third approach is the “Adjusted process layout”.

In any case, the goal of an optimized system is to maximize the energy efficiency of a system while satisfying the other requirements.
2 Gear Pumps

2.1 Design Types

Geared machines transform energy with losses, which, as stated by the DIN ISO 4391, divide into two main types of losses. On the one hand, there are volumetric losses, which are caused by spacing, sealing gaps and pressure- or temperature induced deformation of material. On the other hand, there are hydro-mechanical losses that are caused by friction of moving parts and viscous friction as well as compressibility of the fluid.

<table>
<thead>
<tr>
<th>Type of losses</th>
<th>Definition</th>
<th>Sum of losses for a hydraulic pump</th>
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<tr>
<td>Volumetric losses</td>
<td>( \eta_{\text{vol}} = \frac{Q_{\text{eff}}}{Q_{\text{in}}} ) ( - \frac{Q_{\text{eff}}}{Q_{\text{in}}} )</td>
<td>( \eta_{\text{vol}} = \frac{Q_{\text{eff}}}{Q_{\text{in}}} ) + ( \frac{Q_{\text{vol}}}{Q_{\text{in}}} ) + ( \frac{Q_{\text{d}}}{Q_{\text{in}}} )</td>
</tr>
<tr>
<td>Hydro-mechanical losses</td>
<td>( \eta_{\text{hyd}} = \frac{M_{\text{out}}}{M_{\text{in}}} ) ( - \frac{M_{\text{out}}}{M_{\text{in}}} )</td>
<td>( \eta_{\text{hyd}} = \frac{M_{\text{out}}}{M_{\text{in}}} ) + ( \frac{M_{\text{d}}}{M_{\text{in}}} )</td>
</tr>
</tbody>
</table>

Table 1: Losses of a hydraulic pump as defined by DIN ISO 4391

Gear pumps, within the group of fixed-displacement pumps, are preferably used mainly due to their cost advantage over other types and are divided by their design into external gear pumps and internal gear pumps.

In case of this study, further pump designs are taken into consideration. On the one hand a pressure-compensated internal gear pump. Its design is based on a divided sickle, which reacts to pressure by reducing the spacing between the internal- and external gear and thus minimizing volumetric losses. On the other hand, a helical geared pump that is constructed to reduce flow pulsation and noise emission.

Figure 2: Unifilar drawing of an external and an internal gear pump

In order to measure the overall efficiency, the power losses are divided into the frictional power loss and the leakage power loss. Whereas the frictional power loss is dependent on the hydraulic pressure and the leakage power loss is dependent on the volumetric efficiency. In addition, the internal leakage can significantly be reduced by using a pressure compensated internal gear pump. However, the overall efficiency of the pump will decrease somewhat due to the reduced leakage. To overcome this, a helical geared pump is recommended due to its low pressure pulsation and noise emission.

Figure 3: Sectional drawing of a pressure compensated IGP and exploded view of a helical geared EGP

2.2 Temperature influence

Type and temperature of the oil used must be taken into consideration whilst measuring hydraulic pumps, because of its influence on the pump efficiency. The change of the pump efficiency for the usage of HLP 46 and a reduction of the oil temperature by 10°C from T1=45°C to T2=35°C is used as an example and depicted in Figure 4.

Whilst the oil temperature is reduced, its viscosity rises. Thus, the internal leakage of the pump is reduced whereby its volumetric efficiency improves. However, simultaneously the friction of the pump increases whereby its hydro-mechanical efficiency drops. In consequence, the overall efficiency of the pump increases as well as decreases depending on its point of operation.

Dynamic viscosity

\[ \mu = \rho \cdot e^{-\alpha(T - T_0)} \]

Density

\[ \Delta \rho = -\beta \cdot \rho_1 \cdot (T_2 - T_1) \]

Table 2: Dynamic viscosity and change of fluid density in dependence of temperature change

The reason for temperature dependence of the efficiency can be examined by considering the definitions of density and viscosity of hydraulic fluids as shown in Table 2. A change in the density of a fluid depends on a change in oil temperature. While the viscosity is dependent on an operating- and reference temperature.

Oil temperature near the displacement areas

Figure 5a: Ideal sensor placement

Figure 5b: Actual sensor placement

To implement temperature monitoring into the measurement of pumps, temperature sensors are needed. Figure 5a shows an ideal sensor placement, as near to the displacement chambers of the pump as possible. Due to this problematic execution, a more applicable solution of a configuration for temperature sensors is used as shown in Figure 5b.

Further details and usage of the temperature sensors for the work as described in this paper follow in the affiliated discussion.
3 Experiments

The pump studies are based on the standards defined by the ISO 4409/11/ and 12/. Exemplary, those are used as guidance to establish the actual displacement of a pump and measure pump- and system efficiency, as well as regulations on how to depict such measurement results.

3.1 Test layout and properties

On the one hand, test pumps are examined regarding their efficiency and on the other hand regarding their leakage behavior. Measurements of the efficiency produce data for the degree of energy conversion with regard to the overall operating range of the pump and the measuring unit. Boundaries of the operating range are defined by either the pump or the measurement unit as maximum pressure, volume flow, torque as well as temperature values are exceeded. The measurement of the leakage behavior serves as mean to establish the internal leakage whilst building up pressure against a closed tube end. Here internal leakage is the flow volume of oil pressed through sealing gaps from the pressurized area to the suction area.

Figure 6 depicts the schematics of pump measurements and the positioning of the used data sensors (cf. Table 3) within the system layout.

The switch valve is used to choose between leakage measurement and efficiency measurement. Switch setting A (open) enables the circular flow needed for performing the efficiency measurement while switch setting B (closed) closes the circuit to provide the setup needed to run the leakage measurement. The pressure relief valve protects the overall operating range of the pump and the measuring unit.

Figure 7: Arrangement of temperature measurement

Figure 10

3.2 Test process and analysis

In order to consider the influence of the system temperature on the efficiency of a pump, a concatenation of efficiency- and leakage behavior measurements, starting at room temperature (approx. 25°C) of the oil- and casing temperature and ending at typical operating temperature of pump drives (approx. 40.45°C), is used to collect data at gradually increasing temperature levels.

Measurement data is interpolated over a minimum of three sets of data for different temperature levels. For each measurement value at a given point of operation (a specific rotational speed and pressure), this interpolation is executed to calculate the value at the desired temperature resulting in a new data set for the desired temperature.

The measurement system operates speed-controlled thus delivering speed-depending measurement data (cf. Figure 10b). To process data accordingly, actual volume flow values (cf. Figure 10a) are transformed into actual rotational speed values (cf. Figure 10c), which in turn are used to convert speed-depending into volume flow-depending data.

All data used in the following is generated and converted (in regard to temperature and volume flow) like this and serves as foundation for the subsequent researches.
### 4 Optimization Strategy and Results

#### 4.1 Energy – Flow Analysis

The introduced optimization strategy for variable speed pump drives focuses on the maximization of energy efficiency by minimizing the losses and by best possible adjustment of system and process.

The center of this strategy is the measurement and in-depth analysis of the energy flow of motor-pump-units from its electrical supply to the hydraulic process – from source to function (cf. Figure 11).

**Figure 11: Flow of energy in an electro-hydraulic pump drive**

The energy flow helps identifying the strengths and weaknesses of the components, analyzing interdependencies between the MPU’s components and the interaction with the process.

This analysis allows the direct comparison of pump drives, enables the search for preferred combinations and, together with detailed modelling of the process driven, also its optimization.

**Figure 12: Overview of the partial efficiencies of a variable speed pump drive with electrical motor**

Here, the hydraulic power available for the process always represents the reference point and is depicted in relation to its defining values, effective pressure and effective flow. Additional parameter with influence to the flow of energy, such as type of fluid or fluid temperature, are also considered – they are integrated in the efficiencies of the components (Figure 12).

#### 4.2 Optimization Approaches

##### 4.2.1 Efficient components

Starting point of each optimization is the selection of components as lossless as possible. By direct comparison, the efficiency of competing components can be analyzed and the best option can be chosen. Basis for the comparability is that the reference is fixed for all comparative figures and for all components and systems. Therefore, each operation point is set by the effective pressure and the effective flow, values that are defined by the process and not pump drive specific. Such a comparison for two different types of internal gear pumps is shown in Figure 14.

IGP-A is an internal gear pump with fixed sicle, IGP-B a gap compensated internal gear pump with divided sicle and pierced internal gear (principle by Eckerle). For the volumetric (a), hydro-mechanical (b) and overall pump efficiency (c), Figure 14 displays the differences between the two pump types. As expected, the volumetric efficiency clearly shows the influence of the gap compensation. Especially when there is only minimal flow and a high load is applied, the volumetric efficiency (cf. Figure 14a) of IGP-B is up to 23% better than the efficiency of IGP-A.

**Figure 14: Comparison of pump efficiencies for two different types of internal gear pumps (IGP-A: pump with fixed sicle; IGP-B: pump with compensation by divided sicle – principle by Eckerle) @ Toilc = 40°C**

For the hydro-mechanical efficiency (Figure 14b), this situation is completely inverted. Here, for the whole range of performance, IGP-A is more efficient than IGP-B due to more power losses that are caused by the increased internal friction that is produced by the gap compensation.

Finally, the overall pump efficiency (Figure 14c), the product of volumetric and hydro-mechanical efficiency, shows divided results. Pump IGP-B has, because of less internal leakage, the better efficiency for operating points with high loads and rather low flow. Because of the increased friction, for all the other operating points, IGP-B is less efficient than IGP-A.

In addition to their help with the search for the least lossy component, the comparisons provide the basis for detailed analysis of the energy loss mechanisms or the advantages and disadvantages of certain pump or motor types.

A similar comparison, this time for two external gear pumps with different functional principle, is shown in Figure 15. One pump has straight gearing (EGP-straight) and the other has helical gearing (EGP-helical). Again, all three characterizing efficiencies are displayed. Regarding the volumetric efficiency (Figure 15a), for the most part, the pump with helical gearing shows less leakage than the pump with straight gearing and therefore better efficiency. Nevertheless, it stands out that the helical geared design has, in the range of moderate pressures of
about 100 bar to 125 bar, increased leakage. Up toward maximum pressure, the leakage decreases again and the efficiency is again good, up to 10% better than with the straight gearing. Unlike with the internal gear pumps, the hydro-mechanical efficiency is not completely in favor for one specific pump type. At high flow and with minimal load, the helical gearing provides less friction and a better efficiency. For all other operating points, the efficiency of the pump with straight gearing is superior (Figure 15b).

Figure 15: Comparison of pump efficiencies for two different types of external gear pumps (EGP-Helical: pump with helical gearing; EGP-Straight: pump with straight gearing) @ T_{Fluid} = 40°C

Regarding the overall pump efficiency (Figure 15c), the EGP-straight shows up to 12% better values, especially for low flow. Only for high flow level and very low load, the helical system provides an up to 10% better efficiency.

Exceeding the analysis of available components, there is an interest in further optimizing these components especially regarding their use in applications with variable speed pump drives – which in the first place means reducing their losses. For this purpose, this analysis can also provide initial hints by what measures such an optimized construction can be realized.

4.2.2 Matching pump drive components

After the optimization of single components, the next step is to do the same thing for the whole pump drive unit. Therefore, also the electrical system and its losses are taken into consideration to enable the comparison of systems to each other, as it is shown in Figure 17. The energy flow oriented approach again allows the analysis of the strengths and weaknesses of specific combinations and helps to find the optimal system.

In Figure 17, the comparison of the two external gear pumps is revived (Figure 17a) and correspondingly supplemented. The influence of the electrical components, motor and servo amplifier, to the system efficiency is shown in Figure 17b.

As might be expected, this influence is only minimal. The electro-mechanical efficiency is depending on the rotational speed of the drive shaft and the torque applied. If both pumps were ideal machines without losses, there would be the same rotational speed and torque for both combinations for a specific operating point. This would lead to identical electro-mechanical efficiencies. Caused by the volumetric and hydro-mechanical losses of the pumps, the rotational speed and torque needed to provide flow and pressure are increased. This shift is defined by the pump efficiencies and therefore different for each pump. These, in most cases small, differences effect a slight shift of the motors operating point, thus also changed efficiencies as shown in Figure 17b.

In this case, the result is a slightly higher electro-mechanical efficiency for the system with the helical gearing. This indicates that the motor operating point, which is moved to a higher rotational speed and torque due to the less efficient pump (EGP-helical), is a more efficient one.

The overall efficiency of the system, consisting of pump, motor and servo amplifier, is shown in Figure 17c.

Figure 17: Comparison of efficiencies for two different types of pump drives (EGP-Helical: external gear pump with helical gearing; EGP-Straight: external gear pump with straight gearing)

Another, similar, comparison is shown in Figure 18. The system with the helical external gear pump is compared to a system with a gap compensated internal gear pump, both applied to the same electrical motor and amplifier. Again, both pumps and systems show significant differences in their efficiencies.

Figure 18: Comparison of efficiencies for two different types of pump drives (EGP-Helical: external gear pump with helical gearing; IGP-B: pump with compensation by divided sickle – principle by Eckerle)

The external gear pump with helical gearing has a better efficiency than the internal gear pump when high flow is requested but is less efficient when high loads are applied (Figure 18a). The influence of motor and amplifier, again, is only small (Figure 18b), there is an advantage in combination with the external gear pump of about 3%. Finally, the overall efficiency of the pump drive shows two clearly distinguishable ranges. The left corner of Figure 18c shows better efficiencies for the EGP-system, for all other operating points, the system with the internal gear pump is preferable.

Figure 19: System with highest efficiency – comparison of four combinations

Besides these comparisons of two systems, which aim at a detailed analysis, it is also possible and of interest to compare a whole bunch of systems with different pump or motor principles or from several manufacturers. Figure 19 shows the results of such a comparison for a pump unit that provides up to 24 l/min at a pressure of up to 250 bar.
The system is electrically driven by a servo-drive and four different gear pumps are applied. In Figure 19, for every point of operation, represented by pressure and flow, the system with the best efficiency is displayed.

As one can see, for each of the four combinations, there are operating points with best efficiency. The systems with internal gear pumps seem to prefer higher loads at only small flow whilst those with external gear pumps show better results especially at high flow. In addition to the information about the best system, it is also important to quantify the potential increase of the efficiency. This potential is also very dependent on the point of operation, as can be seen in Figure 20. It shows the differences between the overall efficiency of the best compared to the worst system, in absolute percentages. Especially for small power (little flow and low pressure) and in the range around the nominal power (maximum pressure and high flow), this potential increase in efficiency is only about a few percent.

This can be expected because all existing systems are optimized for use at their nominal power and therefore provide high efficiencies. Vice versa, because of physical laws, the percentages of the losses at low power are high which causes less efficiency. However, it also shows that with partial load, one can achieve up to 15% more overall efficiency by selection of the best possible combination.

4.2.3 Adjusted process layout

So far, all conducted measures targeted a general increase of the efficiency of a single component or pump drive. However, for the user or system manufacturer, the efficiency of a pump drive for a certain process is important. As the results of Figure 19 and Figure 20 show, this massively depends on the operating points and its shares to the process.

In order to take this into account, for the final step of optimization, the process is taken into consideration by specific load cycles. Therefore, the interaction of the pump drive with the driven process is included in the energy flow analysis and as a result, the energy consumption for this process can be calculated.

The load cycle, as displayed in Figure 22a), is specified in form of pressure and effective flow as information over the cycle time. The mechanical power needed to perform this hydraulic cycle is then calculated with the pump powers over the load cycle then leads to the energy, which the system consumes to perform on this specific process.

The exemplarily load cycle in Figure 22a) describes a positioning and clamping process. Via a cylinder, a work piece is moved to the position where it is clamped for a machining step. Then, the cylinder returns to its initial position and the process can be repeated.

For this process, Figure 22b) describes the energy balance, standardized to the hydraulic energy, which is defined to 100%. The necessary energy at the pump drive shaft (E_{ mech}) as well as the electrical energy consumption (E_{elec}) are shown. The differences between the four compared pump drives are significant – at best (Figure 22h, black), the system efficiency is 50%, so one has to provide two times the energy that can be used for the process. The worst system even consumes 220% of the hydraulic energy, which means only 45% efficiency (Figure 22b, yellow). Thus, the combination using the external gear pump with helical gearing, (Figure 22b, black), closely followed by the system using the EGP with straight gearing, are the most efficient options for this specific process.

Furthermore, comparing the mechanical and electrical energy absorption, it shows that in this specific application, the losses split up in equal parts to the pump and the electric drive.

An even higher efficiency than only by choosing the best suitable pump unit can be achieved by also adapting and optimizing the process itself. Such an adjustment is applied to the load cycle in Figure 23. This takes place by a variation of the positioning (0s to 15s in Figure 22a and Figure 23a).

Instead of a quick extension of the cylinder until it reaches the work part and then a slow approach to the clamping position, both parts are carried out with constant speed and therefore, also constant flow. The time needed to perform, as well as the hydraulic energy for one cycle, do not change due to this measure, but the system efficiency can massively be increased. As it is shown in Figure 23b, the electric energy consumption now is merely between 170% (best) and 182% (worst) of the energy available in the hydraulic circuit, meaning an increase in the efficiency up to 59% for the best combination. Furthermore, the best combination has now changed to the system using the EGP with straight gearing (Figure 23h, blue).
Summary, Conclusion and Outlook

This paper explains how energy analysis, based on detailed measurements, can be used to increase the energy efficiency of hydraulic variable speed drive systems. The optimization strategy therefore follows three approaches simultaneously – efficient components, matching pump drive components and adjusted process layout – each having a potential to enhance the energy efficiency. Depending on the optimization goal, the strategy is fitted accordingly, resulting in the usage of a single or a combination of the above stated approaches. However, the greatest possible impact is achieved by taking single components, the drive system as well as the overall process into consideration.

For the examples given in this paper, the overall system efficiency is increased by up to 10%.

Current studies are restricted to gear pumps but will be expanded and include piston pumps and variable displacement pumps in order to generate an overall market overview and to exploit the maximum gain of efficiency.

Volumetric modelling of gear pumps is used to investigate possible changes of the pump design beneficial to raising the component efficiency. Though the development effort has to be rated as high compared to the increase in efficiency that can possibly be achieved. Therefore, the focus of the current investigations is placed on system optimization by best possible adjustment of pump drive and process.

Apart from a mere optimization toward the best possible efficiency, topics like robustness of the system, its lifetime and controllability, etc. must be taken into consideration when constructing or refining drive systems as well as the whole process.

Finally, acquisition- and maintenance costs will also influence the calculation of the total cost of ownership. The question if such a system with an increased efficiency justifies potential additional costs has also to be taken into consideration.

Acknowledgements

The work in this paper is part of the project “EFplus – Effizienzsteigerung und Zuverlässigkeitsoptimierung energiesparender Motorpumpen-Aggregate”, funded by the BMBF (German Federal Ministry of Education and Research, FKZ:13FH0211X4). The permission for publication is gratefully acknowledged.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$I$</td>
<td>Current</td>
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</tr>
<tr>
<td>$M$</td>
<td>Torque</td>
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</tr>
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<td>[rpm]</td>
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<tr>
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Holistic Approach to the System Optimization of a Proportional Valve


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This contribution presents a holistic approach to the system optimization of a highly dynamic proportional valve. The model with lumped parameters which is used for the evaluation of the closed-loop performance is parameterized based on Finite-Element-Method (FEM) data. In addition to the calculation of static characteristic curves, a suitable excitation signal is applied to the transient FEM simulation. The valve dynamics of the current geometrical valve design are identified using the transient simulation results. This new approach enables a fully automated system optimization of a proportional valve. Hence, during the optimization, human expertise is not required.

Keywords: System Optimization, System Identification, Holistic Model, Highly Dynamic Proportional Valve
Target audience: Design Process, Components, Industrial Applications

1 Introduction

A proportional directional control valve represents a mechatronic system. Different domains like electrical engineering, mechanics, and hydraulics are merged into one mechatronic system. The result is a nonlinear system with fast dynamics which is difficult to control. An insight into the development process of mechatronic systems illustrates the sequential and mostly independent development of the individual subsystems. During this development, objectives are used which refer to the characteristics of the respective subsystem as maximizing force and torque, minimizing friction and movable mass. In the case of hydraulic valves, these subsystems are designed to reach the desired hydraulic characteristics of the whole system theoretically with consideration of safety aspects. Domain-specific modeling and simulation tools are utilized and are supported by human expertise. After a certain level of development is reached, the subsystems are assembled into a mechatronic system. From that moment, the overall system performance is essential. To evaluate the system performance a suitable control concept and power electronics are needed. Hence, the closed-loop performance highlights the system quality and proves whether the characteristics of the different subsystems fit each other. However, the characteristics of the several hardware subsystems which are suited best for closed-loop operations are not known precisely in advance. The controller is designed to reach the required system performance and to overcome the possible shortcomings of the hardware subsystems. The problem of the sequential development is for instance discussed in [1] and [2].

Due to the strong nonlinearities of some mechatronic systems, an advanced control concept is required. Regarding industrial applications, cascaded control concept based on nonlinear PID controller are widespread. The reasons for this are the well-studied PID theory and their low computational effort [3]. To reach pre-defined system performance requirements regarding the whole operating range, the controllers are extended with nonlinear characteristic curves for the integral and proportional amplification. Hence, the controller parameters depend on the control error \( e(t) \). The high number of coupled controller parameters requires a fully automated Hardware-in-the-Loop (HIL) optimization. In [4] a multi-criteria evolutionary optimization process for hydraulic valve controllers is presented and is suitable for at least 24 controller parameters. Although this optimization is time expensive, it is the only possibility to adjust the system performance after the mechatronic system hardware is build. Obviously, there is a need to develop a holistic system optimization procedure including optimization of design parameters like geometrical parameters as well as controller parameters. To optimize the hardware design time- and cost-efficiently a model-based approach must be used. The challenge is to develop a holistic simulation environment which can connect the different domains of a mechatronic system and enable controller optimization with an acceptable time effort. Usually, each subsystem is related to a domain-specific modelling tool and a numerical simulation method. Regarding the holistic model-based system optimization, two objectives contrast with each other. To improve the hardware design of a mechatronic system a model with a high level of detail including geometrical parameters is required. The clear impacts of design parameter variations on the physical interactions within a subsystem are essential. Numerical simulation methods to solve approximately partial differential equations like the Finite-Element-Method (FEM) or the Computational Fluid Dynamics (CFD) are well suited for this task. However, they come along with a high computational effort. On the other hand evaluating the closed-loop performance is of particular interest. Both the simulation quality and a low computational effort are needed to enable a controller optimization regarding different input signals. Models with lumped parameters are appropriated for this task. However, models with lumped parameters are not suitable for optimizing the exact geometric design. Due to the model simplifications, the number of physical interpretable parameters is low and many physical relationships between different parameters are omitted. For the coupling of different models and numerical solvers several approaches exist. In [5] an overview about the state-of-art is presented. Probably, the best-known approach is the so-called co-simulation. Either different simulation tools run in parallel and exchange their results at certain times or a superordinate simulation calls a subordinate simulation to perform some calculations at certain times and to transfer the results. The simulation with lumped parameters including the control concept mostly represents the superordinate simulation stage. More detailed models of the subsystems are on the basis of FEM or CFD. For instance, in [6], [7] and [8] detailed multibody simulations are subordinated to a simulation with lumped parameters to realize the closed-loop performance of machines and manipulators. In [9] a subordinated transient FEM is performed for electrical machines and converters. The motivations of these contributions also focus on the simulation-based analysis and design. However, whenever design parameters are modified, a new controller design has to be performed. If this is not the case, the robustness of the controller and the control concept is not fully ensured. A complex controller design as in [4] based on a co-simulation takes too much time and is not sustainable.

Another approach of combining different simulation methods is the parameterization of models with lumped parameters using simulation results of complex and detailed simulations like FEM. Regarding the simulation of electromagnetic actuators, the application of reluctance networks is well documented in the literature. This network method can transform the magnetic flux distribution within an actuator into an equivalent circuit with magnetic resistors. The structures of these reluctance models and the parameterization depend on the current flux distribution and the flux paths. Some examples are presented in [10], [11] and [12] and show a high simulation quality. However, the chosen structure and the parameterization is not fully generalizable to different geometric designs and requires some human expertise whenever the geometrical design changes. The need of human interventions is not applicable in a fully automated optimization process. Another approach is to use static characteristic curves of the different subsystems within a model with lumped parameters. [13] and [14] utilize this approach for the simulation of a holistic system performance of a pressure relief valve and an electro-hydraulic actuator. The static simulations of FEM models of the electromagnetic actuators and CFD models of the hydraulic subsystem provide the static relationship between different quantities like current \( i(t) \), force \( F(t) \), pressure \( p(t) \) and flow rate \( Q(t) \). Nevertheless, models, limited to static characteristic curves, are only applicable for the closed-loop simulation if the impact of dynamic interactions like eddy currents can be neglected. The introduced papers, using the approach...
of the parameterization of lumped parameters by complex simulations, already show the possibility of the holistic model-based system optimization for selected systems. The contribution of this paper is a new holistic system parameterization and optimization process of a fast-acting proportional valve. This new approach relies on the model with lumped parameters in /16/ and /17/ and section 2.1, which is used for the evaluation of the closed-loop performance. This modeling approach enables a high simulation quality for valves with fast dynamics through the application of characteristic curves combined with linear dynamics. It has been proven in /18/ that this model is suitable for a model-based controller design for a proportional directional control valve. This model with lumped parameters is not only parameterized with the help of characteristic curves resulting from static FEM simulations. Whenever the design changes, the linear dynamics are identified using signals generated by a transient FEM simulation. This new system parameterization process aims to realize a fully automated system optimization to exploit the hidden potential of the current valve design. Optimization of the hydraulic subsystem is not a part of this new approach, since hydraulic interactions are not considered during the controller design. The next section presents the selected proportional valve and the closed-loop system. The valve model and the controller are parameterized by the process in section 3. The section 4 shows the closed-loop performance which is used for the evaluation of the current valve parameter setup. Finally, section 5 summarizes the results and provides an outlook on further work.

2 Closed-Loop System

The results of this contribution refer to the hydraulic proportional directional control valve 4WRREH6 of the company Bosch Rexroth AG /15/. Figure 1 presents the cross-section of the 4WRREH. A proportional directional control valve is used to route an oil flow from the pressure port P to the working ports A or B. The supply pressure $p(t)$ and the spool stroke $x_1(t)$ adjust the flow rate $Q(t)$ through the valve. While a working port is opened a pressure relief takes place on the other side between the tank T and the further working port. The control task is the fast and precise positioning of the stroke $x_1(t)$. An internal inductive sensor provides the current stroke. A sensor signal of $x_1(t) = \pm 100$ mm indicates a fully opened working port A or B. To achieve high dynamics the electromagnetic actuator has a double stroke solenoid. The two coils operate in contrary motion directions. Motion is mainly characterized by the actuator and not by pre-loaded springs. Nevertheless, two preloaded springs contribute certain forces. These nearly equal pre-loaded springs ensure the force-type connection of the movable parts. To simulate the valve closed-loop performance a suitable holistic model is required. The holistic performance of this valve is evaluated using the closed-loop performance of the stroke respectively position control regarding different criteria like rise time, settling time and overshoot characteristics.

2.1 Valve Model with Lumped Parameters

In the previous work in /16/ and /17/ an electromagnetic actuator with a single coil which acts against a pre-loaded spring is investigated. The developed model is now ported and extended to the double stroke valve. Figure 2 illustrates the overall system model. It is divided into two current models, a force model and a mechanics model. The actuator input voltages are mapped to the electrical currents which are used to simulate the actuator force. This force leads to a movement of the spool. The linear dynamics $G_A(s)$, $G_B(s)$, and $G_F(s)$, exhibit purely mathematical parameters. These parameters have no physical interpretation. However, the analysis of the force model shows clearly the necessity of these dynamics. A sudden change in a current signal $i_1(t)$ or $i_2(t)$ at the input of the force curves $F_A(i_1, i_2, x_1)$ leads to a sudden change of the static force $F_{stat}(t)$. In reality, dynamic effects occur within the actuator as eddy currents. A magnetic flux linkage $\psi(t)$ and thus also the force $F(t)$ only reach the value pre-determined by the static characteristics after a certain delay. The linear force dynamics model this effect. It is a kind of dynamic correction of the static model approach. From a system-theoretical point of view, the combination of a static nonlinearity with linear dynamics is a common model approach and is called Hammerstein Model. Since the relationship between the current $i(t)$ and the flux linkage $\psi(t)$ is used in an inverted manner within the current models, the current dynamics exhibit a more derivative character. The amplitudes of the currents $i_{stat}(t)$ and $i_{stat}(t)$ at the outputs of the static curves $G_A(V_{PWM, A}, i_1(t), x_1)$ and $G_B(V_{PWM, B}, i_2(t), x_1)$ within the current models are too low. The linear dynamics increase the current amplitudes. Since a double stroke solenoid is used, two separated models are utilized. The basic approximation for every current model is the differential equation like it is described in /19/:

$$ V_{PWM, A}(t) = i_1(t) + i_2(t) \Rightarrow \frac{d i_1(t)}{dt} + R \frac{d i_2(t)}{dt} = \frac{\partial (\psi(t), i(t))}{\partial i(t)} dt $$(1)

This approximation is only valid for slow-acting actuators when dynamical effects like eddy currents can be neglected. The actuator input signals $V_{PWM, A}(t)$ and $V_{PWM, B}(t)$ are pulse-width modulated voltage signals. Since the coils are built identically, the assumption of identical behavior is valid. It applies $G_A(s) = G_B(s)$ and equal ohmic coil resistors $R_A = R_B$. FEM simulations show magnetic field coupling of the separated coils. Thus, the characteristic curves do not only depend on their own current $i_1(t)$ or $i_2(t)$ and the stroke $x_1(t)$. The further dependency on the current in the other coil is modeled as a simple cross coupling. The one dimensional movement of the movable parts as armature and spool constitutes a second order differential equation (mass-spring-damper) with a nonlinear friction term:

$$ F_d(v(t)) = F_d \frac{v(t) - \beta v(t)}{\psi(t)} $$

(2)

with the static friction $F_d$, an approximation parameter $\alpha$ and the damping coefficient $\beta$. The variable $v(t) = x_1(t)$ is the spool velocity. The spring constants $c_1$ and $c_2$ represent the two springs. A fast change of the armature position (step-shaped) also leads to dynamic effects within the actuator. These effects are neglected in this model since the impact is significantly lower compared to the impact of fast changing electrical signals. Usually, fast position changes only occur when hydraulic loads are applied. Hydraulic loads are not investigated in this work since the valve’s controller parameters are optimized with mechanically closed working ports A and B. Further, more detailed information, concerning the model with lumped parameters is presented in /16/ and /17/ and /18/.

2.2 Cascaded Control Concept

The structure of the cascaded valve native control concept is adopted. The inner simple current controllers are maintained. Figure 2 presents the native control structure. However, the complex outer position controller is replaced with a model predictive concept as it is described in /20/. Since the plant is known exactly during offline simulations, it can be used as a part of an advanced control concept. In this way, the number of classic controller parameters can be reduced significantly. In /20/ it has been proven that the sub-optimal model predictive trajectory set control (MPTSC) is real-time capable for the valve’s native sampling frequency of $f_s = 10$ kHz with a closed-loop performance similar to the nonlinear PID controller. The native current controller including the function of the power electronics become a part of the valve model. In every control sampling interval of the native controller of $T_s = 1/10$ kHz the voltage level can be inverted once by a hardware processing unit. The sampling frequency of this unit is above $f_s > 1$ GHz. For reasons of simplification, during the simulation of the valve model the native controller sampling interval is divided into eight equal time steps resulting in a simulation step size of $T_s = 1/80$ kHz for the plant. For the remaining sections, the controller design only focuses on the position controller. A state space controller as the MPTSC requires all $j$ states $x(t) = [x_1(t), x_2(t), ..., x_j(t)]^T$ of the plant. The native nonlinear PID controller only requires the stroke $x_1(t)$.
3 Parameterization Process of the Valve Model

The closed-loop system in section 2 is well suited to evaluate the valve performance or to be used in a model-based controller design. However, the physical relationship between the parameters is insufficient due to the model simplifications. For instance, when changing the coil windings the ohmic resistance and the magnetic characteristics change. In the model in section 2.1 there is no relationship between the characteristic curves and the ohmic resistors $R_A$ and $R_B$. When optimizing directly the parameters within the model with lumped parameters, the characteristic curves must be described by means of polynomial functions. The coefficients of the applied polynomial functions must then depend on the current $i(t)$ and the stroke $x_1(t)$. The additional resulting parameters can be subjected to an optimization. It would be determined how a characteristic curve should look like to provide an improvement in the closed-loop performance. However, the impact of a change of the shape of the characteristic curves on the ohmic resistor, on the movable mass or on the linear dynamics is not reproduced by the model. Furthermore, it is difficult to determine the parameter ranges of this additional polynomial coefficients. Characteristic curves which can be realized technically are not known in advance. Moreover, the mapping of characteristic curves to geometrical design parameters depends on human expertise. Due to its nonlinearity and its fast dynamics, the actuator represents the most significant challenge for the closed-loop control. The idea is to increase the level of detail of the electromagnetic actuator by including a FEM model. Within this model, the coupling of the parameters is more accurate in comparison to a model with lumped parameters. For the FEM simulation a 2D actuator model of the 4WRR EH6 is solved applying Ansys Maxwell. For simplification, a rotational symmetry around the $z$-axis (armature movement direction) is assumed. Figure 1 shows the axes of the 2D FEM model. A new parameterization process of the valve model in section 2.1 based on FEM data is presented in Figure 3. Model parameters which depend on the geometrical design are adjusted as soon as the design of the actuator changes.

3.1 Conversion of Geometrical Design Parameters to Physical Model Parameters

The valve model with lumped parameters is parameterized by geometrical design data and physical parameters. Firstly, some geometrical design changes of the actuator lead to new resistor and mass values. If the spool dimensions or the coil turns are changed, the resistor values $R_A$ and $R_B$ change too. The orthocyclic winding of the coil describes a relationship between the 2D spool dimensions $l_x \times l_z$, the number of coil turns $\lambda$ and the wire gauge including the isolation $d_w$:

$$\begin{align*}
h(x, \lambda, d_w) &= [1 + \left(\frac{\lambda}{l_x} - 1\right) \sin(60^\circ)] d_w. \tag{3}\end{align*}$$

The parameter $\xi$ is the number of wire layers and $h(x)$ the winding height in the $x$-direction. The wire gauge is chosen considering the DIN standard EN IEC 60317-0-1 which specifies the characteristics of copper winding wires. Using the specific resistance and the radius between the axis of rotation ($z$-axis) and each coil winding center the resistor values $R_A$ and $R_B$ can be calculated. When the armatures dimensions change during the optimization the mass of the armature, and thus the overall movable mass $m$, is also adjusted. Therefore, the 2D armature dimensions and the armature material mass density are utilized. The level of detail is not increased for the mechanical subsystem compared to [6/7/8]. Hence, there are further parameters which can be provided directly to the valve model. These include the spring constants $c_1$ and $c_2$ and the spring pre-load forces $F_{p1}$ and $F_{p2}$. The friction parameters of the function $F_f(v(t))$ are kept constant concerning the reference design since an adjustment is not possible with this low level of detail within the mechanics model.
### 3.2 Identification of Valve Dynamics

A change in the actuator’s geometrical design has an impact on the static behavior as well as on the dynamic interactions within the electromagnetic subsystem. The identification procedure of dynamics requires input and output signals. In [16], [17] and [18], measurable signals are investigated. The unmeasurable force during real operation of the valve is reconstructed by the measurable position signal and with the help of the inverse mechanics model. In contrast, within this work, signals simulated by the transient FEM are used for the identification process. Figure 4 presents an overview about the whole identification procedure. An electrical network is not modeled within the FEM. Thus, the electrical current constitutes the excitation signal within the transient FEM.

The induced voltage \( V_{A,FEM}(t) \), the dynamic flux linkage \( \phi_{A,FEM}(t) \) and the dynamic force \( F_{A,FEM}(t) \) represent the required and simulated FEM quantities. Regarding the model with lumped parameters, the current is the connecting signal between the current and force model. To identify the linear dynamic \( g_A(s) \) the input signal \( i_{stat,A}(t) \) and the output signal \( \phi_{stat,A}(t) \) are used. The variable \( g_A(s) \) is the excitation signal and \( i_{stat,A}(t) \) is unknown at first. It is the case that the input signal as well as the dynamics parameter are unknown. The left plot of Figure 5 shows the induced voltage \( V_{A,FEM}(t) \) in coil A as a result of the transient FEM using a pre-defined excitation signal. The left plot of Figure 6 shows the associated excitation signal (trapezoidal shape). The variable \( T \) represents a time constant. Figure 4a summarizes this first needed action to generate identification data. The current in coil B is kept constant at \( i_{B,FEM}(t) = 1 \text{ A} \) and the position is kept constant at \( x_{stat,B}(t) = 0 \text{ \%} \). The smooth magnetic flux linkage generated by the FEM cannot be reproduced by the model approach in section 2.1 since the input of the current model is a pulse-width modulated voltage signal. Hence, it makes no sense to use the FEM flux linkage signal \( \phi_{A,FEM}(t) \) at the input of the characteristic curve \( i_A(\phi_{A,FEM},i_{stat,B},x_1) \) to calculate the dynamics input signal \( i_{stat,A}(t) \). With the knowledge of the shape of pulse-width-modulated actuator input voltage \( V_{PWM,A}(t) \) and the function of the power electronics described in section 2.2, the smooth flux linkage signal can be approximated \( \phi_{FEM}(t) \) with \( \phi_{stat,A}(t) \). Figure 4b illustrates the approximation action. For the approximation of the FEM induced voltage \( V_{A,FEM}(t) \) by equation (1), which is also simulated with a fixed step size of \( t_s = 1/10 \text{ kHz} \), the pulse-width input signal \( V_{PWM,A}(t) \) is reconstructed. For every simulated FEM vector element, all 16 possible combinations of the input voltage signal are tested. As already mentioned, the model with lumped parameters including the power electronics is simulated with \( f_s = 80 \text{ kHz} \). Within a sampling interval of \( t_s = 1/10 \text{ kHz} \) of the outer controller, there are 16 possibilities of inverting the voltage level during a sampling interval step. One of 16 possible switching actions is illustrated in Figure 4b. If the first voltage value at the beginning of a sampling interval is fixed, then only eight possibilities exist. The switching sequence which leads to the minimal quadratic error between the flux linkage signal \( \phi_{FEM}(t) \) simulated by transient FEM and the approximated flux linkage signal \( \phi_{FEM,A}(t) \) by equation (1), regarding the current sampling interval, is chosen. Figure 5 illustrates the resulting approximation of the induced voltage \( V_{A,FEM}(t) \) (based on the reconstructed input voltage \( V_{PWM,A}(t) \)) and the magnetic flux linkage \( \phi_{FEM}(t) \). The time integration of the induced voltage leads to the magnetic flux linkage in the right plot of Figure 5. The comparison with the results of the transient FEM, highlights the high but limited quality of approximation. Only high amplitudes of induced voltages cannot be reproduced by the valve model with lumped parameters. This results in an increasing approximation error (for example at \( t = 4.8 T \)). A fast current change leads to a high amplitude of the induced voltage. Hence, a suitable current excitation signal is required for generating identification signals. For instance, a step-shaped excitation signal would lead to high amplitudes of the induced voltage. Thus, the trapezoidal shaped excitation signal with a variable steepness of the flanks constitutes a possible approach. For this reason, Figure 4b shows a feedback to Figure 4a. If a geometric design leads to highly dynamic electromagnetic interactions within the actuator, the slope of the flanks can be reduced automatically. The fit of the approximation of the FEM flux signal \( \phi_{FEM}(t) \) indicates a poorly conditioned excitation signal. Regarding the force model, no requirements exist for the current excitation signal.
To reduce the number of potential parameters is required. A linear dynamic model for a controller design process is essential. The excitation provides input signals which are well known. However, the trapezoidal signal is also used for time integration of the voltage signals leading to the magnetic flux linkage. Concerning different reference parameter modifications, the character of the current dynamics modifies the values of a characteristic curve of the whole plant is modified. Although the model is similar to the native behavior, the more dependencies it exhibits from more nearly related dynamics requires the use of a solver for nonlinear least-squares problems. A Newton-type solver is usually solved with a Newton-type solver for nonlinear optimization problems. For the identification of linear dynamics in the time domain, the dependency on the sampling step size is rather low. This, however, is the high number of controller parameters regarding the native nonlinear PID position controller. The global optimization algorithm may not achieve convergence within an acceptable time. To reduce the number of controller parameters significantly, a model predictive control concept is suitable. In this case, the whole plant is a part of the control concept and is known exactly during the optimization through the parameterization process in Figure 3. Figure 7 illustrates the closed-loop performance of the proportional valve with the reference design. The model with lumped parameters is parameterized applying the process in Figure 3 and the valve reference design. MPTSC with only three parameters is tuned to reach a closed-loop performance which is similar to the native nonlinear PID control concept. This three additional parameters can be included into an optimization vector. Thus, the assumption is that the extension of the system optimization vector with three controller parameters has only a low impact on the global optimization process. The current signals \( i_A(t) \) and \( i_B(t) \) exhibit valve common characteristics. The closed-loop performance regarding the position can be now used to evaluate different criteria as rise times, settling times and overshoots concerning different reference step amplitudes for the varying design. Whenever the design changes the prediction model is adopted inherently. The current signals can be used to define additional constraints like maximum power losses. A suitable optimization algorithm for multi-criteria optimization problems like the NSGA-II /22/ modifies the values of a pre-defined optimization vector. Elements of this optimization vector can either be geometrical, physical or controller parameters. In Figure 3 the optimization algorithm constitutes the outer loop. Although the model predictive approach reduces the number of controller parameters, there are still plenty of different geometrical parameters. To realize a useful system optimization, a pre-selection of the essential parameters is required. A sensitivity analysis contributes the needed information. Every construction parameter is changed by a factor of ±5% while the other parameters are kept constant. A static and a transient FEM simulation provide the sensitivity. The maximum changes in the characteristic curves and the pole-zero description of the linear dynamics are considered. Since the FEM is time-consuming, a global sensitivity analysis is not performed. An important consideration for the sensitivity analysis or the optimization is the adherence to restrictions concerning not recommended design. For instance, adherence to certain dimensions provided by human expertise, avoiding additional air gaps or overlap of different FEM elements should be taken into account. For this reason, the distantly a FEM element is related to an axis (x-axis or z-axis) the more dependencies it exhibits from more nearly related elements and thus geometrical parameters. Thus, the dimensions of the outer elements depend on the dimensions of the elements near the axes. The optimization process and the valve model are simulated with Matlab/Simulink.

4 Development of the Holistic Valve System Optimization Process

To evaluate the impact of the design variations on the system performance, a controller design process is essential. It makes sense to include the controller parameters into the set of optimization parameters. However, the problem...
5 Summary and Conclusion

This contribution presents a novel holistic approach to the system optimization process of a fast-acting proportional valve. A model with lumped parameters enables the simulation of the closed-loop performance and thus the evaluation of the holistic system performance. By utilizing a 2D FEM model of the electromagnetic actuator within the optimization loop, the level of detail of this nonlinear subsystem increases. A process is developed which maps geometrical design parameters to the required model parameters. For the simulation of dynamical effects like eddy currents of fast-acting electromagnetic actuators linear dynamics are identified based on transient FEM simulation results. The parameterization with FEM data results in a closed-loop system which shows a realistic closed-loop performance. This new parameterization process is fully automated and is suitable for the holistic valve system optimization without requiring human expertise. Further work is concerned with the description of the optimization setup in more detail. The progress of the evolution and the discussion of the results are of central interest.
\[ V_L(t) \text{ Induced voltage} \]  
\[ V_{LA}(t), V_{LB}(t) \text{ Approximation of induced voltage for coil A and B} \]  
\[ V_{LA,FEM}(t), V_{LB,FEM}(t) \text{ Induced voltage for coil A and coil B from transient FEM} \]  
\[ V_{PWM}(t) \text{ Pulse-width modulated voltage} \]  
\[ V_{PWM,LA}(t), V_{PWM,B}(t) \text{ Pulse-width modulated input voltage for coil A and B} \]  
\[ v_L(t) \text{ Ohmic voltage drop} \]  
\[ v_{LA}(t), v_{LB}(t) \text{ Ohmic voltage drop for coil A and coil B} \]  
\[ \mathbf{x}(t), \mathbf{x}_k \text{ State vector of the plant, } \mathbf{x} \text{ at time instance } k \]  
\[ x_{ref}(t) \text{ Reference state vector for control} \]  
\[ x_1(t) \text{ Stroke, position} \]  
\[ x_{ref}(t) \text{ Reference position for control} \]  
\[ \psi(t) \text{ Magnetic flux linkage} \]  
\[ \psi_{LA}(t), \psi_{LB}(t) \text{ Approximation of flux linkage for coil A and B} \]  
\[ \psi_{LA,FEM}(t), \psi_{LB,FEM}(t) \text{ Flux linkage for coil A and B from transient FEM} \]  
\[ \alpha, \beta \text{ Friction model parameters} \]  
\[ \lambda \text{ Number of coil turns} \]  
\[ \zeta(t) \text{ Number of winding layers in } x \text{ axis direction} \]  

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System optimization by means of an integrated design: the Dana case.

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The paper presents the Dana methodology to address the integrated design for winch systems. Beginning with the analysis of the generic expected performance of the system, the main issues and tasks are evaluated; moreover, the design workflow and the main benefits of integrated design are described with particular attention to the strong team working required to fulfill the defined target in the most efficient way.

Different sub-systems are analysed: the hydraulic motor-winch coupling, with particular attention to clocking speed and its relationship with motor non-uniformity grade and specific reducing gear-ratio to improve hydraulic-mechanic coupling, the hydraulic control system with the possibility to integrate several different functions in a compact and efficient solution, the winch torque sensor and motor angular sensor, which are specifically designed to merge with the components, provide fundamental information for the defined control strategy and also for safety assurance and the central control unit and its software providing an efficient-integrated control strategy and a user-oriented capability for personalization.

Keywords: Integrated design, hydraulic systems, efficiency, control strategy
Target audience: Mobile Hydraulics, Design Process

1 Introduction

Today, system optimization is one of the biggest challenges and one of the keys to improve overall performance of operating machines in every field of application. It is well known that an integrated design leads to an optimization of the whole system, in terms of efficiency, packaging and overall performance.

Team working, that leads to concurrent engineering, is one of the key-points of integrated-design success. In concurrent engineering /1/, an attempt is made to perform design and other related activities simultaneously rather than in series as in the case of traditional design. This may result in a reduction of the duration of the design project, cost savings, and better quality of the final design /2/.

As already said the first advantage of the team working in concurrent engineering is the significant reduction of the lead time for two main reasons: the common start and the easier and faster communications.

Common start means that the kick-off meeting of the project involves all the different areas (mechanical, electronic, hydraulic and software) together. In this way, all the different matters are analysed together from the very first moment and there is no need to arrange different meetings with the different suppliers. Moreover, with this kind of approach all the different designers can start working at the same time. This assure a reduction of the lead time without any real modification of the project.

Moreover, Dana has a team of experts, inside the same company, for each component of the system (mechanics, hydraulics, electronics, software) leading to a significant reduction in communication intervals and enabling the increasing of shared technical knowledge information leading to an easier and faster communication.

Furthermore, nowadays in every application a large amount of work is carried out by the electronic controller. In most cases, in fact, the firmware embedded on the controller must manage all the input and output resources needed both to control the system and to assure the safety /3/. Complexity management is the key to success for mobile machinery where the variety of customers and applications requires individual solutions /4/.

This paper deals with the optimization of a complete winch system by means of the integrated design of the different components. The development is supported with the calculation results of simulation tools in order to improve the design. The various sub-systems are tested and different solutions are compared as a result of both virtual validation and real tests; Dana has different available testing facilities that allow a two-stage test, first on subsystem and then on the complete system. Finally, real field solutions will be presented and analysed, benefits highlighted in terms of performance, while also referring to minor dimensions, simple maintenance or improved safety. By referencing all the collected data, it will be possible to define an optimized total system related to the integrated design methodology /5/.

To conclude, a future outlook will be presented including new integrated approaches including condition monitoring and enhanced connectivity addressing challenges to ensure even greater benefits to the integrated-design approach.

2 Case 1: Free-wheeling winch system with hydraulic unit incorporating safety function

The first example presented is a special application of a winch and its hydraulic circuit for Anchor Handling Tug Supply. Anchor Handling Tug Supply (AHTS) vessels function is to handle anchors for oil rigs, tow them to location and anchor them; AHTS vessels are fitted with winches for towing and anchor handling, having an open stern to allow the decking of anchors, and having more power than common Platform Supply Vessels to increase the bollard pull (Figure 1).

The winches are specifically designed for anchor handling operations and, according to safety standards, must have arrangements for quick anchor release; a proportional valve has been specifically developed with free falling system that in case of danger disengage the winch releasing immediately the anchor.

The hydraulic load sensing valve is designed to fulfil all required functions; this hydraulic unit has been developed incorporating the most advanced state of the art valve element concept, where it provides velvet smooth controllability, and it can ensure trouble free operation under demanding conditions.
More in details, the proportional valve designed to control the winches has been splitted in modular blocks which have the following characteristics:

- The basic hydraulic winch block contains the most commonly used functions. The counterbalance should have a cavity that allows a dummy (bypass) cartridge so one or both counterbalance valves can be replaced if function is not needed.
- The special functions block is separated in to 3 different blocks that fulfill the special/experimental functions in addition to the Basic block:
  - Hoisting winch, emergency lowering, mooring, constant tension
  - Anchor handling winch (variable pump)
  - Anchor handling winch (fixed pump)

The different parts of the system, in particular the winch and the hydraulic part, are developed in parallel by manufacturers who works side by side; this lead to a great advantage because the whole system has been thought out as a whole and thus all the performance have been optimized.

The other big advantage for the customer is that he receives the complete system and not two, or more, separate parts that must be assembled: if a customer has to fulfill a specific function or overcome a specific problem he can demand all the design to our engineers that will decompose it in two or more parts, develop each part and then recompose all the parts before giving the complete solution to the customer.

3 Case 2: High power to move the world (Bent axis motor for gearboxes & winches)

The advantage of axial piston units is the high nominal pressure level, variability of the displacements, beneficial efficiencies, through drive capability and low mass of inertia [6]. Typically, axial units are used in applications with high technical requirements, for example construction, agricultural, offshore and industrial machinery [7].

In the specific application reported in Figure 2 the hydraulic motor-winch coupling has been developed, with particular attention to clocking speed and its relationship with motor non-uniformity grade and dedicated reducing gear-ratio to improve hydraulic-mechanic coupling. Integrated in the winch there is also an overload transducer that, once known the geometry of the machine, is used to monitor and control the generated torque.

Figure 2: Mobile Crane equipped with Dana motor – winch system.

In this project hydraulic, mechanic and electronic engineers worked together on the entire system. This co-design has allowed to modify the characteristics of all the different components of the system and thus to include different features that have improved the efficiency and the usability of the machine.

More in detail, to achieve the regular rotation of the drum, with the minimum possible speed, the minimum displacement of the hydraulic motor was increased of 8%; with the new setting of the minimum motor displacement the speed at which the vibration of the drum starts (clocking) had a significant drop and the starting torque has been increased of 30%. Obviously, the operator of the crane noticed the correspondent reduction of winch speed, but it was evaluated as still acceptable.

The pressure setting of the motor control was set to 150 bar; this value corresponds to rise 2 ton with direct line pull at the minimum motor displacement.

The motor displacement is adjusted to $V_{g\ min}$ when the solenoid valve is switched on and if the operating pressure rises beyond the pressure setting (150 bar), the pressure limiting device overrides the electric two position control and the motor swivels out to the max motor displacement $V_{g\ max}$. This assures a completely safe lifting of the load.

The overload transducers and the position transducer, together with the electronic controller, mounted on the winch enable to know the torque transmitted on the drum – motor axis and an approximation of the load being lifted. Moreover, the speed sensor which is mounted on the motor, once defined on the electronic controller the system geometry, can provide information also on the winch control direction. When all these sensors are redundant and correctly connected with the electronic controller also the CAT 3 safety requirements are achieved (see Figure 3).

Figure 3: System architecture with redundant sensor for safety requirements.

Moreover, the case of the motor has been modified in order to directly connect the drain hoses between motor and valve; in this way there are no external hoses, the apparatus is more compact thus the installation and the maintenance of the whole system is easier.

The whole system has been tested on the testing tower (Figure 4), a specific experimental equipment where test engineers can reproduce different working conditions, even tougher than the real ones, and evaluate the complete system behavior in order to optimize it.

Figure 4: Testing tower and its technical features.

4 Case 3: Winch with integrated motor also for extreme environment

Integrated design is one of the key to guarantee the correct functioning of complex systems also in extreme environmental condition (i.e. very low temperature). After a first design phase, performance and quality are improved by a consistent concept and guidelines for testing and reliability. Standardized test procedures with improved effectiveness cover the complete range of functional, endurance and environment testing. Oil contaminations, aeration as well as performance at very low temperature are considered.
A challenging example is a winch with integrated motor that could work at -46°C (Figure 5). This system has been co-designed starting from the standard winch EP1524, to offer same external dimensions and layout than the standard product.

Thanks to the synergies and collaboration established between the Power Transmission team with the Hydraulic and Electronic team, it has been possible to determine the critical parts and to improve them. The possibility to work on each single component of the whole system allow us to validate, in a very short time, each component within our specialized test department, reducing the overall time needed for the project. In this way, the trial and error has been eliminated because the different problems have been solved by the single specialist.

For example, steel instead of cast iron has been chosen in order to improve the impact strength at low temperature and for an easy manufacturability a two-parts construction spindle has been adopted. Moreover, PTFE and PU oil seals specific for low temperature have been adopted. The case of the motor has been modified thus the system is ready to be connected with an oil heating flushing system.

Looking at graph of Figure 7 it can be seen that, in the worst condition (CW rotation), after 10s all the cold oil present inside the circuit is replaced by hot oil. This means that the volume of oil \( V \) necessary to warm up the system is:

\[
V = Q_p \cdot t = 2.75l
\]  

Where \( Q_p \) is the flow rate addressed by the external pump and \( t \) is the time.

Another test has been performed in order to determine the torque needed to make a complete rotation of the winch as a function of the temperature. From a standard motor – winch coupling the internal part of the motor has been removed and has been substituted by a special shaft which allow to measure the torque by means of a torque-wrench. The measurements have been performed at different temperature as reported in Figure 8.

\[
\text{Motor & winch -- } T = -46°C - \text{CCW rotation}
\]

\[
\text{Motor & winch -- } T = -46°C - \text{CW rotation}
\]
5 Case 4: Smart and scalable system control design

Manufacturers of mobile machinery require components that have been optimized to their specific application and to do that a large amount of work is carried out by the electronic controller. The firmware embedded on the controller, in fact, must manage all the input and output resources needed both to control the system and to assure the safety.

Brevini® Electronics Bricks is a firmware development tool studied for a fast development of the application; it is a user-friendly interface, based on NetBean IDE, that can be used to build a specific firmware application. It is based on an intuitive graphical approach thus no programming skills are required (see Figure 9).

Some specific libraries, called Apps, are already available, such as the ones for the area limitation, the load limitation for mobile cranes, the outrigger self-levelling, the solenoid valves PWM outputs management, the analogic and CANopen transducers management, while an infinity of customized libraries can be developed for each specific application.

The control strategy already developed and implemented are “plug and play” with the other Dana components (such as directional control valve, MAV board, …) thus the start-up time is further reduced.

With a traditional approach, to control a whole machine it is necessary to determine the architecture of the control, write the firmware with a mid-level programming language (i.e. C++ language), debug the software and upload it on the master controller of the machine. Using the Brevini® Electronics Bricks it is still necessary to determine the architecture of the control but there is no need to have a programmer that knows the mid-level programming language because the firmware can directly be built with the graphical interface and the single App doesn’t need to be debugged (because they are already debugged). The firmware has to be uploaded on the master controller. Consequently, using the Brevini® Electronics Bricks the system engineer could directly prepare the firmware to be uploaded on the master controller reducing the development time of about 25%.

6 Summary and Conclusion - Integrated design evaluation

The Dana approach to the system optimization by means of integrated design has been described. Dana, with its broad expertise in complementary fields of engineering – mechanics, electronics, hydraulics and software – has the capability to improve the performance of each single component of the complete system. With this kind of approach, it is possible to obtain an overall improvement of the system performance without the need to stress only a single part. High quality and performance are ensured by a team of experts supported with means of simulation and testing.

As an example, by means of the co-design of the winch and the motor for the mobile crane application, the minimum torque has been increased of 30% and the vibrations had a significant drop. Moreover, the improvements are not only in the overall performance of the system but also in the compactness and in the simpler maintenance of the system: the different parts have been engineered in order to be directly connected between themselves, without external hoses. This lead to a reduction of both the installation time and of the possible future maintenance and in a more compact system.

Finally, the possibility to use a customized control design made with Brevini® Electronics Bricks has been shown. With this new approach the development time could be further reduced of about 25%.

In the next months the Dana integrated approach will be expanded to include condition monitoring and enhanced connectivity. In this way it will be possible to ensure even greater benefits to the integrated-design approach.
Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$Q_p$</td>
<td>Pump flow rate</td>
<td>[l/min]</td>
</tr>
<tr>
<td>$t$</td>
<td>time</td>
<td>[s]</td>
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<tr>
<td>$T$</td>
<td>Temperature</td>
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<tr>
<td>$V$</td>
<td>Volume</td>
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Pressure Loss in Unsteady Annular Channel Flow

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The paper presents a methodology for calculating the pressure loss in unsteady flows through concentric annular channels. The momentum equation in axial direction is solved in the Laplace domain to obtain the unsteady radial velocity distribution. Based on the velocity profile, the relation between pressure loss and area-averaged flow velocity is derived. A time domain representation of this equation is provided for harmonically oscillating flows. For arbitrary temporal distributions of the flow, the inverse Laplace transform of the exact weighting function for each possible radius ratio is cumbersome, the annular channel flow is approximated by a plane channel. An error analysis shows that this approximation introduces errors less than 1% for channel geometries down to radius ratios of 0.45. The approximated weighting function is transformed into the time domain by using the residue theorem from complex analysis. The resulting convolution integral can be used in one-dimensional hydraulic system simulation software.

**Keywords:** concentric annular channel, frequency-dependent friction, unsteady flow, hydraulic simulation
**Target audience:** Design Process

1 Introduction

An annular channel is formed if a cylinder (radius \( r_2 \)) is mounted in a pipe (inner radius \( r_1 \)), see Figure 1. The present paper is limited to concentric annular channels, i.e. the axis of the pipe and the axis of the cylinder always coincide. Annular channels appear in various engineering applications ranging from tube heat exchangers to spool valve clearances. A key parameter to characterize annular channels is the ratio \( \phi = r_1/r_2 \) of the cylinder radius and the pipe’s inner radius. Since in most hydraulic engineering applications the gap height \( h = r_2 - r_1 \) is very small (e.g. few micrometres in sealing gaps), the radius ratio is typically close to unity.

![Figure 1: Geometry of a concentric annular channel.](image)

A typical engineering problem with respect to annular channels is the calculation of the pressure loss \( \Delta p \) for a given flow rate \( Q \) or area-averaged velocity \( \bar{u} = Q/A \). For laminar steady flow (\( \partial \bar{u}/\partial t = 0 \)), an analytical expression for the pressure loss is known, see e.g. IDELCHIK [1]. For unsteady laminar flow with a given temporal distribution of the flow rate, a reasonable first guess would be to take the instantaneous value \( \vec{u}(t) \) and calculate the unsteady pressure loss based on this value. This method is referred to as the quasi steady approach and is a common practice not only for annular channels but for pipe flows in general. The quasi steady approach gives exact results for unsteady flows with relatively low frequencies. For highly dynamic flows like water hammer problems, the experiments conducted by HOLMBOE and ROULEAU [2] (performed with cylindrical pipes without an inner cylinder) could demonstrate that the quasi steady approach fails to predict the correct shape of the pressure transients, see Figure 2:

![Figure 2: Pressure transients for a typical water hammer experiment [3].](image)

As can be seen in the diagram, the quasi-steady approach significantly underestimates the unsteady pressure loss (and hence amplitude damping) during a water hammer event. In such cases of highly dynamic flow, the effect of so-called frequency-dependent friction has to be taken into account. If frequency-dependent friction is used in the simulation, the correct shape of the pressure transients is matched very well. For laminar flows through circular pipes without an inner cylinder, a universal method for taking frequency-dependent friction into account has been published by ZIELKE [3]. The respective solution for annular channels is derived in the subsequent sections.

2 Exact solution

The pressure loss in any channel flow depends on the velocity distribution over the cross section of the respective channel. Hence, in order to calculate the pressure loss in annular channels, the velocity distribution has to be obtained.

2.1 Radial velocity distribution

Assuming nearly incompressible flow through annular channels of constant cross-section, the resulting velocity field is fully described by the radial distribution of the axial component \( u_z(r,t) \) of the flow velocity. For simplicity, this quantity will hereafter be addressed as \( u(r,t) \) since there is no other relevant velocity component. The theoretical derivation of the radial velocity distribution is based on solving the momentum equation (NAVIER-STOKES equation) in the direction of the pipe axis. Taking the aforementioned assumptions into account, the momentum equation reduces to the following differential equation:
\[
\frac{\partial u}{\partial z} + \frac{1}{\rho} \frac{\partial p}{\partial z} = \nu \left( \frac{\partial^2 u}{\partial r^2} + \frac{u}{r} \right) \tag{1}
\]

By performing a Laplace transform, the partial time derivative turns into an algebraic expression incorporating the Laplace variable \( s \), which simplifies the equation further:

\[
su'' + \frac{1}{\rho s} \frac{su'}{s} - s \left( \frac{\partial u}{\partial r} \right) = 0 \tag{2}
\]

Taking into account that the pressure does not depend on the radial coordinate, the equation above represents an ordinary differential equation. By introducing the non-dimensional radial coordinate \( R = r \sqrt{s/\nu} \), this ODE can be transformed into a modified Bessel differential equation of zeroth order. The general solution reads:

\[ u'(R) = c_1 J_0(R) + c_2 K_0(R) - \frac{1}{s \rho} \frac{\partial p}{\partial z} \tag{3} \]

Here, \( J_0 \) and \( K_0 \) denote the modified Bessel functions of zeroth order. The constants \( c_1 \) and \( c_2 \) are chosen such that the no-slip condition is satisfied at the surfaces of the pipe and the cylinder:

\[ u'(R_1) = u'(R_0) = 0 \tag{4} \]

Applying these boundary conditions, the solution reads:

\[ u'(R) = \frac{1}{s \rho} \frac{\partial p}{\partial z} \left\{ \ln (\frac{s}{\nu}) \left[ J_0(K_0(R_0) - K_0(R_1)) - J_0(K_0(R_0) - K_0(R_1)) \right] - 1 \right\} \tag{5} \]

To simplify the representation of this expression in the subsequent sections, the following abbreviations are introduced:

\[ \begin{align*}
J_0 &= J_0(R_0) - J_0(R_1) = J_0(R_0) - J_0(qR_0) \tag{6} \\
K_0 &= K_0(R_0) - K_0(R_1) = K_0(R_0) - K_0(qR_0) \\
N_0 &= N_0(R_0) - N_0(R_1) = N_0(R_0) - N_0(qR_0)
\end{align*} \]

Using these abbreviations, the velocity profile can be expressed as:

\[ u'(R) = \frac{1}{s \rho} \frac{\partial p}{\partial z} \left\{ \ln (\frac{s}{\nu}) \left[ J_0(R_0) - J_0(R_1) \right] - 1 \right\} \tag{7} \]

### 2.1.1 Steady flow

For the limit of steady flow \( (s \to 0) \), the velocity profile approaches the following expression:

\[
\lim_{s \to 0} u'(R) = u(r) = \frac{c_2}{4s} \frac{\partial p}{\partial z} \left[ \frac{r^2}{c_3^2} + \ln \left( \frac{r^2}{c_3^2} \right) (1 - q^2) \right] - 1 \tag{8} \]

To generalize the representation of the velocity profile, it is expressed using a non-dimensional radial coordinate \( r' \). This coordinate is defined in such a way that it assumes the values \( r' = 0 \) at the cylinder’s outer surface and \( r' = 1 \) at the pipe wall:

\[ r' = \frac{r - R_0}{R_1 - R_0} \tag{9} \]

For the graphical representation of the radial velocity distribution, the flow velocity \( u(r) \) is normalized with its maximum value \( u_{\text{max}} \):

\[ u' = \frac{u(r)}{u_{\text{max}}} \tag{10} \]

The velocity maximum \( u'_{\text{max}} = 1 \) is located at the coordinate \( r'_{\text{max}} \):

\[ r'_{\text{max}} = \frac{1}{1 - q} \tag{11} \]

The normalized flow velocity \( u' \) for different radius ratios \( q \) is plotted against the non-dimensional radial coordinate \( r' \) in Figure 3:

![Figure 3: Non-dimensional radial velocity profile for steady laminar flow through annular ducts.](image)

With vanishing gap height \( (q \to 1) \), the velocity profile approaches a parabolic shape as it is known from the plane channel POISEUILLE flow between two parallel flat plates of infinite width. The velocity profiles for radius ratios \( 0.5 < q < 1 \) are not plotted separately since they virtually coincide with the velocity distribution of the plane channel. Accordingly, the velocity maximum moves towards the gap centre for \( q \to 1 \):

\[ \lim_{q \to 1} r'_{\text{max}} = \frac{1}{2} \tag{12} \]

### 2.1.2 Harmonically oscillating flow

The volume flow \( Q \) and the area-averaged velocity \( \bar{u}' \) are obtained by integrating the velocity profile over the flow area \( A = \pi(r_2^2 - r_1^2) \):

\[ \bar{u}' = \frac{Q}{A} = \frac{1}{A} \int u' \, dA = \frac{1}{\pi r_2^2 (1 - q^2)} \int_{r_1}^{r_2} u' r' \, dr' = \frac{1}{s \rho} \frac{\partial p}{\partial z} \left[ 2(c_3 K_0 + 3 J_0) \right] \tag{13} \]

Here, the following abbreviations were used:

\[ \begin{align*}
J_1 &= J_1(R_0) - J_1(qR_0) \\
K_1 &= K_1(R_0) - K_1(qR_0)
\end{align*} \tag{14} \]

Combining equations 7 and 13, one obtains the radial velocity distribution as a function of the mean flow rate:
\[ u'(R) = \frac{L_0(R)X_0 - K_0(R)Y_0 - N_0}{2L_0(R)X_0 + J_0(R)Y_0} \]  

(15)

The velocity profile for unsteady flow depends on the Laplace variable \( s \). For the analysis of harmonically oscillating flows, \( s \) can be replaced by \( i\omega \). Instead of the angular frequency \( \omega \), the non-dimensional gap WOMERSLEY number \( W_0 \) is used. Unlike the usual definition of the WOMERSLEY number \( W_0 \), the gap WOMERSLEY number is calculated by replacing the diameter with the hydraulic diameter \( d_h \) of the annular channel (\( d_0 = 2b \)).

\[ W_0 = \frac{d_h}{2} \sqrt{\frac{\rho}{\mu}} = h \sqrt{\frac{\rho}{\mu}} = (r_c - r_i) \sqrt{\frac{\rho}{\mu}} \]  

(16)

The non-dimensional radial velocity profile \( u' = u'(R)/u_{\text{max}} \) for harmonically oscillating annular channel flows is plotted for gap WOMERSLEY numbers \( 1 \leq W_0 \leq 100 \) in Figure 4:

As can be seen, the differences between the velocity distributions of the annular channel with a radius ratio of \( q = 0.1 \) and the plane channel \( q = 1 \), dotted lines) are large for small gap WOMERSLEY numbers. Both velocity profiles practically coincide with their steady flow counterparts from Figure 3. With increasing \( W_0 \), the deviation between the velocity distributions becomes smaller. For large gap WOMERSLEY numbers \( W_0 = 100 \), the velocity profiles of the plane and annular channel \( q = 0.1 \) are virtually indistinguishable. Hence, even channels with quite small radius ratios behave like a plane channel at sufficiently high frequencies. It should be noted that the velocity maxima move towards the walls with increasing gap WOMERSLEY number. This phenomenon is known as the RICHARDSON annular effect from pipes without an inner cylinder [4].

2.2 Pressure loss as a function of area-averaged velocity

The pressure loss per unit length is the result of the shear stresses \( \tau'_r \) and \( \tau'_z \) acting at the cylinder and pipe walls:

\[ \pi(r_c^2 - r_i^2) \frac{dp}{dx} = \pi(\tau'_r + \tau'_z) \]  

(17)

The shear stresses are proportional to the gradient of the velocity profile at the wall:

\[ \tau'_r = \eta \frac{du}{dr} \bigg|_{r=r_c} \]  

(18)

\[ \tau'_z = -\eta \frac{du}{dr} \bigg|_{r=r_i} \]  

Combining equations 13, 17 and 18, one obtains the following relation between the unsteady pressure loss per unit length and the area-averaged velocity:

\[ \frac{\Delta p}{\Delta x} = \frac{\eta}{R^2} \frac{R_i^2}{(1 - q^2)R_i^2} \frac{u'_z}{\mu} = \frac{\eta}{R^2} F' u'_z \]  

(19)

Here, \( F'(s) \) denotes a non-dimensional function in which all frequency-dependent characteristics of the pressure loss are concentrated.

2.2.1 Steady flow

For the limit of steady flow, equation 19 converges towards the following expression:

\[ \lim_{\Delta x \to 0} \frac{\Delta p}{\Delta x} = \frac{\eta}{R^2} u_{\text{ave}} \]  

(20)

For vanishing relative gap height \( q \to 1 \), the pressure loss law equals the one of a plane channel of the same gap height \( h \):

\[ \lim_{\Delta x \to 0} \frac{\Delta p}{\Delta x} = \frac{\eta}{R^2} \frac{h(1 - q^2)}{1 + q^2 + \frac{(1 - q^2)}{h} u_{\text{ave}}} = \frac{12n}{h} \]  

(22)

2.2.2 Harmonically oscillating flow

If \( u(t) \) is assumed to be harmonically oscillating (e.g., like the flow provided by a reciprocating pump), equation 19 can be easily transformed into the time domain. Assuming a temporal variation of the mean flow velocity of the form \( \bar{u}(t) = \bar{u}_0 \sin(\omega t) \), the pressure loss per unit length is given by the following equation:

\[ \frac{\Delta p(t)}{\Delta x} = \bar{u}_0 \frac{\Delta u}{\Delta x} \sin(\omega t + \phi) = \bar{u}_0 \frac{\Delta u}{\Delta x} \sin(\omega t + \phi) \]  

(23)

Here, \( \Delta p/\Delta x \) refers to the pressure loss per unit length calculated by inserting \( \bar{u}_0 \) into equation 20 (quasi-steady approach). The magnification factor \( V' \) and the phase angle \( \phi \) are plotted against the gap WOMERSLEY number \( W_0 \) in Figures 5 and 6.
The integral can be solved by using the residue theorem from complex analysis. The residue theorem states that the contour integral along a closed curve equals the sum of the residues of the integrand:

\[ \frac{1}{2\pi i} \int_{c-i\infty}^{c+i\infty} e^{sx}W'(s) \, ds = \sum_{j} \text{Res}(e^{sx}W'(s))_{s=s_j} \]  

If the integrand of the contour integral can be represented as a quotient of two functions \( X(s) \) and \( Y(s) \), the residue at a simple pole \( s = s_j \) is given by:

\[ \text{Res} \left( \frac{X(s)}{Y(s)} \right)_{s=s_j} = \frac{X(s_j)}{Y'(s_j)} \]  

Hence, in order to evaluate the integral, the derivative of the numerator of \( W'(s) \) has to be evaluated at the pole \( s_j \). Analysis of the weighting function shows that the first (and trivial) singularity is located at \( s_j = 0 \). All other poles are on the negative real axis. Since the weighting function is dependent on the ratio \( q \), the positions of the poles vary with this parameter, too. It can be shown that for the technically important limit \( q \to 1 \) (sealing gaps), the position of the first nontrivial pole tends to \( s_1 \to -\infty \). This behaviour represents a serious hindrance for the numerical evaluation of the weighting function. Therefore, the exact solution for the annular channel is replaced by the plane channel approximation.

### 3 Plane channel approximation

The discussion of the steady flow velocity profile in section 2.1.1 revealed that the velocity profile of annular channel flows converges to a parabolic shape if the radius ratio \( q \) approaches unity. For the limiting case \( q \to 1 \), the relation between pressure loss and area-averaged flow converges to the expression for plane channels as well. Hence, the annular channel with a radius ratio close to unity can be thought of as a perturbed variant of the plane channel. Based on this interpretation, the plane channel approximation is developed.

#### 3.1 Velocity profile

Compared to the exact differential equation, the momentum equation of the plane channel approximation lacks a viscous term on the right hand side which represents the effects of the curvature of the velocity profile. Since a Cartesian coordinate system is used for the plane channel approximation, the radial coordinate \( r \) is replaced with the vertical coordinate \( y \):

\[ su + \frac{1}{\rho} \frac{dp'}{dz} = \frac{d^2u'}{dx^2} \]  

The general solution of this differential equation reads:

\[ u'(Y) = c_1 \sinh Y + c_2 \cosh Y - \frac{1}{\rho} \frac{dp'}{dz} \]  

In the general solution, the non-dimensional \( y \)-coordinate was used:

\[ Y = y \frac{\rho}{\rho_0} \]  

Application of the no-slip condition at \( y = h/2 \) and \( y = -h/2 \) leads to the velocity profile:

\[ u'(Y) = -\frac{1}{\rho} \frac{dp'}{dz} \left[ \text{sech} \left( \frac{H}{2} Y \right) \right] \cosh Y - 1 \]  

Here, \( H \) denotes the non-dimensional gap height \( H = h/\sqrt{\nu} \). For the limit of steady flow, the parabolic velocity profile is obtained as expected.
\[ \lim_{y \to 0} v'(y) = \frac{1}{2} \frac{\partial p}{\partial z} \left( y^2 - \frac{h^2}{4} \right) \]  

(34)

3.2 Pressure loss as a function of the area-averaged velocity

Based on the velocity profile, the pressure loss can be given as a function of the area-averaged velocity \( \bar{u} \). The pressure loss for plane channels depends on the inner and outer shear stresses as follows:

\[ \Delta p' = \frac{\tau_i' + \tau_o'}{h} \]

(35)

The shear stresses are given by:

\[ \tau_i' = \eta \frac{\partial u'}{\partial y} {y'}^2 \]

(36)

\[ \tau_o' = \eta \frac{\partial u'}{\partial y} {y'}^2 \]

Inserting these definitions into equation 35, we obtain the following relation between pressure loss per unit length and the pressure gradient:

\[ \Delta p' = -\frac{2}{h} \tanh \left( \frac{H}{2} \right) \frac{\partial p'}{\partial x} \]

(37)

The area-averaged velocity is given by:

\[ \bar{u} = \frac{1}{A_k} \int u' \, dA = \frac{1}{h} \int \left( 2 \bar{u} - \int \frac{1}{sp} \frac{\partial p'}{\partial z} \tanh \left( \frac{H}{2} \right) - 1 \right) dy \]

(38)

Combining the two equations above gives the approximated relation between pressure loss and average velocity:

\[ \Delta p' = -\frac{h}{H} \frac{\partial p'}{\partial x} \bar{u} \]

(39)

3.2.1 Steady flow

For the case of steady flow, the pressure loss is given by:

\[ \lim_{s \to 0} \Delta p' = -\frac{12}{h} \bar{u} \]

(40)

This result is consistent with equation 22 of the exact solution.

3.2.2 Arbitrary unsteady flows

For the plane channel approximation, a universal inverse Laplace transform of the weighting function can be provided. The approximated weighting function is given by:

\[ W'(s) = \frac{h^2}{2} \coth \left( \frac{h}{2} \sqrt{s} \right) \]

(41)

The distance between the roots of the poles approaches \( \sqrt{s_1} - \sqrt{s_2} = 2 \pi \sqrt{h^2} \). The poles are used to evaluate the residues. For the approximated weighting function, the residues are given by:

\[ \text{Res}_{s = s_j} \left( \frac{h^2}{2} \coth \left( \frac{h}{2} \sqrt{s} \right) \right) = \frac{8}{\pi} \frac{\tanh \left( \frac{h}{2} \sqrt{s} \right) - \text{csch}^2 \left( \frac{h}{2} \sqrt{s} \right)}{\sqrt{s} \sqrt{s}} \]

(42)

For the trivial pole \( s_0 = 0 \), the residue is given by:

\[ \text{Res}_{s = s_0} \left( \frac{h^2}{2} \coth \left( \frac{h}{2} \sqrt{s} \right) \right) = 8e^{-\pi^2 \bar{s}} \]

(43)

For all other poles \( s_j \), the residue equals:

\[ \text{Res}_{s = s_j} \left( \frac{h^2}{2} \coth \left( \frac{h}{2} \sqrt{s} \right) \right) = 8e^{-\pi^2 \bar{s}} \]

(44)

Hence, the approximated weighting function in the time domain is given by:

\[ W(t) = 12 + 8 \sum_{n=0}^{\infty} e^{-\pi^2 \bar{s} t} \]

(45)

Clearly, the weighting function can be decomposed into a constant part (quasi-steady approach) and a time-dependent part (frequency-dependent friction). The dynamic weighting function \( W_d(t) \) is obtained if the constant part is subtracted from the weighting function:

\[ W_d(t) = W(t) - 12 \]

(46)

Analysis of equation 45 shows that the sum of residues converges very slowly for small times. The behaviour for small times \( t \to 0 \) in the time domain corresponds to the behaviour for large values of the Laplace variable \( s \to \infty \) in the Laplace domain. Hence, the dynamic weighting function is developed into a power series at \( s \to \infty \) in order to obtain a more suitable time domain expression for small times. For large arguments \( s \), the hyperbolic cotangent tends faster towards unity than \( s \) tends towards infinity. Using this fact, the power series reads:

\[ \lim_{s \to \infty} W'(s) = \frac{h^2}{2} \coth \left( \frac{h}{2} \sqrt{s} \right) - \frac{1}{2} \left( \frac{h}{2} \sqrt{s} \right) - \frac{1}{4} \left( \frac{h}{2} \sqrt{s} \right)^2 - \frac{1}{8} \left( \frac{h}{2} \sqrt{s} \right)^3 - \cdots \]

(47)

The inverse Laplace transform of the dynamic weighting function is then given by:

\[ W_d(t) = \frac{2}{\sqrt{\pi} t} - \frac{8}{\sqrt{\pi} t^3} + 16 \frac{1}{3 \sqrt{\pi} t^3} + 16 t + \frac{128}{(3 \sqrt{\pi}) t^3} + 32 t^2 + \frac{1024}{15 \sqrt{\pi} t^3} - \cdots \]

(48)
Here, $t_n$ refers to the normalised time $t_n = t \nu /h^2$. For normalised times $t_n < 0.0023$, the above equation should be used instead of equation 45. The dynamic weighting function is plotted against the normalised time in Figure 7:

![Figure 7: Dynamic weighting function versus normalised time.](image)

Now that the dynamic weighting function is known, the overall pressure loss can be expressed as follows:

$$\frac{\Delta p(t)}{\Delta z} = \frac{12 \eta t}{h^2} \ddot{u}(t) + \int_0^t \frac{\ddot{u}(t)W(t-t)}{\Delta z} dt_1$$

With this equation, all required information to calculate the pressure loss for a given temporal distribution of $\ddot{u}(t)$ is provided. For practical calculations using a computer, the efficient approaches presented by TRIKHA [5] or SCHOHL [6] should be used.

### 3.3 Error Analysis

Due to the approximation by plane channel model, an error is introduced into the pressure loss calculation.

![Figure 8: Relative error of the plane channel approximation versus gap Womersley number.](image)

As can be seen in the diagram, the plane channel approximation introduces errors $\epsilon < 1\%$ if the radius ratio is above $\chi = 0.45$. This error can be decreased further if the quasi-steady part of the pressure loss is replaced by the exact expression (equation 20).

### 4 Summary and Conclusion

The findings of the paper can be summarized as follows:

- The pressure loss per unit length $\Delta p/\Delta z$ for steady laminar flow through an annular channel is given by:

$$\frac{\Delta p}{\Delta z} = \frac{\eta}{r_o^2} \frac{\dot{u}}{1 + \chi^2 + (1 - \chi^2) \dot{u}}$$

- The area-averaged flow velocity $\ddot{u}$ is given by:

$$\ddot{u} = \frac{Q}{A} = \frac{Q}{\pi r_o^2 (1 - \chi^2)}$$

- The pressure loss for steady flow through annular channels of vanishing gap height $h \to 0$ ($\chi \to 1$) equals the pressure loss of a plane channel with the same gap height:

$$\frac{\Delta p}{\Delta z} = \frac{12 \eta}{h} \ddot{u}$$

- For oscillating pipe flow with $\ddot{u}(t) = \ddot{u}_0 \sin(\omega t)$, the pressure loss is given by:
\[ \Delta p(x) = \frac{V}{h} \frac{\eta}{\nu} \frac{8 y}{1 + \varepsilon^2 + \left(1 - \frac{\nu}{k}\right) \ln \varphi} \]

- The magnification factor \( V \) and the phase angle \( \varphi \) are plotted as functions of the gap WOMERSLEY number in figures 5 and 6. The gap WOMERSLEY number is given by:

\[ W_\text{a} = \frac{\eta}{\nu} \left( \tau_2 - \tau_1 \right) \frac{m}{\nu} \]

- For small angular frequencies \( \omega < \nu^2/\nu \), the magnification factor \( V \) approaches unity and the phase angle \( \varphi \) tends to zero (quasi-steady limit).

- For arbitrary temporal distributions of the flow velocity, the pressure loss is given by a convolution integral. The convolution integral features a weighting function that depends on the radius ratio. Hence, a different inverse Laplace transform would have to be derived for each radius ratio. Instead, the annular channel flow is approximated by a plane channel model.

- The approximated pressure loss for arbitrary temporal distributions is given by:

\[ \Delta p(t) = \frac{12 \eta}{h^3} \bar{u}(t) + \int_0^t \frac{\partial \bar{u}}{\partial t}(t_1) W_d(t - t_1) \, dt_1 \]

- The error due to plane channel approximation is largest for steady flow. For radius ratios \( \varepsilon > 0.45 \), the error is less than 1 \%

- For an efficient evaluation of the convolution integral, the methods presented by TRIKHA or SCHOHL should be used [5] [6].

### Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A )</td>
<td>Flow area</td>
<td>m²</td>
</tr>
<tr>
<td>( \cosh(x) )</td>
<td>Hyperbolic cotangent of ( x )</td>
<td>-</td>
</tr>
<tr>
<td>( d_n )</td>
<td>Hydraulic diameter, ( d_n = 2h )</td>
<td>m</td>
</tr>
<tr>
<td>( F )</td>
<td>Function used in the determination of unsteady pressure loss</td>
<td>-</td>
</tr>
<tr>
<td>( h )</td>
<td>Gap height, ( h = r_o - r_i )</td>
<td>m</td>
</tr>
<tr>
<td>( H )</td>
<td>Non-dimensional gap height, ( H = h/\sqrt{\nu} )</td>
<td>-</td>
</tr>
<tr>
<td>( i )</td>
<td>( \sqrt{-1} )</td>
<td>-</td>
</tr>
<tr>
<td>( I_n(x) )</td>
<td>Modified Bessel function of the first kind and ( n )th order with the argument ( x )</td>
<td>-</td>
</tr>
<tr>
<td>( K_0(x) )</td>
<td>Modified Bessel function of the second kind and ( n )th order with the argument ( x )</td>
<td>-</td>
</tr>
<tr>
<td>( L )</td>
<td>Length of the annular channel</td>
<td>m</td>
</tr>
<tr>
<td>( p )</td>
<td>Pressure</td>
<td>kg·m⁻¹·s⁻²</td>
</tr>
<tr>
<td>( r )</td>
<td>Radial coordinate</td>
<td>m</td>
</tr>
<tr>
<td>( r' )</td>
<td>Non-dimensional radial coordinate</td>
<td>-</td>
</tr>
<tr>
<td>( s )</td>
<td>Laplace variable</td>
<td>s⁻¹</td>
</tr>
</tbody>
</table>

\( \varphi \) | Hyperbolic secant of \( x \) | - |
| \( t \)  | Time coordinate | s |
| \( t_a \) | Normalised time | - |
| \( u \)  | Axial component of the flow velocity | m·s⁻¹ |
| \( \bar{u} \) | Area-averaged flow velocity | m·s⁻¹ |
| \( u' \) | Non-dimensional flow velocity profile | - |
| \( W \)  | Weighting function | - |
| \( W_d \) | Dynamic weighting function | - |
| \( W_{o,h} \) | Gap WOMERSLEY number | - |
| \( y \)  | Cartesian coordinate for the plane channel model | m |
| \( Y \)  | Non-dimensional \( y \)-coordinate, \( Y = y\sqrt{\nu} \) | - |
| \( z \)  | Axial coordinate | m |
| \( \varepsilon \) | Error | % |
| \( \eta \) | Dynamic viscosity of the fluid | kg·m⁻¹·s⁻¹ |
| \( \nu \) | Kinematic viscosity of the fluid | m²·s⁻¹ |
| \( \varrho \) | Ratio of outer cylinder radius \( r_o \) and inner pipe radius \( r_i \) | - |
| \( \rho \)  | Fluid density | kg·m⁻³ |
| \( \tau \)  | Shear stress | kg·m⁻³·s⁻² |
| \( \omega \) | Angular frequency | s⁻¹ |
| \( x^* \) | Laplace-transform of the quantity \( x \), \( x^* = L(x)(s) \) | [s] |

### References


Piston Slippers for Robust Water Hydraulic Pumps

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Water hydraulics are used for applications which require an environmental safety standard for the fluid. In comparison to oil, lubrication with water is a challenging aspect because of the fluid’s lower viscosity. Wear and leakage in water lubricated contacts require lower pressure loads. In order to estimate the possible load carrying capacity in water hydraulics, the tribological contact between the piston slipper and swash plate in axial piston machine and respectively eccentric shaft in radial piston machines is investigated. For this purpose simulations based on the Reynolds-Equation are carried out and analysed.

**Keywords:** Water Hydraulics, radial piston pump, piston slipper, hydrostatic compensation, hydrodynamic load carrying capacity

**Target audience:** Water Hydraulics

1 Introduction

Being mostly replaced by oil-hydraulics, water hydraulics are nowadays still used, e.g., for press and mining applications and in food and pharmaceutic industries. These applications rely on the main advantages of using water as a pressure medium which are that water is not flammable and tap water as a source is non-toxic. In comparison to oil, water also has several disadvantages as being a poor lubricant. Regarding cavitation in hydraulic systems, water has a much higher vapor pressure. Therefore, cavitation more likely occurs, if for example the static pressure locally drops due to high velocities.

A current research project at IFAS focuses on raising the pressure level and therefore the power density of compact piston pumps for water hydraulics using tap water. State of the art water-lubricated axial piston pumps are only available on the market for a pressure level of maximum 16 MPa. Compared to a 35 MPa pressure level of oil hydraulic pumps, the remaining potential is relatively high (Figure 1). The present limitation is given due to wear and leakage in tribological contacts. In order to raise the pressure level, lubrication with water has to be investigated thoroughly. For the geometry of an axial piston machine a variety of research has already been done.

Manring et al. /3/ investigated the effect of linear deformations using an analytical solution of a hydrostatic slipper bearing. Calculations were done for concave and convex deformations. A convex deformation means that the gap height at the outer radius of the slipper collar is greater than at the inner radius. The deformation of a concave slipper is vice versa. A concave deformation leads to a higher, a convex deformation to a lower load carrying capacity. The leakage is increased in both cases. Within another research project Manring et al. /4/ measured the pressure profile of a hydrostatic slipper bearing. The test bench was equipped with pressure sensors on different radii of the slipper bearing. From the measurements Manring was able to calculate the load carrying capacity. The investigation focused on different ball socket geometries within the piston slipper.

Kazama /5/ developed a simulation model for the slipper/swash plate contact for mixed lubrication. He used the Average Flow Model of Patir and Cheng and the asperity-contacting model of Greenwood and Williamson in his simulation. Therefore this simulation model contains several side-effects that occur during mixed lubrication.

Kazama investigated the effect of an eccentric position of the piston pressure force which leads to greater inclination angles of the slipper and also higher contact pressure.

Liu et al. /6/ developed a test bench for experiments in search for materials for water hydraulic axial piston pumps. The test bench consisted of a turning swash plate (wobble plate) which actuated a piston and its slipper. Therefore the slipper/swash plate contact as well as the piston/cylinder contact could be tested. Liu et al. experimented with slippers made up of PEEK running against metal and ceramic coating. The wear value of the slipper is reduced to 38 % for the ceramic coating (stainless steel 32 µm, ZrO2 11.9 µm).

Rokala /7/ investigated the contact between slipper and swash plate for axial piston machines in order to verify the function of the tribological contact in a variable displacement water pump. For this purpose he developed a test bench containing a swash plate with a variable angle and a piston/slider assembly. Rokala measured the gap height of the slipper during operation to be about 7 to 10 µm. Furthermore a simulation of the deformation of the slipper was carried out and the resulting pressure profile was calculated. Rokala concludes that the development of a variable displacement water pump is possible.

The presented literature dealt with axial piston water hydraulic machines (APM). The research at IFAS focuses also on the principle of a radial piston machine (RPM). Radial piston pumps are typically used for high pressure applications, for example in common rail diesel pumps. For this research paper, piston machines with a displacement ranging from 23 cm³ to 46 cm³ are considered. The nominal speed is 1500 rpm and the maximum pressure is 21 MPa. The investigation and design of the tribological contact of the piston slipper, i.e., the simulation of the load carrying capacity, is discussed. The aim is to develop a sustainable tribological contact with a reasonable compromise between load carrying capacity and water leakage. The calculations base upon the viscosity of water and respectively HFA fluid. The use of tap water requires a more detailed focus on hydrodynamics because of the lower lubrication condition compared to HFA fluid.

![Figure 1: Pressure level for oil and water (left), principle of a radial piston machine (right)](image_url)
2 Lubrication with Water

Regarding lubrication with water, basically the same conditions as for oil lubrication are given. For the Reynolds-Equation used to describe a sliding wedge contact, the only condition referring to the fluid is that it needs to be Newtonian. Water is a Newtonian fluid. Therefore the same hydrodynamic theory as for oil lubrication can be applied. Due to the lower viscosity compared to oil, the gap height of the fluid film will be reduced in the case of water lubrication. In order to estimate the gap height, a sliding wedge geometry as shown in Figure 2 is used. Two different methods are compared, demonstrating the gap height’s change if the same hydrodynamic pressure built-up is assumed as for using a HLP 32 oil.

\[
\frac{h_{\text{water}}}{h_{\text{oil}}} = \sqrt{\frac{\mu_{\text{water}}}{\mu_{\text{oil}}}} = \sqrt{\frac{1}{30}} = 0.32
\]

The fluid film thickness \(h_{\text{water}}\) is reduced to 32% of the film thickness of the HLP 32 oil. A different method is to assume the same tilting angle for both lubrication models and to change only the gap height. The calculation leads to a quotient in Eq. 2 for the nominal gap height:

\[
\frac{h_{\text{water}}}{h_{\text{oil}}} = \sqrt{\frac{\mu_{\text{water}}}{\mu_{\text{oil}}}} = \sqrt{\frac{1}{30}} = 0.18
\]

The fluid film thickness \(h_{\text{water}}\) is reduced to 18% of the film thickness of the HLP 32 oil.

Furthermore the reduced fluid film height causes an increased interference with side effects compared to oil lubricated contacts, e.g., interaction of the surface roughness or elastic deformation due to high loads /1/. Therefore the hydrodynamic load carrying capacity should be considered regarding several aspects. The research focuses on the impact of geometry and gap height on the load carrying capacity.

3 Slipper Geometry and Calculation

The geometry of the two slipper types, axial piston machine (APM) and radial piston machine (RPM), are shown on Figure 3.

\[
q_c = \frac{F_{\text{cone}}}{F_{\text{Piston}}} = \frac{\mu r g \cdot (D^2 - D^2_{\text{Pocket}})}{\mu g \cdot (D^2_{\text{Slipper}} - D^2 - D^2_{\text{Pocket}})} = \frac{D^2_{\text{Slipper}} - D^2_{\text{Pocket}}}{D^2_{\text{Pocket}}} \cdot \frac{1}{\ln(D^2_{\text{Slipper}}/D^2_{\text{Pocket}})} \cdot 2\pi \cdot r \cdot r
\]

For this equation the piston slipper has a circular geometry and a logarithmic pressure drop for the sliding surface is assumed. This assumption bases upon a constant gap height, which causes the pressure drop due to leakage, and no sliding motion of the piston. But the remaining, not compensated load would actually close the gap. This means that the theoretical compensation does not apply to real contacts.

Furthermore if the distribution of the gap height is changed (e.g., during motion), the pressure distribution changes as well. Therefore a compensation ratio \(q_{\text{CInst}}\) is defined, including only the inner pocket of the slipper pad. This area is independent from any movement and gap height, representing a constant share of the hydrostatic compensation.

\[
q_{\text{CInst}} = \frac{F_{\text{Pocket}}}{F_{\text{Piston}}} = \frac{\mu r g \cdot (D^2 - D^2_{\text{Pocket}})}{\mu g \cdot D^2_{\text{Pocket}}} = \left(\frac{D^2_{\text{Pocket}}}{D^2_{\text{Slipper}}}\right)^2
\]

The comparison between inner \((q_{\text{CInst}} = 0.58)\) and total theoretical compensation \((q_c = 0.93)\) for the chosen geometry indicates that the load carrying capacity depends strongly on the pressure profile over the collar of the slipper. This means that a change occurring during tilting leads to a different load carrying capacity.

Table:

<table>
<thead>
<tr>
<th>Geometry</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(D_{\text{Piston}})</td>
<td>17 mm</td>
</tr>
<tr>
<td>(D_{\text{Pocket}})</td>
<td>13 mm</td>
</tr>
<tr>
<td>(D_{\text{Slipper}})</td>
<td>20 mm</td>
</tr>
<tr>
<td>(R_{\text{Cone}})</td>
<td>40 mm</td>
</tr>
<tr>
<td>(q_{\text{CInst}})</td>
<td>0.93</td>
</tr>
<tr>
<td>(q_{\text{CInst}})</td>
<td>0.58</td>
</tr>
</tbody>
</table>
Especially for water lubrication the pressure drop over the sliding surface should be calculated using the Reynolds-Equation, Eq. 5.

\[
\frac{\partial}{\partial r} \left( \frac{h^3}{12 \mu} \frac{\partial p}{\partial r} \right) - \frac{w_r}{2} \frac{\partial h}{\partial r} + \frac{u_r}{2} \frac{\partial h}{\partial r} = \frac{\partial h}{\partial t}
\]

For this research the dynamic change of the gap height is neglected (\(\frac{\partial h}{\partial t} = 0\)). Furthermore ideal smooth surfaces are considered and because no model for solid contact pressure is implemented, only fluid film pressure can be calculated. The Reynolds-Equation is solved based upon a cylindrical coordinate system which fits the geometrical appearance of the fluid film in the slipper contact.

In water hydraulics, plastics are often used as materials. This causes a relatively high deformation compared to metals which could lead to a higher load carrying capacity. In order to allow this increase, the theoretical compensation \(q_{C,th}\) is set to a value of about 93%. The impact of deformation is discussed in detail in section 5.

The described principles of the compensation and load carrying capacity are applied to the slipper contact of a radial piston pump. In this type of machine, the piston slipper is in contact with the eccentric shaft and therefore the surfaces are curved (Figure 4).

Due to the fact that the concave radius of the slipper \(R_{c,curv}\) is slightly larger than the convex radius of the shaft \(R_{c,conv}\), the gap height already differs geometrically. This effect is taken into account by calculating the difference of the gap height and using it for an unwrapped simulation. As shown in Figure 4, the assumed logarithmic pressure drop is increased (solid line vs. dashed line). The radial piston machine is investigated with three radius differences (\(\Delta R_{RPM} = R_{curv} - R_{Ecc} = 10, 20, \) and 40 \(\mu m\)). If the radius difference is set to zero, the results would be equal to the axial piston slipper.

For all types of slippers the same hydrostatic compensation as shown in Figure 3 is used in order to give a comparison of the load carrying capacity which is given as the effective force ratio \(q_{C,eff}\).

\[
q_{C,eff} = \frac{F_{max}}{P_{area}} = \frac{\int_{P_{area}} P \cdot dA}{P_{area} \cdot \frac{D_{piston}}{4}}
\]

4 Results of static Reynolds-Equation

4.1 Parallel Gap
As a reference, a static simulation without an inclination or motion of the slipper is carried out. In this case the results equal the analytical solution which is used for the expression of the theoretical hydrostatic compensation \(q_{C,th}\) shown before. The pressure profile depicts a logarithmic pressure drop along the slipper collar, Figure 5. For the simulation a nominal gap height of 0.5 \(\mu m\) is assumed.

The results show that the load carrying capacity given as the value of \(q_{C,eff}\) (shown in Table 1) of the radial piston slipper is lower than of the axial piston slipper. Furthermore an increased radius difference \(\Delta R_{RPM}\) leads to a reduction of the load carrying capacity and an increased leakage. As said before, the results compared to the analytical solution will differ, if the slipper is simulated with an inclination angle and constant motion. This is presented in the following sections.

4.2 Inclination Angle
Once the slipper is tilted against the normal vector of the corresponding surface, the pressure distribution is unbalanced and able to produce a tilting torque trying to balance the tilting position of the slipper. This means the fluid film bears a normal force as well as a tilting torque.

![Figure 4: Radial piston slipper (left), unwrapped for calculation (right)](image)

![Figure 5: Pressure distribution along the slipper collar (0.5 \(\mu m\) gap)](image)

![Table 1: Effective compensation and leakage](image)
The results show that the slippers of the radial piston machine require a greater inclination angle in order to bear the same torque as the axial piston machine’s slippers. This means that the ability for bearing tilting torques is higher for axial piston slippers. Eventually the radial piston slipper with a radial difference of $\Delta R_{RPM} = R_{Curve} - R_{Ecc} = 40 \, \mu m$ is not able to bear a torque load of 0.25 Nm because due to the large inclination angle the gap height is set to zero at its minimal value (grey shaded cells in Table 2). Therefore the fluid film cannot bear the load itself and part of the load is carried by solid metal contact.

4.3 Constant Motion

For the hydrodynamic simulation of the slipper, a constant linear velocity $v_{rel}$ of 1 m/s in direction of the x-axis is assumed. In case of the radial piston machine this value represents a rotational speed of about 500 rpm. At first the same inclination angle of $0.001^\circ$ as before and a nominal gap height of $0.5 \, \mu m$ are used. The plots of this simulation are shown in Figure 8. Compared to Figure 7 the additional pressure built up due to shear flow can be seen.
the shear flow, is increased. The radial piston slipper with $\Delta R_{\text{rpm}} = 40 \, \mu m$ is not able to carry the tilting torque, even during motion.

<table>
<thead>
<tr>
<th>Fixed Inclination Angle ($\beta = 0.001^\circ$)</th>
<th>Fixed Tilting Torque ($T_{\text{tilt}} = 0.15 , \text{Nm}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$q_{C, \text{eff}}$ (ml/min)</td>
<td>$T_{\text{tilt}}$ (Nm)</td>
</tr>
<tr>
<td>APM</td>
<td>0.942</td>
</tr>
<tr>
<td>RPM</td>
<td>0.933</td>
</tr>
<tr>
<td>RPM 10 $\mu m$</td>
<td>0.918</td>
</tr>
<tr>
<td>RPM 20 $\mu m$</td>
<td>0.898</td>
</tr>
</tbody>
</table>

Table 3: Tilting torque (1 $\mu m$ nominal gap height)

5 Effect of Deformation

The use of plastics as material for water hydraulic components leads to greater deformations compared to metal components used in oil hydraulics. Because the deformations are of the same magnitude as the actual gap height, the effect on the load carrying capacity cannot be neglected. For this approach, the hydrostatic pressure distribution for an axial piston machine is calculated using ideal geometries. Then the deformation due to the pressure distribution is calculated using Ansys FEM. Afterwards the result is used as input for the Reynolds–Equation. This is repeated iteratively until the change of the deformation between two iterations is less than 0.01 $\mu m$. The slipper consists of two components: a steel body and an implemented sliding disc made of PEEK. The sliding disc contains the geometry of the collar for the pressurised fluid film.

The deformation of the piston slipper is shown in Figure 9. The scale of the deformation is exaggerated by a factor of 550.

Figure 9: Deformation of the piston slipper (Ansys FEM)

The export of the gap height distribution to the simulation software and the resulting pressure distribution is shown in Figure 10.

Figure 10: Piston slipper deformation; gap height distribution (left) and pressure distribution (right)

The result of the pressure distribution is comparable to the results of Rokala /7/. The deformation of the slipper results in a load carrying capacity with factor of $q_{C, \text{eff}} = 1.067$, meaning that the slipper is over compensated and therefore will actually open the gap. The leakage value of 0.811 ml/min is increased by factor 5 compared to an undeformed slipper with a gap height of 0.6 $\mu m$. The effect of concave deformations which lead to a higher load carrying capacity and also a major increase in leakage fits well to the calculations of Manring /3/.

6 Summary and Conclusion

Due to the relatively low viscosity of tap water, the reduction of the gap height in water-lubricated tribological contacts defines the need for a robust design in water hydraulics. For this purpose the contact between piston slipper and swash plate in axial piston machines and respectively eccentric shaft in radial piston machines has been investigated. The research used a simulation of the hydrodynamic load carrying capacity calculated via the Reynolds–Equation for different operating conditions.

The results indicate that the gap height is in the range of 0.5 $\mu m$ in order to bear the normal load force and the tilting moment. The axial piston machine’s slipper has the greatest overall load carrying capacity. For the radial piston machine the results show that the slippers need to be manufactured with respect to an almost parallel clearance in the lubrication contact. Then the load carrying capacity is of a fair value compared to axial piston machines. The results have not yet been validated via measurements. But compared to oil hydraulics the gap height is reduced due to the lower viscosity of the fluid. The resulting gap height is within the magnitude of the surface roughness which indicates that solid contact and therefore mixed friction will occur in the contact.

Therefore plastic materials are used for water lubricated contacts. The investigation shows that the resulting deformation is of the same magnitude as the gap height itself which affects the fluid film pressure and the load carrying capacity. Due to the use of plastic materials the deformation has already a major impact at relatively low pressures compared to oil hydraulics. In an oil hydraulic piston machine, metal materials can be used for both surfaces and therefore critical deformations occur at much higher pressure levels. But for designing a water lubricated slipper contact, the deformation has to be considered.

Due to the small clearances necessary to ensure a fair load carrying capacity, the micro hydrodynamic effects should be taken into account. This entails an additional pressure built up due to the surface roughness on the one hand and a contact pressure built-up due to solid metal contact on the other. Another option is to use tailored surface textures which lead to a higher hydrodynamic load carrying capacity. Doing so, the already discussed poor lubrication with water needs to be modelled in detail and calculated thoroughly.
7 Acknowledgements

The research work leading to this publication was funded by the German Federal Ministry for Economic Affairs and Energy under the reference ZF4199603KO6 as a cooperation project between Institute of Fluid Power Drives and Systems (IFAS), RWTH Aachen University and Hauhinco Maschinenfabrik GmbH & Co.KG. The responsibility for the content of this paper lies with the authors. The authors are grateful for the funding.

References


Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>μ_{water}</td>
<td>Dynamic viscosity of water</td>
<td>(Pa ∙ s)</td>
</tr>
<tr>
<td>h_{nom}</td>
<td>Nominal gap height by lubrication with water</td>
<td>(mm)</td>
</tr>
<tr>
<td>D_{piston}</td>
<td>Diameter of piston</td>
<td>(mm)</td>
</tr>
<tr>
<td>D_{ocket}</td>
<td>Diameter of pocket in slipper contact, inner diameter</td>
<td>(mm)</td>
</tr>
<tr>
<td>D_{Slipper}</td>
<td>Outer diameter of slipper contact</td>
<td>(mm)</td>
</tr>
<tr>
<td>R_{cave}</td>
<td>Concave radius of the piston slipper</td>
<td>(mm)</td>
</tr>
<tr>
<td>R_{conv}</td>
<td>Convex radius of the eccentric shaft</td>
<td>(mm)</td>
</tr>
<tr>
<td>ΔR_{RPM}</td>
<td>Difference of slipper radius and eccentric shaft radius of an RPM</td>
<td>(μm)</td>
</tr>
<tr>
<td>q_{c,th}</td>
<td>Theoretical compensation of normal load force</td>
<td>(-)</td>
</tr>
<tr>
<td>q_{c,in}</td>
<td>Compensation ratio of the pocket</td>
<td>(-)</td>
</tr>
<tr>
<td>q_{c,e}</td>
<td>Effective compensation of normal load force</td>
<td>(-)</td>
</tr>
<tr>
<td>p_{HP}</td>
<td>Piston displacement chamber pressure</td>
<td>(bar)</td>
</tr>
<tr>
<td>p_{(r,ϕ)}</td>
<td>Pressure profile in slipper contact</td>
<td>(bar)</td>
</tr>
<tr>
<td>A_{eff}</td>
<td>Area in slipper contact for pressure profile</td>
<td>(mm²)</td>
</tr>
<tr>
<td>β</td>
<td>Inclination angle</td>
<td>(°)</td>
</tr>
<tr>
<td>T_{Fluid}</td>
<td>Tilting torque</td>
<td>(Nm)</td>
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Digital pumps using high speed on/off valves to control fluid entering and leaving the piston cylinder displacement chamber can increase efficiency by eliminating the leakage and friction associated with the port plate. Leakage scales with the displacement because the displacement chamber is only pressurized during a portion of the piston stroke. This work investigates the modeling, prototyping, and testing of two prototype digital pumps. The first prototype actuated on/off valves using electrical solenoids; the second configuration used mechanical cams. The mechanical actuation improved the repeatability and accuracy of the valves, matching or exceeding the performance of the electrically actuated prototype while eliminating all transducers and electronics. The mechanically actuated pump operated at 86% efficiency (full displacement) and 58% efficiency (25% displacement).

Keywords: Digital Hydraulics, Inline Piston Pump, Efficiency, Digital Pump/Motor

Target audience: Mobile Hydraulics, Digital Hydraulics, Piston Pumps

1 Introduction

A U.S. Department of Energy study reported that fluid power systems account for up to 5% of all energy transferred in the United States. Approximately 7-8% of the CO₂ emissions in the U.S. alone can be attributed to fluid power. During the time of this study, average fluid power system efficiency was 20%. Increasing this by just 5% would lead to savings of up to $20 billion and 90 million tons of CO₂ each year /1/. This is overwhelming motivation that supports the impact of improving system efficiency. Pumps are found in almost all fluid power systems and significantly influence system efficiency. As more efficient system architectures are being developed it is important that better pumps are available to support these advanced configurations /2/.

Digital hydraulics is one method of improving efficiency. Digital hydraulics systems are any systems "having discrete valued component(s) actively controlling system output" /3/. Digital hydraulics can be used at the system level or at the component level.

Improving pump efficiency will have a positive impact on reducing the energy losses in fluid power systems. State of the art pumps can be very efficient when operating at peak conditions, but when the pump displacement is lowered, efficiency can drop to as far as 30% /4/. Minimizing the drop in efficiency that comes from lowering operating displacement would greatly improve overall pump and system performance.

In order to improve pump efficiency at lower displacements, digital hydraulics is used to control the amount of fluid entering and leaving the pumping chamber for each piston. By placing a high speed on/off valve at the inlet and outlet port for each piston, the flow can be directed through either port as desired. Mechanical and electrical actuation of the valves were prototyped and tested. Operating strategies were developed to successfully allow for variable displacement operation.

2 Operating Strategies

When using high speed on/off valves there are multiple possibilities for controlling the displacement of pump/motors. All the methods described here, partial flow diverting (PFD), partial flow limiting (PFL), sequential flow diverting (SFD), and sequential flow limiting (SFL), control when the displacement chambers are connected to the pump/motor inlet and outlet, and do not vary the piston stroke. The piston stroke is always ‘full displacement’ and travels between maximum top and bottom range.

When operating as a pump using partial flow diverting (PFD), the inlet valve opens to let the pumping chamber fill with fluid as the piston travels to bottom dead center (BTC). While the piston makes its return towards top dead center (TDC), the inlet valve remains open allowing some flow to be diverted back into the low-pressure port. When the desired displacement volume is remaining in the pumping chamber, the inlet valve closes. After the fluid is pressurized, the outlet valve opens allowing this fluid to exit through the high-pressure port. This is shown in Figure 1.

For partial flow limiting pumping (PFL), the piston again starts at TDC with the inlet valve open. Shown in Figure 2, this valve only remains open until the desired flow volume is in the pumping chamber. Then the inlet valve closes and the piston continues to complete the entire intake stroke without allowing any more fluid to enter the chamber. At BDC there is a vapor void in the fluid and the piston begins the compression stroke towards TDC. Both valves remain closed until the fluid is pressurized. Once the chamber pressure reaches the same level as the system pressure the high pressure port opens and the fluid is pumped from the chamber into the system.

Figure 1: Partial flow diverting operating strategy /5/

Figure 2: Partial Flow Limiting /5/
In Sequential Flow Diverting (SFD) pumping, individual pistons operate at either full or zero displacement during an entire stroke cycle (BDC-TDC-BDC); none of the displacement chambers are partially filled as in PFD or PFL. A discrete number of pistons associated with the overall desired displacement operate at full displacement (for example, using two pistons on a five piston unit would provide 40% displacement). The remainder of the pistons operate at zero displacement by directing the fluid as follows. The output valve remains closed throughout the entirety of the cycle to prevent any flow through the high pressure port. The input valve remains open for the entire cycle (Figure 3). Similar to the PFD strategy, only the desired amount of flow stays in the chamber to be pumped out of the high pressure port. Since the desired amount of flow for these pistons is zero, all of the fluid is pushed back out of the low pressure port.

Sequential Flow Diverting (SFD) is similar to SFD as it uses a predetermined number of pistons operating at full displacement with the rest operating at zero. For the pistons operating at zero displacement, the outlet valve is still kept closed for this strategy. The difference from SFD is that the inlet valve also remains closed in this strategy. Flow limiting allows only the desired amount of flow to be taken into the chamber. Since the desired amount of flow is zero, the inlet valve never opens. Each pumping chamber creates a suction during this operating mode.

3 Pump Design

In order to build a digital variable displacement pump that would be comparable to existing units, the prototype pump was designed using an off-the-shelf pump as a starting point. A CAT Model 1861 fixed displacement 3-piston inline pump was used as the base unit for this research. The rotation of the input shaft is converted into the linear motion of the pistons through a crank-slider mechanism that can be seen in Figure 4. The CAT pump was selected mainly because of its modularity and the seals used. The block assembly that contains the displacement chambers and valves is easily removed to be replaced with the modified valve block necessary for digital displacement control. Lip seals are used between the displacement chambers and the pistons which separates the chambers from the crank case eliminating the need for a lubricating gap.

Two separate valve blocks were created; one designed for electrical actuation and the other for mechanical actuation. Both valve blocks utilize the same base unit and on/off valves, only the actuation method is changed.

3.1 Electrically Actuated Valve Block

The electrically actuated valve (EAV) design utilizes solenoid operated valves to control both input and output ports. When an electrical signal is sent to the valves a magnetic coil opens the valve. The EAV block is comprised of two parts. The first part of the block bolts to the crankcase. It contains a low-pressure seal that prevents leakage to the exterior of the pump and a drainage port which returns any fluid to the case. The second part of the block houses the displacement chambers and the valves for each piston. Each displacement chamber has four internal connections shown in Figure 5. Port A and Port B are the operating ports that can both act as the high or low pressure ports allowing for four-quadrant operation. Port S connects to a check valve as a safety mechanism that will drain back to tank if both ports A and B are closed and too high of a pressure is reached. The last connection is to a pressure transducer.

Sequential Flow Limiting (SFL) is similar to SFD as it uses a predetermined number of pistons operating at full displacement with the rest operating at zero. For the pistons operating at zero displacement, the outlet valve is kept closed for this strategy. The difference from SFD is that the inlet valve also remains closed in this strategy. Flow limiting allows only the desired amount of flow to be taken into the chamber. Since the desired amount of flow is zero, the inlet valve never opens. Each pumping chamber creates a suction during this operating mode.

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The EAV unit is controlled using the National Instruments Veristand software environment. A PXI-8108 controller runs a MATLAB Simulink model in the Veristand environment. The controller is responsible for opening and closing the on/off valves which in turn controls the output pressure and operating displacement. Adjustments to the operating conditions are all made through the controller and require the pressure transducers to be accurate and operational.
3.2 Mechanically Actuated Valve Block

The mechanically actuated valve prototype (MAV) makes use of half masking cams to control the on/off valves at the displacement chambers. The half masking cams are two identical cams that are in the high state and low state 50% of the time. By rotating one cam relative to the other, the amount of time in the high state can range from 50%-100% and the amount of time in the low state ranges from 0%-50%. The MAV uses the same CAT pump and the same valve block. The actuation part of the valve block was redesigned for the MAV. Due to geometric constraints, on/off valves were not able to be placed at both ports. Instead, an on/off valve was placed at the inlet and a check valve was placed at the outlet. This limits, in this case, the operation to pumping only. In contrast to the EAV having four internal connections to the displacement chamber, the MAV has only three internal connections to the displacement chamber. Seen in Figure 7, the first connection is made to the on/off valve (1) at the chamber inlet. The second is the check valve at the outlet (2). Using a check valve at the outlet eliminated the need for the additional port used for a safety check valve in the EAV (3). The last connection to the displacement chamber is used for a pressure transducer that is used solely for data collection purposes; it is not needed for normal operation using the MAV.

Variable cam control is achieved by two parallel, dual-input, planetary gear systems. Each planetary set controls one half-mask of a variable cam. Figure 8 shows the gearing configuration. With this system, a self-locking worm gear can turn the ring of the planetary set allowing the sun gear to be phased relative to the rotation of the planetary carrier even while it is spinning. This is accomplished for both masks allowing them to be phased to any angle relative to each other and the pump shaft, which is tied to the planetary carrier, during operation. Due to the locations of the displacement chambers in this inline unit, three mechanically linked half-masking cams were used. In this system, one half-mask controls the displacement of the unit while the other can be adjusted to vary the valve timing and reclaim the maximum amount of compressed fluid energy from the pressurized fluid on the down stroke.

4 Experimental Testing

A regenerative test stand developed by Holland (2012) and Merrill (2012) was used to test both units. The necessary data acquisition equipment for both units was included on the test stand. Flow meters and pressure transducers were included on the stand at the inlet and outlet of the unit. Pressure transducers were also used in each of the displacement chambers in the valve block to monitor individual displacement chamber pressure. Both units were tested at 25%, 50%, 75%, and 100% displacement. Each displacement variation was tested at 34 bar and 103 bar at speeds of 300 and 500 rpm. The individual units were also tested under additional operating conditions.

5 Results

The stock CAT pump was tested independently on the experimental test stand. The hydraulic efficiency was measured for the same operating conditions as the MAV and EAV with the exception of 300 rpm /6/. Since the unmodified configuration is a fixed displacement pump, data is only available at 100% displacement.

5.1 EAV Results

The EAV was tested in all four operating strategies for both pumping and motoring. The overall hydraulic efficiency for pumping is shown in Figure 9. Sequential flow limiting was the most efficient operating strategy followed by sequential flow diverting, partial flow limiting and then partial flow diverting. It is important to note that the baseline maximum efficiency of the unmodified check ball unit under similar conditions is 91%, shown by the red circle on the 100% displacement vertical axis. This efficiency is somewhat lower than conventional bent axis and swashplate units at full displacement since an extra set of seals separate the crankcase chamber from the pumping pistons. However, this does allow the unit to pump water and corrosive fluids as originally designed as a check ball pump for car washes.
5.2 MAV Results

The MAV was only tested using partial flow diverting for pumping because of the geometric constraints of the design. Figure 11 shows the overall hydraulic efficiency for pumping at 300 rpm for the mechanically actuated prototype at each pressure.

Figure 11: Overall hydraulic efficiency for pumping at 300rpm

The EAV was also tested as a motor but the results are not relevant in this comparison as the MAV, as configured, cannot be tested as a motor. The results for the EAV motoring tests can be found in Holland (2012).

The electrical power required for control of the valves was also monitored for the EAV. The total electrical energy requirements can be seen in Figure 10. The order of operating strategies to require the most power is the inverse of the order of most efficient.

Figure 10: Electrical Energy Requirements, pumping, 500rpm /6/

Figure 12 shows the MAV overall hydraulic efficiency for pumping at 500 rpm at each pressure. The efficiency of the base CAT pump tested at 103 bar is included as a reference point. Comparing both figures shows the negative relationship increasing speed has on hydraulic efficiency.

Figure 12: Overall hydraulic efficiency for pumping at 500rpm
5.3 Comparison of Actuation Techniques

Because of the design of the mechanically actuated valve prototype, it was only operated in partial flow diverting for pumping. Figures 13 and 14 compare the overall hydraulic efficiency (electrical actuation energy for the EAV is not included) of the mechanical and electrically actuated prototypes using the partial flow diverting operation strategy. Data for the fixed displacement base CAT pump is included where it is available.

The MAV proved to be easier to operate than the EAV. There are no sensors or controllers required for use or adjustment. The EAV requires pressure sensors and a controller for normal operation as well as switching between operating strategies and adjusting the parameters. The MAV displacement and pre-compression timing can be adjusted by hand by turning two knobs. The EAV unit is smaller and is more compact compared to the MAV, but the wires and sensors do make the unit more complicated.

6 Conclusions

Mechanical actuation of the valves proved to be an attractive way to achieve high efficiency variable displacement units not requiring expensive sensors and electronics. The MAV achieved higher efficiency at 50% displacement and slightly higher efficiency at 75%. Although performance slightly dropped at 25% and 100%, the MAV unit performed almost as well or better than the EAV. The efficiency of the MAV could be increased by eliminating some of the gearing and not limiting the valves to be the same size as used in the EAV. The EAV, using solenoids, is more limited in valve size since actuation forces are relatively small compared to a mechanical cam system. In addition to the better performance, the mechanically actuated prototype was able to achieve variable displacement without the use of any electronics and the simple control of the turning of a knob. Additional improvements are possible by implementing this technology on a radial piston pump. Only one cam would be needed for each set of inlet and outlet valves, regardless of the number of pistons, and cam design (pressure angles) is easier with larger cam diameters.

7 Future Work

In the future, changing the piston orientation to a radial unit could provide multiple benefits. A radial unit would allow all valves to be controlled by a single central cam and would allow for a larger cam and less gearing. Improvements to solenoid operated valves would lead to greater success with the electrically actuated prototype. Faster valve transition time for larger valve areas would increase repeatability and allow for larger flows.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
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<tr>
<td>TDC</td>
<td>Top Dead Center</td>
</tr>
<tr>
<td>BDC</td>
<td>Bottom Dead Center</td>
</tr>
<tr>
<td>PFD</td>
<td>Partial Flow Diverting</td>
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<td>PFL</td>
<td>Partial Flow Limiting</td>
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<td>SFD</td>
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<td>SFL</td>
<td>Sequential Flow Limiting</td>
</tr>
<tr>
<td>EAV</td>
<td>Electrically Actuated Valves</td>
</tr>
<tr>
<td>MAV</td>
<td>Mechanically Actuated Valves</td>
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The efficiency shows the most variability at 50% displacement because this is the point where valve timing is the most critical. The valves must switch where the piston velocity and flow forces are at a maximum. This demonstrates the effects of the poor repeatability of the solenoid operated valves used on the EAV.

Besides the numerical data collected, there are many other ways to compare the two different actuation techniques. Both prototypes have different operating strategy capabilities. The EAV is able to operate using any four of the strategies and is able to actively switch between them. Thus it is capable of four quadrant operation. It can act as a pump or a motor independently in both CW and CCW. In the MAV prototype the outlet port is pressure actuated rather than controlled with an on-off valve like in the EAV. This eliminated the ability to use the sequential operating strategies. The half masking cams also limit the valves to being in either state a maximum of 50% of the time which eliminates partial flow limiting. This limits the MAV prototype to only partial flow diverting. Four quadrant operation can be enabled in the MAV by including an on/off valve at the outlet of each displacement chamber.
References


Three Dimensional Simulation of Gerotor with Deforming Mesh by using OpenFOAM

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A new-born design and construction of a mini gerotor metering pump with trochoidal-teeth is presented. The technical innovation in this new-born design is to study the fluid dynamic effects of interteeth and lateral clearances by using OpenFOAM toolbox, an open source CFD software. This work is based on two critical aspects, the deforming of the mesh following the solid gears rotation, a complex interaction between mesh and gear profile surface that has to maintain a moderate quality of the mesh, and the simulation by means of a new boundary condition of the interteeth contact, reproducing actual contact points between the rotors. The possibility of contact point simulation by means of a proper mesh motion model is also suggested.

Keywords: Gerotor pump, Computational Fluid Dynamics, Dynamic Mesh, Leakage

Target audience: Mobile Hydraulics, Design Process

1 Introduction

The technology of gerotor pumps progresses towards significant number of sectors such as life science, industrial and mechanical engineering. This remarkable growth is based on its three main advantages: simplicity, versatility and performance /1/. Moreover, recent paradigm, like environmental concerns, drive the industry towards additional applications leading to growing demand for pumps that can improve their efficiency /2/. As gerotor pump specifications become more demanding and design cycles shorter, the conceptual stage becomes the most valuable guide to a cost effective design process. Here, numerical simulation with open source software appears to be the cost effective design process that leads the designer to a new gerotor pump unit with satisfactory performance and efficiency indices. More specifically, with regard to miniaturized gerotor pumps, few studies are available in the open literature with the exception of Mancò et al. /3/. The current methods of analysis of conventional components cannot be directly applied to mini components owing to geometry scale factor, kinematic and fluid dynamics. In small scales, the current knowledge, design criteria and know-how used in conventional sizes become questionable, as it is conditioned by the demanding axial and radial clearances /4/.

In this work, a new-born design and construction of a mini gerotor metering pump (mGp) with trochoidal-teeth profile intended to work at low rotational speed and low pressure is presented. Owing to manufacture tolerances and gear working performance, an interteeth radial clearance could appear tip-to-tip on mated teeth and sideways/lateral axial clearance between body pump and trochoidal-gear set. The technical innovation in this new-born design is to study the fluid dynamic effects of interteeth and lateral clearances by using OpenFOAM toolbox, an open source CFD software. As a consequence, the leakage flow in the clearances can be estimated and its important fluid dynamics effects owing to the mini size of the gerotor pump and the low working pressure and rotational speed are shown. This work is based on two critical aspects with respect to numerical methodology: (i) the deforming of the mesh following the solid gears rotation, a complex interaction between mesh and gear profile surface that has to maintain a moderate quality of the mesh by means of a dynamic-coupled mesh interface and (ii) the simulation by means of a new boundary condition of the interteeth contact, reproducing actual contact points between the rotors. For the first point, instead of using deforming and local remeshing, the use of an arbitrary coupled mesh inter-face (ACMI) approach has been adopted. An arbitrary mesh interface (AMI) allows the simulation of fluid flow across adjacent disconnected mesh domains. Nevertheless, it has the limitation that both boundaries have to be completely covered by each other. An ACMI allows the use of an AMI with partially overlapped patches. For the second point, the interteeth contact is simulated adapting the viscous wall model, although it is explored the possibility of leaving out this model if a proper motion model from mesh can be found.

2 The mini-Gerotor pump

The mini gerotor pump (mGp) is an internal gear pump with trochoidal-teeth profile. Basically, a gerotor pump consists of a pair of gears: an inner rotor with external teeth called the inner/ internal gear and an outer ring with internal teeth called the outer/external gear (see Figure 1). The two gears are mated so that each tooth of the internal gear is theoretically always in sliding contact through the line of contact with a tooth of the external gear, i.e., interteeth contact occurs; these points are known as contact points named \( P_k \) in Figure 1. Both gears are eccentric and rotate in the same direction but at different speeds, since the external gear has one tooth more than the internal gear, and consequently, the internal gear is slightly faster than the external gear.

![Figure 1: The internal gear with trochoidal-teeth profile, where Z is the number of external gear teeth.](image-url)
3 The numerical model

The operating principle of a mini gerotor pump presents a main challenge in this numerical model: the mesh domain decomposition. The mesh domain decomposition comprises the interprofiles domain with dynamic meshing and the housing domain with static meshing. In previous works, the continuously deforming of the mesh in the interprofiles domain has forced remeshing when mesh quality decreases excessively. Then, the corresponding fields have to be mapped from one low-quality mesh to the new one. This method has been previously explored and reported with an external gear pump by the authors with 2D /5/ and 3D simulation /4,6/ with satisfactory results. Nevertheless, the leakage in the clearance disk (marked as “couple” disk in Fig. 3) at the top and bottom part of gears was not considered, and the fast mesh quality decay requires a very frequent remeshing and fields mapping. In the present work, these two issues are overcome.

3.1 The mesh

3.1.1 Meshing of pump case and clearance disk

The body pump fluid domain has been meshed from a 3D-CAD model, using snappyHexMesh. This tool provides a high-quality hexahedral-dominant mesh. Two steps are performed to generate the mesh for the gear pump fluid domain: background mesh and refine region mesh. The background mesh, generated with blockMesh, seems not to receive the attention that certainly deserves owing to its great importance to achieve a good final mesh of the fluid domain. The common used regular hexahedron element is not the most appropriate. Instead, a better mesh is obtained by using a radial configuration because of the main cylindrical geometry of the pump fluid domain. This background mesh along with the gerotor pump body geometry is shown in Fig. 3(a).

With regards to the clearance disk, in order to get more control of the number of layers, element type and cell size, the domain has been meshed directly with the blockMesh tool obtaining a layered structured mesh (see Fig. 3(b)).
In order to overcome this issue, it has been explored the generation of the interteeth domain mesh with an extrusion of a 2D mesh generated with NETGEN /8/. Again, in order to automatize the process, a python code has been created using the module of Salome /9/. The generated mesh is displayed in Figure 5. The cell sizes are now more uniform, but the mesh is unstructured and the quality is not so high.

In gap regions of the domain, where it will be contact between gears, the number of cell layers is less than in the case of blockMesh generated mesh. Nevertheless, it is considered that this cell density is not crucial for the fluid flow calculation, since, actually, there will be no relative flow in this zones when contact point is simulated.

On the other hand, one of the challenges pursued in the present work is to maintain the mesh quality in the interprofile fluid domain with the gears motion. There are two extreme approaches to this problem. First, the mesh is moved with the geometry, and it is remeshed when the quality reaches a minimum threshold value. This is the approach used in previous numerical works /5,6/ and, as pointed earlier, has the inconvenience that it requires frequent remeshing when the relative velocity is high. Second, a slip condition could be adopted for the mesh motion. In this case, the relative position of the mesh should remind constant whiles geometry moves. This leads to an incorrect cell size grading with teeth motion. The dynamic mesh process with these two extreme approaches and its results are depicted in Figure 6 for the motion in the zone of the first contact point (tip-tip contact point).

It is proposed to mix both approaches with a partial slip condition. In this approach the mesh does not move with the geometry, but with a particular angular velocity. This velocity should take a value lower than the angular velocity of the inner profile, which is the faster.

In previous works this velocity has been set has the average value between the velocity of internal profile and the velocity of the external profile. Also, the centre of rotation of the mesh has been placed in the average point between the centres of rotation of both profiles. This is a simple calculation that works properly with the contact point model proposed by this group in previous publication /1,2,6/, but has the inconvenient that the mesh does not follow the contact points positions, except for the first one (tip-tip), as shown in Figure 7.

A more accurate calculation of the angular velocity of mesh should take into account the velocity of the contact points. This angular velocity will be then not uniform, but it will be a function of the angular position of the mesh point and of the rotational position of profiles. It is a more laborious method, but it could worth if the contact point viscosity model can then be avoided.
On the other hand, the usual way to calculate the slip velocity, by projecting the velocity difference in the plane of the geometry (see Figure 6) works fine when the number of teeth is large and concavity is low (that is, normal to geometries do not differ much of radial direction) as reported in previous works /10/ and in the results section of the present paper.

Figure 8: Point mesh calculation with slip of difference of velocities method. \( \mathbf{v}_G \) is the velocity of the geometry (profile) points and \( \mathbf{v}_M \) is the velocity of the mesh points.

Alternatively, when normals to geometry have a considerable tangential component, it is more accurate to calculate the point mesh position by projecting the end point of expected mesh velocity (see Figure 9).

Figure 9: Point mesh calculation with projecting of mesh velocity method. \( \mathbf{v}_G \) is the velocity of the geometry (profile) points and \( \mathbf{v}_M \) is the velocity of the mesh points.

Finally, when mesh boundary points are moved, it is required to properly transfer this movement to the internal mesh. In this case, also two approaches can be considered. The first is to calculate the movement cells quality criteria. Unfortunately, official release of OpenFoam does not provide such functionality. Nevertheless, S. Menon /11/ developed a dynamic adaptive remeshing solver for tetrahedral meshes in OpenFoam, based on spring -analogy and the Mesquite library /12,13/. The second is to use the standard laplacian velocity based solver of OpenFoam. This group is exploring both options. The first requires a laborious process of coding and, probably, it should need a tetrahedral mesh, which would probably worsen the simulation performance. The second is giving bad quality mesh motion, but this group thinks that it can be improved with a proper mesh velocity definition.

3.1.3 The coupling of mesh domains

The arbitrary mesh interface (AMI) is an OpenFOAM tool related to disconnected, adjacent, mesh domains being particularly useful for rotating geometries, as it is the case presented in this work. The pump fluid domain and the interprofile fluid domain require separate meshes for static and rotating regions of geometry, respectively. To couple both meshes’ domains, the arbitrary couple mesh interface (ACMI) is used. The ACMI allows to couple patches that partially overlap with each other.

Special care and a precise and laborious procedure are taking to prepare the ACMI patches in the mini pump case in order to establish a methodology for the programmed dynamic simulation. Actually, in the first stage of the present project, a bug was detected in the calculation of flow mass through ACMI, but it was corrected in the version 4.0 of OpenFOAM.

Another important issue related with the ACMI is the handling of a dynamic mesh. In general, when the meshes associated to the ACMI are moving as rigid solid, it works correctly. But, when the dynamic mesh is handled by the laplacian solver, the velocity interpolation assigns an average velocity, between its corresponding velocity due to interprofiles domain mesh motion and the null mesh velocity of disk points, to the interface points, leading to an incorrect deformation of the cells in the interteeth interface zone. This generates high skew and, eventually, negative volume cells, in the mesh interface. This issue has been overcome by defining the point motion in the ACMI adjacent to interteeth domain with the velocity in an “offset” plane (typically, 1 mm above the interface) and overriding the velocity calculated by the solver. This is a considerable computation time consuming procedure, but the alternative is diving in the ACMI code in order to provide a way to separate motion zones, which are beyond the scope of the present work.

3.2 The solver

The solver, called gerotorDyMFoam, has been created as a modification of the pimpleDyMFoam, included in the OpenFoam distribution. The solver uses the finite volume method (FVM) to solve the transient Navier–Stokes equations for incompressible fluid flow. It allows the use of relatively large time steps thanks to the hybrid PIMPLE algorithm. The main modification on the solver is the inclusion of the contact point model as a transport model for viscosity, although this model could be avoided when a proper definition of the mesh velocity is properly set, as explained in section 3.1.2. Since the present work is focused mainly in the development of numerical method for ACMI and dynamic mesh, no turbulence model has been considered for the sake of simplicity, and all the simulations are laminar.

4 Results and Discussion

So far, results with large number ( \( Z=11 \) ) of smooth teeth, with viscosity contact point model, can be presented. Simulations with projected mesh motion and high concavity teeth are still in progress, since it is not straightforward to maintain mesh quality. The reward will be to leave out the contact point viscosity model, since contact points will be constrained to mesh motion.

The simulations of a gerotor pump of theoretical volumetric capacity \( Q_t=7.83\times10^{-6} \text{ m}^3/\text{s} \) have been conducted with a 5.25 Mcells with a clearance disk of 50 \( \mu \text{m} \) in a HPC cluster using 64 cores. The instantaneous flow rate, normalized with theoretical one, is displayed in Figure 10. Geometric flow rate, computed with GeroLAB /10/ is also shown for the sake of comparison.
5 Summary and Conclusion

A new method for simulation of a new born mini gerotor pump is presented. This method is based on the mesh decomposition in three parts. The pump case is meshed with snappyHexMesh, and the clearance disk with blockMesh. Two alternatives are suggested for the interteeth domain. On one hand, it can be meshed with blockMesh, obtaining a structured high quality mesh. On the other hand, it can be meshed with NetGen. Also, the possibilities of surface mesh in profiles and internal mesh have been discussed. The maintenance of mesh quality is not straightforward when the number of teeth is low and the concavity of interprofiles chamber is high. This is a work in progress. Results have been presented for high number of teeth and low concavity, showing a leakage of 3% in the clearance disk and revealing the peak of leakage in-between gearing cycles.

6 Acknowledgements

This research program, Project No. DPI2013-42031-P, receives financial support from the Ministry of Economy and Competitiveness of Spain and is co-financed by FEDER funding of EU; authors acknowledge this financial support. The authors also would like to acknowledge the companies AMES Spain and MAVILOR Spain for providing the necessary support.

The authors thankfully acknowledge the computer resources, technical expertise, and assistance provided by the Supercomputing and Visualization Center of Madrid (CeSViMa) and the Spanish Supercomputing Network (RES), Project No. FI-2017-2-0005.

References


Experimental study on churning losses reduction for axial piston pumps

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The proportion of churning losses increases significantly with the increasing speed, thus churning losses reduction has a significant influence on the efficiency improvement in axial piston pumps. In this paper, a test pump with nano-coating is proposed, and analyzed in details. The analysis shows that the surface energy and friction coefficient on the outside surface of cylinder block are reduced due to the decrease of surface roughness and wettability on the nano-film. Experimental results indicate that energy losses of the proposed nano-coated test pump are reduced by 12–37%. Some of the conclusions in this paper may provide a suitable novel guidance for improving the friction-reducing abilities in axial piston pumps.

Keywords: Churning losses, cylinder block, nano-coating, axial piston pumps
Target audience: Pumps & Motors, Aeronautics and Astronautics

1 Introduction

Axial piston pump is one of the most important hydraulic power unit of fluid power systems due to its high power density. However, it also has obvious disadvantages such as low efficiency and high noise level, especially with the increase of operating parameters. The energy losses in axial piston pump have mainly three parts, the volumetric losses caused by the leakage flow losses, the mechanical losses due to the friction losses and churning losses caused by the internal rotating components stirring the hydraulic fluid. The impulse losses of the fluid flow and compression losses are the rest sources of losses.

Volumetric losses and mechanical losses have been much investigated in axial piston pumps to reduce energy dissipation in the past few decades. The main focus is on the piston/cylinder block pair, slipper/swash plate pair and cylinder block/valve plate pair. Murrenhoff et al. /1, 2/ showed that coated pistons and surface texturing of axial piston pumps on the energy efficiency can reveal very low friction between piston and bushing. Ivantysynova et al. /3, 4/ also presented a contoured piston to improve the load carrying ability of the gap in an axial piston pump. Hooke et al. /5, 6/ measured clearances under slippers subjected to tilting couples and then confirmed the theoretical predictions. The numerical models with detailed valve plate and piston barrel kidney geometries have been used to analyze and optimize the pump design to improve the efficiency /7, 8/. Xu et al. /9/ introduced several suggestions for decreasing extra cylinder block tilting moments and inhibiting the tilting effect on the volumetric losses and mechanical losses in a high-speed axial piston pump.

The churning losses, which are caused by the internal rotating components stirring the fluid in axial piston pumps, have been a research focus in recent years because its effect is major for pumps with high speed of 10000–15000 rpm applied in aeronautics and astronautics, in comparison with axial piston pumps with nominal speed of 1500–3000 rpm. The first theoretic approach was done by Jang /10/ in 1997. Xu et al. /11, 12/ investigated the primary distribution of churning losses in axial piston pumps at various ranges of speed. The company Parker added a power boost insert unit in hydraulic motors, and the effective power output increased by 5 kW but only at the 9400 rpm rotational speed /13/. In recent years, Murrenhoff et al. /14, 15/ further analyzed the insert for flow optimization in an axial piston pump by CFD-simulations, and the benefit of the insert was up to 3% for an axial piston pump at high rotational speeds on the test-bench. Rahmfeld et al. /16/ invented a dry bent-axis motor with 5000 rpm rotational speed, the dry casing can effectively reduce the churning losses, but the inevitable friction will generate heat without oil lubrication cooling.

The goals of all the researches /10-16/ on churning losses reduction of axial piston pump are to design and optimize the fluid state in the casing filled with oil. However, the weight and heat of the axial piston pump may be increased obviously if the insert and dry casing cannot be designed correctly. Moreover, some of these methods have satisfactory results on churning losses reduction only at high speed. Thus, this paper proposes a test pump with nano-coating to reduce the churning losses both at low speed and high speed. Despite the fact that effects of nano-coating on churning losses have no evidence in the existing literature, there are some detailed researches on nano-coated slippers to reduce the friction between slipper and swash plate at low speed in axial piston pumps. Rizzo et al. /17-19/ showed that more than 20% friction reduction can be achieved using the nano-coating methodology. The contributions of nano-coated slippers play an important role in guiding the study of churning losses in axial piston pumps.

The aim of this paper is to analyze the effects of the nano-coating on churning losses in axial piston pumps at both low speed and high speed. The effect of nano-coating on contact angle, surface energy, friction coefficient and churning losses of axial piston pumps are investigated based on the test bench. Some encouraging results are obtained with nano-coated test pump, which can be used in the design of high-efficiency axial piston pump.

2 Estimation of churning losses

The two major sources of churning losses are illustrated in Figure 1, the churning losses associated with the cylinder block and the churning losses associated with piston/slider units. One is the viscous friction force due to the rotation of cylinder block, and the other is the pressure drag due to the circling motion of the pistons and slippers in the casing filled with oil.

Figure 1: Two major sources of churning losses.

Figure 2 shows the experimental results of churning losses torque due to the rotating cylinder block and circling pistons at various speeds according to Ref. /12/. The rotating cylinder block has more influence on churning losses than the circling pistons at high speed in axial piston pumps as shown in Figure 2. It can be explained by that the total energy dissipation due to the rotating cylinder block transforms laminar viscous friction losses into turbulent shear stress losses. And the drag coefficient due to the circling pistons decreases among pistons. It can be concluded that the effect of the rotating cylinder block on churning losses in axial piston pumps should be paid more attentions at high speed. Thus, the analysis in this paper is major on the outside surface of the cylinder block, which is the contact surface between the cylinder block and fluid in the circumferential direction.
3 The functional layer and samples characterization

The rotating cylinder block is a dominant influence on churning losses at high rotation speeds, so the outside surface of the cylinder block is tried to evaluate its response to the nano-coating, which is the contact area between the cylinder block and fluid in the circumferential direction as shown in Figure 3. The surface of the nominal cylinder block has a metallic luster. On the contrary, the surface of the cylinder block after nano-coating forms transparent film. It shows the non-metallic luster and the reflected light is weak compared with the surface of nominal cylinder block. The coating is made from a colloidal suspension in water of the metal alkoxides in presence of an acid catalyst. Then the coating metal oxide is sprayed on the outside surface of the cylinder block and is molded by high sintering. The handled surface is tested to explore its characteristics, namely contact angle with water, contact angle with HLP 32 hydraulic oil and surface energy.

Figure 3: Nominal cylinder block and nano-coated cylinder block.

Figure 4 shows the scanning electron microscopy (SEM) images of the cylinder block surfaces, which are before and after nano-coating. The cylinder block surface becomes smooth after nano-coating, indicating less roughness on the surface. The roughness is reduced from 0.96 μm to 0.15 μm, which is down by 84% after nano-coating from the confocal laser scanning microscopy (CLSM) results. And the typical nano-oxide particles appear on the cylinder block surface. The energy dispersive spectroscopy (EDS) results show that silicon content increases significantly in comparison with iron content after nano-coating as shown in Figure 5. The EDS results further indicate that there exists nano-sized silicon oxide and forms the nano-film on the cylinder block surface after nano-coating.

Table 1: Static contact angles in different samples.

<table>
<thead>
<tr>
<th>No.</th>
<th>Before nano-coating</th>
<th>After nano-coating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>96.51° ± 5.1°</td>
<td>108.07° ± 5.6°</td>
</tr>
<tr>
<td>Oil</td>
<td>29.2° ± 3.2°</td>
<td>64.7° ± 3.0°</td>
</tr>
</tbody>
</table>

From contact angle data, the following information can be obtained that the static contact angles of both liquids after coating are significantly larger than those before coating. It can be indicated that nano-coating results in a decrease in the wettability of the two liquids on the cylinder block surface.

In addition, the surface tension can be measured by the thin layer method, the adsorption process of the adsorbent material can be dynamically recorded and the surface energy of the solid is calculated. The surface tension of pure water and HLP 32 hydraulic oil under the corresponding experimental conditions are $\gamma_{water-vapor}$ and $\gamma_{oil-vapor}$.
The measured static contact angle with the surface tension is obtained by the classical Young's equation

\[ \gamma_a \cos \theta = \gamma_a - \gamma_d \quad (1) \]

where \( \theta \) is the static contact angle, \( \gamma_a \) is the surface tension (mJ/m²) between the liquid and vapor, \( \gamma_d \) is the surface tension (mJ/m²) between the solid and vapor, and \( \gamma_a \) is the interfacial tension between the solid and liquid (mJ/m²).

The equation can be deduced by the Berthelot (geometric mean method) rule given by D.Y. Kwok and A.W. Neumann /28/

\[ \cos \theta = -1 + 2\beta \exp \left( -\beta \left( \gamma_a - \gamma_d \right) \right) \quad (2) \]

where \( \beta \) is the experimental constant (m²/mJ)².

Because the surface energy values of both samples can be calculated by Equation (2), the nano-coating clearly reduces surface energy with respect to an untreated sample. Low surface energy is a fundamental parameter, together with surface nanostructure, to obtain oleophobicity /17-19/, which is the repellence towards low surface tension liquids. This characterization clearly shows that the best performances in terms of high repellence and low surface energy are obtained for the cylinder block after nano-coating.

4 Performance evaluation on test bench

It is not so easy to analyze the churning losses directly using a piston pump because the friction losses cannot be segregated among the three friction pairs and these frictions have a high influence on the measured churning losses. Therefore, a churning losses test bench has been designed at the State Key Laboratory of Fluid Power and Mechatronic Systems, Zhejiang University in order to assess the behavior of churning losses in axial piston pumps /12/.

As shown in Figure 7, a churning losses test bench consists of a test pump, a temperature sensor, a pressure sensor, a torque/speed sensor and an electric motor. The details of the related sensors are listed in Table 2. The test pump is designed based on a high-speed axial piston pump prototype with a theoretical volumetric displacement of 3 ml/r used for aircraft in Reference /9/.

The geometric dimensions and information regarding the test pump can be seen in Table 3. The test pump has only a rotation cylinder block in order to eliminate the friction losses, and the shaft is supported by one pair of identical shielded and sealed ball bearings. The oil is only contacted with the fixed end faces of the shields in bearings, and the rolling elements between the inner ring and the outer ring are immersed in the greases. The bearings are at the same grease lubrication under the wet and dry casing. Thus, the churning losses power is deduced by subtracting the wet and dry casing torque acting on the shaft as shown in

\[ P = M_d \omega = (M_c - M_w) \omega \quad (3) \]

where \( \omega \) is the rotation angular velocity, \( M_c \) is the experimental churning losses torque, \( M_w \) is the experimental torque acting on the shaft with oil in the casing, and \( M_d \) is the experimental torque acting on the shaft without oil.
The churning losses power can be reduced by 12–37% with nano-coating applied on the traditional cylinder block in axial piston pump. It should be mentioned that some of the conclusions in this paper may provide a suitable novel guidance for improving the friction-reducing abilities in axial piston pumps in the future.

6 Acknowledgements
This work was supported by the National Basic Research Program of China (973 Program) (No. 2014CB046403) and the National Natural Science Foundation of China (No. U1509204).

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>γw-vap</td>
<td>Surface tension of pure water</td>
<td>[mJ/m²]</td>
</tr>
<tr>
<td>γoil-vap</td>
<td>Surface tension of HLP 32 hydraulic oil</td>
<td>[mJ/m²]</td>
</tr>
<tr>
<td>γlv</td>
<td>Surface tension between liquid and vapor</td>
<td>[mJ/m²]</td>
</tr>
<tr>
<td>γsv</td>
<td>Surface tension between solid oil and vapor</td>
<td>[mJ/m²]</td>
</tr>
<tr>
<td>γsl</td>
<td>Interfacial tension between solid and liquid</td>
<td>[mJ/m²]</td>
</tr>
<tr>
<td>θ</td>
<td>Contact angle</td>
<td>[degree]</td>
</tr>
<tr>
<td>β</td>
<td>Experimental constant</td>
<td>[(m²/mJ)²]</td>
</tr>
<tr>
<td>Pc</td>
<td>Experimental churning losses power</td>
<td>[W]</td>
</tr>
<tr>
<td>ω</td>
<td>Rotation angular velocity</td>
<td>[rad/s]</td>
</tr>
<tr>
<td>Mc</td>
<td>Experimental churning losses torque</td>
<td>[Nm]</td>
</tr>
<tr>
<td>Ms</td>
<td>Experimental torque acting on the shaft with oil</td>
<td>[Nm]</td>
</tr>
<tr>
<td>Md</td>
<td>Experimental torque acting on the shaft without oil</td>
<td>[Nm]</td>
</tr>
</tbody>
</table>

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Investigation of the Aerodynamics Characteristics of the Integrated Motor-Compressor

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The objective of this work is to design and investigate the aerodynamic performance of a novel integrated motor-compressor. The integrated motor-compressor integrates the axial-flow compression into the electromagnetic function by designing the airfoil-shaped rotor of the electric machine to provide compression. Hence, the integrated motor-compressor is both an axial-flow compressor and an electric machine. It is capable of providing axial flow compression and electromagnetic torque at the same time. In this work, the aerodynamic design of the proposed machine is done and evaluated by both analytical method and computational fluid dynamics (CFD). The effect of attack angle to the blade lift and drag force is investigated. The effect of solidity to the axial-flow compressor performance is also evaluated. The electromagnetic performance of the proposed machine is investigated by motor sizing equations and finite element analysis (FEA).

Keywords: Aerodynamics, axial-flow compressor, electric motor, FEA, CFD

Target audience: Pumps and Motors, Novel Production Technologies

1 Introduction

Compressors are commonly defined as dynamic compressors and positive displacement compressors. The positive displacement compressors trap and transport the fluid in a confined space to achieve high pressure. The dynamic compressors dynamic transfer the energy in the continuously flowing fluid to achieve a pressure increase. Axial-flow compressors and centrifugal compressors are two basic types of dynamic compressors. The radius of flow streamlines in a centrifugal compressor is increasing while the radius of an axial-flow compressor is almost constant through all the stages. The centrifugal compressor has larger stage pressure increase than the axial-flow compressor. While in an axial-flow compressor, it has multiple stages to achieve large pressure ratio, and it has a significantly larger mass flow rate than a centrifugal compressor /1/.

Axial-flow compressors are widely used in gas turbines, aircraft engines, power plants and gas transportation stations. There are multiple stages in axial-flow compressor and a compressor stage consists a rotor stage and a stator stage. Flow in an axial-flow compressor is accelerated by the rotating compressor rotor stages and diffused by the compressor stator stages to achieve high-pressure rise. Axial-flow compressors have the advantages of very high mass flow rate, high efficiency, and high total pressure concerning /2/. Figure 1(a) shows STC-SX model of axial-flow compressor designed by Siemens which can be used for blasting furnace air and FCC (Fluid Catalytic Cracking) air; it is typically driven by the steam turbine, and electric motor depends on energy resources and application types. Figure 1(b) shows an example of in-line electric motor driven STC-SX axial-flow compressor for an FCC plant /3/. The in-line motor-driven compressor has the advantages of faster control speed, highly flexible, and higher efficiency compared with the conventional turbine-driven axial-flow compressors /4/. However, the in-line motor driven axial-flow compressor has challenges on motor cooling, system compactness, and motor locations as the axial-flow compressor need to rotate at a very high speed to accelerate the flow. The complicated structure results in the large size of the in-line motor-driven compressor system. In addition, the concerns of high-speed electric motor which can be used to drive compressors are increasing. Windage losses and temperature issues have become caught more attention for high-speed electric motor design /5/.

In order to solve the challenges of in-line high-speed electric motor-driven axial-flow compressor, a novel design that integrates an electric motor with an axial-flow compressor is proposed and is shown in Figure 1(c) /6/. The rotor of the electric motor is curved as airfoil; it can provide both electromagnetic torque and accelerating the flow of compression function. In Figure 1(c), the integrated motor-compressor is working as the first stage of the multistage axial-flow compressor, and it is connected to the later compressor stages through the same shaft. Hence, the integrated motor-compressor is part of the multistage compressor, and it can provide mechanical energy to drive the whole compressor system /7/. The proposed machine can increase the compactness of the compressor system and provide self-cooling benefit.

However, few researches have been done on investigating the aerodynamic characteristics of the integrated machine /D2/. Hence the contribution of this paper is to investigate both the electromagnetic performance and aerodynamic performance of the proposed integrated motor-compressor. In Section II, the electromagnetic and aerodynamic design of the proposed integrated machine is presented by analytical method. The aerodynamic performance is provided in Section III by meanline method and computational fluid dynamics (CFD). Section IV provided the electromagnetic performance of the proposed machine by finite element analysis (FEA). Section V follows the conclusion.

2 Electromagnetic and Aerodynamic Design of the Integrated Motor-Compressor

The design of the flux-switching integrated motor-compressor is electromagnetically and aerodynamically cross-coupled. The dimensions of the proposed machine have a very large impact on the performance of motoring and compression aspect. In this section, the electromagnetic and aerodynamic design of the integrated motor-compressor is provided.
2.1 Electromagnetic Design

The rotor pole of the electric motor needs to be robust as the airfoil-shaped rotor of an integrated motor-compressor needs to spin at a very high speed to provide compression function. In addition, permanent magnets should be on the stator instead of on the rotor for reliability concerns. Therefore, electric motor with salient pole rotor is a good candidate for the integrated motor-compressor. The candidate electric motor topologies for the integrated motor compressor includes flux-switching permanent magnet (FSPM) machine, doubly salient PM (DSPM) machine, flux reversal PM (FRPM) machine, and switched reluctance machine (SRM) can be used as the motor topology of the proposed machine which is shown in Figure 2. Among the candidates, 12-slot, 10-pole FSPM machine shown in Figure 2(d) is selected as the motor topology for the integrated motor-compressor for its large torque density, sinusoidal back-EMF, and robust structure which are suitable for high-speed operations.5/.

The electric motor can be designed by sizing equations, and the electromagnetic performance can be calculated by the modified sizing equations for the integrated motor-compressor. The fundamental frequency of the FSPM machine is calculated in Equation (1) where \( \omega \) is the rotating speed in rpm, \( N \) is the number of rotor poles.

\[
f_e = \frac{N \omega}{60}
\]

(1)

The peak value of back-EMF is calculated in Equation (2) where \( K_m \) is the magnetizing coefficient, \( K_t \) is the stator tooth width to pole pitch ratio, \( K_{aero} \) accounts for the airfoil curvature effect of the rotor pole, \( N_t \) is the number of stator slots, \( N_e \) is the winding turns per phase, \( L_e \) is the effective length of the machine, \( D_r \) is the stator inner diameter, and \( B_{0,zt} \) is defined as peak airgap flux density.

\[
E_{st} = 2\pi^2 N_e B_{0,zt} K_m K_t K_{aero} \left( \frac{L_e}{N_t} \right) D_r L_e
\]

(2)

The torque and output power of the electric motor are calculated in Equation (3) and (4) where \( \eta \) is the motor efficiency, \( A_{w,zt} \) is the electric loading in A/m.

\[
P_e = \frac{\sqrt{2} E_{st}^2}{2} K_m K_t K_{aero} N_e B_{0,zt} A_{w,zt} D_r L_e
\]

(3)

\[
T_e = \frac{\sqrt{2} E_{st}^2}{4} K_m K_t K_{aero} N_e B_{0,zt} A_{w,zt} D_r L_e
\]

(4)

2.2 Aerodynamic Design

As the rotor of the integrated motor-compressor is the rotor of both an electric motor and axial-flow compressor, the shape of the rotor needs to be carefully designed that there is an inherent trade-off associated with the design of the compressor blade. Specifically, the width of the blade should not be too thick in order to reduce the blade profile loss, however from an electric motor standpoint the width of the rotor pole must be thick enough to avoid saturation. Hence, tradeoff of the blade thickness is a key concern in this design. The NACA65(4)-421 blade is selected for the rotor for its large thickness which satisfies the tradeoffs of the blade thickness concerns.11/ Figure 3 shows the compressor rotor blades nomenclature with flow directions. Figure 4 shows the corresponding velocity triangle calculating the flow properties where \( V \) is the flow velocity, \( U \) is the blade velocity, \( W \) is the flow relative velocity to the blades which need to be at least 0.3 Ma for measurable pressure increase.1/ The subscription 1 and 1.5 refers to entering and exiting the rotor stage. The flow angle \( \theta \) in Figure 3 equals the sum of flow attack angle \( \alpha \) and blade stagger angle \( \gamma \). The flow turn angle \( \gamma \) reflects the deflection angle of the relative velocity before and after the rotor stage. The axial-flow compressor solidity \( \sigma \) is defined as the ratio of blade chord length \( c \) over blade pitch \( s \), and the solidity is commonly small to avoid large incidence losses.9/ The aspect ratio \( AR \) of the blade equals to the blade height over blade chord length.

2.2.1 Dimensions

The design of the dimensions of the integrated motor-compressor is mainly constrained by the trade-off between electromagnetic and aerodynamic requirements. The solidity and blade aspect ratio is the key parameter which can be used as a metric to design the integrated motor-compressor. NACA65(4)-421 blade has a blade thickness of 21% of blade chord length, and it is selected for the integrated machine. As in the FSPM machine, when the thickness of the rotor pole is 1.4 times of the thickness of the stator tooth, maximum back-EMF can be achieved. Therefore, in this work, the thickness of rotor pole should be larger than that of the stator tooth. The blade thickness can be calculated as a function of solidity shown in Equation (5) where \( h_t \) is the hub-to-tip ratio defined as the ratio of the radius of rotor hub to rotor tip, and \( D_r \) is the rotor diameter. The blade thickness can also be derived as a function of aspect ratio shown in Equation (6). The relations between aspect ratio and solidity is derived in Equation (7). In axial-flow compressors, the hub-to-tip ratio is usually greater than 0.7/2, and in this design, the hub-to-tip ratio is set to be 0.7.

\[
B_r = 0.21 \sigma \left( \frac{1}{2} h_t - 1 \right)
\]

(5)

\[
B_r = 0.21 \frac{D_r (1 - h_t)}{2AR}
\]

(6)

\[
AR = \frac{N_t (1 - h_t)}{\pi \sqrt{h_t^2 + 2}}
\]

(7)

The stator tooth thickness \( S_t \) is calculated by Equation (8), where \( \gamma \) is the blade stagger angle.

\[
S_t = \frac{\pi (D_r + 2)}{36} \cos \gamma
\]

(8)

The dimension of the rotor can be calculated by adjusting the solidity and aspect ratio. In this design, the solidity is set to be 1.5, the aspect ratio is set to be 0.375, and the blade chord length is set to be 40 mm. Then the rotor diameter is designed to be 100 mm satisfying the blade thickness requirement and is about 1.2 times of the stator tooth thickness.
2.2.2 Design of Attack Angle, Incidence Angle and Deviation Angle

The attack angle, incidence angle and deviation angle need to be designed to define a minimum-loss inlet and outlet angle for the blade /10/. The attack angle is designed based on achieving smooth blade surface pressure distributions and is formulated in Equation (9). For NACA 65 series blade, \( \alpha = 1, K_t = 1 \) is the incidence correlations, \( \alpha/c = 0.5 \) defines maximum camber occurs at half chord.

\[
\alpha' = 3.6K_tK_{c2} + 0.35720\left(\frac{V^2}{c}\right)\frac{\rho_0}{\rho} - 0.06
\]  

The designed incidence angle is formulated in Equation (10) based on equivalent circular arc camber lines, where \( \alpha' \), zero camber incidence angle and \( m \) is correlation slope factor /11/.

\[
\iota = K_{i1}(\alpha' + n \theta)
\]  

The designed deviation angle is formulated in Equation (11) corresponding to the designed incidence angle, where \( \delta' \), zero camber deviation angle, and \( m \) is correlation slope factor /12/.

\[
\delta = K_{i1}(\delta' + m \theta)
\]

The deflection angle of the blade then is derived in Equation (12) showing the turn angle of the flow by the blade.

\[
\beta_{21} = \theta + \iota - \delta
\]

The designed attack angle, incidence angle, deviation angle, and deflection angle are functions of solidity, and they are shown in Figure 5 and Figure 6. In this design, as the solidity is set to be 1.5, therefore the designed attack angle, incidence angle, deviation angle, and deflection angle are 10 degrees, 6.7 degrees, 6.1 degrees, and 10.6 degrees, respectively.

2.2.3 Velocity Triangle Calculation

As is shown in Figure 4, the flow entering the rotor stage is accelerated by the rotor, and the flow relative velocity is turned by the deflection angle. The rotor operating speed need to be very high to enable pressure rise as the rotor diameter is only 100 mm, therefore in this design, the rotating speed is set to be 35000 rpm. In the velocity triangle, the speed and angles are calculated using meanline method; then the blade meanline speed is 155.8 m/s.

In order to determine the flow inlet absolute velocity, there are several parameters need to be considered such as De Haller number, flow coefficient, hub-to-tip-ratio, degree of reaction, and blade loading coefficient. De Haller number determines diffusion rate of an axial-flow compressor, and it is calculated by outlet relative velocity over inlet relative velocity. De Haller number is commonly larger than 0.7 to avoid the separation of the boundary layer. Flow coefficient determines the ratio of inlet velocity to the blade velocity. Hub-to-tip ratio determines the ratio of rotor hub to the blade tip, and it is usually greater than 0.7. The degree of reaction is defined as the ratio of the change of enthalpy across the rotor to the entire stage. Blade loading coefficient is defined as rotor work over blade kinetic energy. In heavy loaded stages, the blade loading coefficient is larger than 1 /2/.

As the proposed integrated motor-compressor locates at the first stage of the multistage axial-flow compressor, without inlet guide vanes, the flow can enter the rotor stage axially. The inlet flow absolute velocity is set to be 140 m/s. Then, the De Haller number equals to 0.84; flow coefficient equals to 0.89, blade loading coefficient equals to 0.6, degree of reaction equals to 0.84. It is noted that the stage with greater than 0.5 reaction has low exit loss for the small static pressure rise in the stationary blades. The blade loading coefficient and degree of reaction at different solidity are shown in Figure 7 /1/.

The compressor performance can be calculated using the velocity triangle and meanline method. The power required by the rotor to turn the flow is calculated in Equation 13 where \( m \) is mass flow rate, \( r \) is the rotor radius, \( \omega \) is the rotational velocity in rad/s.

\[
P_{rot} = \frac{m \cdot r \cdot \omega}{2}\]

The stagnation temperature before and after the rotor and total pressure rise can be calculated by Equation (14), (15), and (16), respectively. \( T_0 \) is the stagnation temperature in K, subscripts 1 and 1.5 refer to the inlet and outlet of the rotor stage, respectively. The work done by the rotor is defined as \( w_{12} \), \( c_p \) is the constant-pressure specific heat of gas, and \( h \) is the heat capacity ratio. The total pressure ratio of the proposed machine at different solidity are shown in Figure 8. It can be seen that the total pressure increases with the increase of solidity.

\[
T_{1.5} = T_0 + \frac{w_{12}}{c_p}
\]

\[
T_{1.5} = T_0 - \frac{w_{12}}{c_p}
\]

\[
P_{11} = \frac{P_{11.5}}{P_0} \left(\frac{T_{1.5}}{T_0}\right)^{k-1}
\]

The rotor stage efficiency considering stagnation pressure losses is calculated in Equation (17) where \( \zeta_{43} \) is the stagnation pressure loss coefficient for the rotor, and it is defined as 0.014 \( \alpha/c \) /13/.}

\[
\eta = 1 - \frac{0.014\alpha/c}{\cos\alpha_1 \cos^2(\alpha - \theta_{21})/2\rho}
\]
3 Aerodynamic Performance of the Proposed Integrated Motor-Compressor

The design parameters are shown in Table 1.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor speed (n) [krpm]</td>
<td>35</td>
</tr>
<tr>
<td>Hub diameter (D_{hub}) [mm]</td>
<td>70.0</td>
</tr>
<tr>
<td>Stator stack length [mm]</td>
<td>20</td>
</tr>
<tr>
<td>Stator outer diameter (D_o) [mm]</td>
<td>150</td>
</tr>
<tr>
<td>Blade tip diameter (D_{tip}) [mm]</td>
<td>100.0</td>
</tr>
<tr>
<td>DeHalller number</td>
<td>0.84</td>
</tr>
<tr>
<td>Hub-to-tip ratio</td>
<td>0.7</td>
</tr>
<tr>
<td>Flow coefficient</td>
<td>0.89</td>
</tr>
<tr>
<td>Blade loading coefficient</td>
<td>0.6</td>
</tr>
<tr>
<td>Degree of reaction</td>
<td>0.84</td>
</tr>
<tr>
<td>Solidity</td>
<td>1.5</td>
</tr>
<tr>
<td>Aspect Ratio</td>
<td>0.375</td>
</tr>
<tr>
<td>Camber angle, (\theta) [deg.]</td>
<td>10</td>
</tr>
<tr>
<td>Designed attack angle, (\alpha) [deg]</td>
<td>10</td>
</tr>
<tr>
<td>Designed incidence angle [deg]</td>
<td>6.7</td>
</tr>
<tr>
<td>Designed deviation angle [deg]</td>
<td>6.1</td>
</tr>
<tr>
<td>Deflection angle [deg]</td>
<td>10.6</td>
</tr>
<tr>
<td>Axial flow velocity, (V_1) [m/s]</td>
<td>140</td>
</tr>
<tr>
<td>Mass rate of flow [kg/s]</td>
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<tr>
<td>Compression Power, (P) [kW]</td>
<td>5.07</td>
</tr>
<tr>
<td>Stage efficiency</td>
<td>98.3%</td>
</tr>
<tr>
<td>Ideal total pressure rise</td>
<td>8.8%</td>
</tr>
</tbody>
</table>

Table 1: Integrated motor-compressor design parameters

3.1 Power and Efficiency

The compression required power and rotor stage efficiency based on Equation (13) and (17), when solidity equals to 1.5, the compression required power is about 5kW, and the rotor stage efficiency is 98.28%. The total pressure ratio is 1.088.

Figure 9 shows the variation of rotor stage efficiency according to different solidity. The rotor stage efficiency decreases with the increase of solidity. Figure 10 shows the required compression power to accelerate the flow versus different solidity. The compression power increases with the increase of the solidity. Hence, there is a tradeoff determining the solidity. In this design, as there is a requirement on the blade thickness, solidity is set to be 1.5.

3.2 Effect of Attack Angle

In the previous section, the attack angle is designed to minimize the losses, and the designed attack angle is 10 degrees. In this section, 2-D CFD modeling steady-state blade will be used to investigate the effect of different attack angles. Figure 11 shows the velocity distributions of the flow at multiple attack angles.
4 Electromagnetic Performance of the Proposed Integrated Motor-Compressor

The electromagnetic design is done by modified motor sizing equations; the electromagnetic performance is evaluated by FEA in this section. Figure 16 shows the comparison of proposed integrated flux-switching motor-compressor with a conventional FSPM machine.

When the attack angle is small, the velocity of the flow is low, and therefore the forces on x-direction and y-direction are small, resulting in small lift forces. When the attack angle is large, the flow velocity is large, and there is large flow separation observed. Therefore, when the attack angle is 10 degrees or 11 degrees, high-velocity flow with little separation can be observed.

Figure 12 shows the lift force and drag force in the x-y coordinates. Based on the x-direction and y-direction blade forces obtained from the CFD, lift force and drag force can be calculated. The lift force is calculated in Equation (18), and the drag force is calculated in Equation (19) where $F_x$ and $F_y$ are the blade forces on x-direction and y-direction, $\alpha$ is the attack angle.

\[
L = F_x \cos \alpha - F_y \sin \alpha \\
D = F_x \sin \alpha + F_y \cos \alpha
\]

Based on Equation (18) and (19), the lift force and drag force at different attack angles can be calculated, and they are shown in Figure 13. Based on Figure 13, when attack angle equals to 10 degrees, maximum lift force and minimum drag force can be obtained. After 10-degree attack angle, the lift forces starts to drop and therefore the 10-degree designed attack angle is verified.

As the designed attack angle is verified, a 3-D model is calculated by CFD, the temperature distribution in K and velocity distribution in Ma are shown in Figure 14 and 15. In Figure 14, the stagnation temperature at the outlet of rotor stage is around 310 K while in meanline method, the stagnation temperature at rotor outlet is 315 K. CFD results are more accurate than meanline method as the meanline method cannot account for the blade aspect ratio which is a very important parameter in this design.
compressor with conventional flux-switching PM machine. Figure 17 shows the flux density distribution of the two machines. The flux linkage, back-EMF, and rated torque comparison are shown in Figure 18, Figure 19, and Figure 20, respectively.

In Figure 18 and Figure 19, the flux linkage and back-EMF of the proposed integrated motor-compressor are very close to conventional FSPM machine. In Figure 20, the average rated torque of the proposed machine is around 85% of that of the conventional FSPM machine. However, the torque is still much larger than the compression torque, and the torque ripple is significantly reduced. The mechanical output power from FEA is 7.9 kW, which is larger than the compression required 5kW power. By increasing the electric loading of the proposed machine, it is capable of driving multiple compressor stages.

5 Summary and Conclusion

In this work, the aerodynamic and electromagnetic design of the integrated motor-compressor is investigated. The evaluations of the blade thickness as well as compressor blade solidity are provided. The aerodynamic performance is investigated by meanline method and CFD; it turns out that increasing the solidity will increase the total pressure ratio and compression power, but the rotor stage efficiency will decrease. The effect of attack angle is also evaluated, when the designed attack angle is 10 degrees, maximum lift force can be obtained. The 3-D CFD verified the meanline method calculation and is more accurate. The electromagnetic performance is verified by FEA, the mechanical output power is sufficient to provide compression function and it can be further increased by increasing the electric loading.

6 Acknowledgements

This research has been funded in part by the National Science Foundation (NSF) under CAREER award number 1552942.

I would like to express my special thanks of gratitude to Prof. Nellis who gives me a lot of insight advices in doing this project.

References


Control strategy for a direct driven hydraulics system in the case of a mining loader

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As a response to the strict government emissions regulations, hybridisation of non-road mobile machinery is required. In this paper, behaviour and efficiency of a hybrid mining loader is studied. The full prototype with implemented DDH (Direct Driven Hydraulics) units had been built; however, its performance was unsatisfactory – a large undershoot and steady-state error of 34% persisted. Therefore, a new control strategy was suggested to overcome the issues. Performance of the system was enhanced by applying a fuzzy PID controller. As a result, reference tracking was significantly improved compared to the conventional PID control case and steady-state error of 1% was achieved, while the overall efficiency was kept high in the range of above 50%.

Keywords: fuzzy control, direct driven hydraulics, mining loader, efficiency
Target audience: Mobile Hydraulics, Mining Industry, Control

1 Introduction

The problem of air pollution is becoming more and more severe, and especially non-road mobile machinery (NRMM) contributes to this pollution by emitting considerable amounts of nitrogen oxides and particulate matter into the air [1]. Strict emissions regulations have been proposed [2], and in order to comply with them and decrease energy consumption, the hybridisation of NRMMs is essential. One of the solutions for the hybridisation of such machinery lies in replacing the traditional working hydraulics with a Direct Driven Hydraulics (DDH) setup. In DDH, a hydraulic pump is powered by an electric motor, and it directly controls the actuator [3]. The usage of an electric motor provides shorter response time, which also improves efficiency by reducing losses. Since conventional valve control is not utilized, throttle losses are avoided and heat generation is reduced.

Mining loaders, large NRMMs used in the mining industry, have been the target of such hybridisation. For instance, the conventional hydraulic system of a mining loader was replaced with DDH during the Tubridi project [4], and basic expectations concerning its functionality were met. However, the control of the system was unsatisfactory, as the movements of the machine during its working cycle were insufficiently smooth and the error of 34% of the reference position was present during the EL-Zon project (in the case of 1040 kg payload). Furthermore, it was necessary to retune the controller with each change in the payload. Moreover, there was room for improvement in the overall energy efficiency of the system. In order to overcome these problems, the present study suggested to utilise a different control method. Due to complexity of the system and its high nonlinearities, fuzzy PID control was chosen, as a highly accurate mathematical model is not necessarily needed for designing this type of control. This control technique has already been implemented in a number of hydraulic and electro-hydraulic systems, and it has proven suitable for such application [5 - 7].

Therefore, this study aims to develop new control strategy in order to overcome issues, such as the need of retuning the PID controller with each change of payload and large steady-state error in reference tracking. The study evaluates validity of the proposed fuzzy PID controller using the test setup assembled during the Tubridi project (of which the present study is a continuation), and the results demonstrate improvement in energy efficiency and general behaviour of the system.

This paper is organized in the following way: first, Section 2 briefly introduces the test setup; second, Section 3 explains the basics of fuzzy control and its implementation in this specific case; then, Section 4 illustrates results and finally, Sections 5 and 6 present the discussion and conclusions, respectively.

2 Test setup

This section introduces the full-size mining loader prototype utilised in this experiment. The original working hydraulics of the EJC 90 mining loader (Figure 1) had been replaced with the DDH setup during the Tubridi and EL-Zon projects. The original hydraulic transmission was replaced with a fully electromechanical one. The diesel engine is not mechanically connected to the drive train, which makes its architecture series hybrid. It produces electric power, which is then buffered with a li-ion battery. The DDH setup consists of two separate units. One of them is utilised for boom movement control, while the other handles the bucket. The only difference between the two units is that the boom unit consists of two double-acting cylinders, whereas only one is present in the bucket unit, as can be seen in Figure 2. For more details about the setup, please refer to [6]. The utilised components of DDH units are demonstrated in Figure 2 and Table 1, for more details please refer to [8].

Figure 1: EJC 90 mining loader with dimensions in centimetres and inches

In DDH, the speed of the electric motor defines the flow produced by the hydraulic motors, hereafter referred to as pumps, which then determines the flow to the cylinder chambers. The pumps are connected to the motor shaft via a gearbox. Speed and torque data from the electric motor is utilised by dSPACE control desk software to control the basic movements of the bucket and the boom. The program either receives user input via a joystick (manual mode) and performs the movement or executes a given cycle (automatic mode).
The motors’ acting speed, torque and temperature is fed back to the Sevcon motor controllers. The motors are connected to the hydraulic system as shown in Figure 2. The dSPACE program receives information about cylinder position, temperature and pressure from the sensors and collects data about the hydraulic system. The reference tracking of the cylinder position is accomplished by the fuzzy PID controller, whose working principle is described in detail in the next section.

Figure 2: Models of DDH units of bucket (left) and boom (right)

<table>
<thead>
<tr>
<th>Component</th>
<th>Model</th>
<th>Additional information</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Electric motor</td>
<td>Motenergy ME1304</td>
<td></td>
</tr>
<tr>
<td>2 B-side pump</td>
<td>HYDAC PG100</td>
<td>Boom: PG100-013+011; Bucket: PG100-022x2</td>
</tr>
<tr>
<td>3 A-side pumps</td>
<td>HYDAC PG100</td>
<td>Boom: PG100-008x2 X2; Bucket: PG100-016x2 X2</td>
</tr>
<tr>
<td>4 Pump pressure relief valve</td>
<td>HYDAC DB10P-01</td>
<td></td>
</tr>
<tr>
<td>5 Anti-cavitation valve</td>
<td>HYDAC RV12A-01</td>
<td></td>
</tr>
<tr>
<td>6 Safety valves</td>
<td>HYDAC WS16ZR-01</td>
<td></td>
</tr>
<tr>
<td>7 Hydraulic cylinder</td>
<td>EJC90 original</td>
<td></td>
</tr>
<tr>
<td>8 Motor controller</td>
<td>Sevcon Gen 4 Size 6; DC/AC converter</td>
<td></td>
</tr>
<tr>
<td>9 Lithium-titanate battery</td>
<td>Altairnano 96V</td>
<td></td>
</tr>
</tbody>
</table>

Table 1: Components of bucket and boom DDH units [Courtesy of T. Lehmspalte, Aalto University. 2016]

The system diagram is illustrated in Figure 3. The grey lines on the diagram correspond to the signals sent via a CAN bus. The joystick is connected via the CAN bus to the dSPACE control desk program. For monitoring purposes, the dSPACE program also receives basic information about the battery (its state of charge, voltage, current). The reference signal is then sent via the CAN bus to the Sevcon DC/AC motor controllers, which are powered by the battery. Sevcon motor controllers control the boom and bucket electric motors. The boom electric motor is in speed-control mode, while the bucket electric motor operates in torque-control mode. The information about motors’ acting speed, torque and temperature is fed back to the Sevcon motor controllers. The motors are connected to the hydraulic system as shown in Figure 2. The dSPACE program receives information about cylinder position, temperature and pressure from the sensors and collects data about the hydraulic system. The reference tracking of the cylinder position is accomplished by the fuzzy PID controller, whose working principle is described in detail in the next section.

Figure 3: Block diagram of the system

3 Fuzzy PID control

In this section, the basics of conventional PID and fuzzy PID controllers are discussed.

The well-known discrete-time transfer function of a PID controller is expressed by the following equation:

$$u(k) = K_p e(k) + K_i \sum_{i=1}^{k} e(i) + K_d \Delta e(k)$$

(1)

where $K_p$, $K_i$ and $K_d$ are proportional, integral and derivative gains, respectively, $T_s$ is the sampling period, $u(k)$ is the control signal, while $e(k)$ is the error between the reference and the process output signal; $\Delta e(k) = e(k) - e(k-1)$. These parameters need to be determined, which can be rather challenging, especially in case of a hydraulic system. Inherent excessive nonlinearities and time-variant characteristics are present and, therefore, a control method based on a linear model, such as a PID controller, does not provide adequate results.

In order to improve the behaviour and efficiency of the system, a new self-tuning control strategy was implemented based on the fuzzy algorithm. Due to complexity of the system and its high nonlinearities, fuzzy PID control (Figure 4) was chosen. In this study, the parameters were selected according to the gains of previously used PID controllers and by trial and error testing. The control works as follows: position error (real position subtracted from the reference) and its derivative (velocity) are fed into the fuzzy block. It then calculates the gains of parameters according to predefined rules, which are then sent to the PID controller. The controller is implemented within the MATLAB Simulink real-time model.
The ranges of input variables are \([-100, 100]\) for \(e\) in [mm] and \([-150, 150]\) for \(\dot{e}\) in [mm/s]. These variables represent cylinder position error and its derivative – velocity, respectively. In the process of fuzzification, firstly, it is determined to which degree inputs belong to the fuzzy sets, via linguistic values – membership functions \(\gamma_i\), defined as: \(e, \dot{e} = [\text{NB}, \text{NS}, \text{Z}, \text{PS}, \text{PB}]\). The abbreviations stand for negative big, negative small, zero, positive small and positive big, respectively. Negative big and positive big are trapezoid-shaped functions, while the remainder is triangular (Figure 5). Then, fuzzy operator is applied (minimum as AND method and maximum as OR method) to the rules, which are given in Table 2. After that, the implication method (minimum in this case) is implemented, and the result is a fuzzy set represented by a membership function. Next step is aggregation, the phase in which all the rules are combined in order to make a decision (the aggregation method used is maximum). Finally, defuzzification is applied, and its result is a single number. “Centroid” was chosen as the defuzzification method. There exist five membership functions for fuzzy outputs: \(K_p, K_i, K_d = [\text{VS}, \text{S}, \text{M}, \text{B}, \text{VB}]\), where the abbreviations stand for very small, small, medium, big and very big, respectively. The ranges of the parameters are: \(K_p = [150, 200]\), \(K_i = [1, 10]\) and \(K_d = [10, 20]\) for the boom control and \(K_p = [5, 10]\), \(K_i = [1, 3]\) and \(K_d = [3, 5]\) for the bucket control. The parameter ranges are tuned independently. Table 3 contains the parameters of the membership functions of the controller outputs. Mamdani’s fuzzy inference method was adopted.

![Figure 4: Fuzzy PID controller structure](image)

![Figure 5: Membership functions of the controller input variables – position e and velocity error \(\dot{e}\)](image)

![Figure 6: Control surface of the \(K_p, K_i, K_d\) parameters of the boom fuzzy PID](image)

<table>
<thead>
<tr>
<th>(K_p / K_i / K_d)</th>
<th>(e)</th>
<th>NB</th>
<th>NS</th>
<th>Z</th>
<th>PS</th>
<th>PB</th>
</tr>
</thead>
<tbody>
<tr>
<td>(e)</td>
<td></td>
<td>NB</td>
<td>NS</td>
<td>Z</td>
<td>PS</td>
<td>PB</td>
</tr>
<tr>
<td>NB</td>
<td>VB</td>
<td>VS / M</td>
<td>VB / VS / S</td>
<td>B / S / VS</td>
<td>B / M / S</td>
<td>M / M / M</td>
</tr>
<tr>
<td>NS</td>
<td>VB</td>
<td>VS / M</td>
<td>B / S / S</td>
<td>M / S / S</td>
<td>M / M / S</td>
<td>S / M / M</td>
</tr>
<tr>
<td>Z</td>
<td>B / S / M</td>
<td>M / S / S</td>
<td>M / M / S</td>
<td>M / S / S</td>
<td>S / B / M</td>
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<tr>
<td>PS</td>
<td>B / S / B</td>
<td>M / M / M</td>
<td>S / M / M</td>
<td>S / B / M</td>
<td>VS / VB / B</td>
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<tr>
<td>PB</td>
<td>M / M / VB</td>
<td>S / M / B</td>
<td>S / B / M</td>
<td>VS / VB / B</td>
<td>VS / VB / VB</td>
<td></td>
</tr>
</tbody>
</table>

Table 2: Rule table of the fuzzy inference system

Figure 6 illustrates data from Table 2: it shows control surfaces of the parameters \((K_p, K_i, K_d)\) of the boom fuzzy PID. It can be seen from the table that the control surface of the bucket fuzzy PID parameters has exactly the same shape, since the rules are identical. The ranges of parameters are scaled to match the corresponding data from Table 3.

## 4 Experimental investigation: results

Experimental investigation was performed on the full-size prototype described in section 2. A cycle was utilised to simulate the real operation of a mining loader. In Figure 7 references for boom and bucket position are demonstrated, as well as overall efficiency and regeneration efficiency (example data). The graphs are divided into sections with vertical lines in order to demonstrate the distinct stages of the cycle. (It should be noted that in the graph that represents bucket position, the highest value corresponds to the lowest bucket position and vice versa.) The cycle sequence where first, the (loaded) bucket is lifted to its upmost position (1-3), the boom is then lifted (4-5), next the bucket performs a “dropping” movement while the boom is resting (6-9), then the boom is lowered to the ground (10-11) and, finally, the bucket is lowered to the starting position (12-13). From this figure, it should be noted that highest efficiency level is present during lifting of the boom (4), while power regeneration is present during lowering of the boom (10). The equations utilised for calculating efficiency and regeneration efficiency are...
cylinder velocities were utilised. “Nominal cylinder velocities” in this context refers to the velocities that are defined by the position reference table (input cycle).

The overall system efficiency $\eta$ that was calculated as the ratio of mechanical power $P_{\text{mech}}$ and battery output power $P_{\text{bat}}$:

$$\eta = \frac{P_{\text{mech}}}{P_{\text{bat}}}.$$  \hfill (2)

The regeneration efficiency $\eta_{\text{reg}}$ was defined as ratio of battery and mechanical power (in this case, mechanical acts as input power, while battery power is understood as output, as power generated due to dynamic braking gets stored in the battery):

$$\eta_{\text{reg}} = \frac{P_{\text{bat}}}{P_{\text{mech}}}.$$  \hfill (3)

In the Equations (2) and (3), mechanical power was calculated as:

$$P_{\text{mech}} = F_{\text{boom}} v_{\text{boom}} + F_{\text{bucket}} v_{\text{bucket}},$$  \hfill (4)

where $F_{\text{boom}}$ and $F_{\text{bucket}}$ are forces acting on load pins of the boom and the bucket, respectively, while $v_{\text{boom}}$ and $v_{\text{bucket}}$ are cylinder velocities of the boom and the bucket, respectively. Battery power was calculated as:

$$P_{\text{bat}} = U_{\text{bat}} I_{\text{bat}} + U_{\text{bat}} I_{\text{bucket}},$$  \hfill (5)

where $U_{\text{bat}}$ is battery voltage and $I_{\text{boom}}$ and $I_{\text{bucket}}$ are battery current measured by the boom and bucket motor controller, respectively. Example of data needed for these calculations is shown in Figure 8, from which it can be seen that the load force and current measurements are highly oscillating, therefore oscillations and peaks are present in power and efficiency graphs. During the performed cycle, fluctuation of the current consumed by DDH units comes as a response to the required load. Negative values of current during lowering phase of the boom corresponds to regenerative energy, which is charging the battery (increase in state of charge is present).

It should be noted that even though fuzzy PID has been implemented for both boom and bucket control, the focus of this study was on boom reference tracking. Therefore, only boom position graphs are demonstrated in this paper. However, the efficiency is calculated and shown for the overall system, including both boom and bucket units.

An example of previous unsatisfying results with the conventional PID control is demonstrated in Figure 9.

The fundamental issue with the conventional PID control was a fairly huge undershoot: the boom used to reach only 76 % of the reference, which corresponds to the steady-state error of 34 %. The main reason for such large error was that the controller was tuned for one specific weight of payload. Therefore, it either had to be retuned...
with each change of payload, or the steady-state error in the position reference tracking persisted, which is one of the main reasons why this study was done. The efficiency level for this example case was around 52 %. To overcome the issues existing with the conventional PID controller, fuzzy PID was designed. The experimental results with the proposed fuzzy PID control are presented below in Figure 10 - Figure 13.

Proposed fuzzy PID controller demonstrated excellent self-adaptability with regard to various velocities and payloads (Figure 10 - Figure 13). The steady-state error was improved from 34 % to 1 % (from 88 mm to less than 3 mm), compared to the conventional PID control. In addition, the response time was shortened from 2 s to 0.3 s with proposed fuzzy PID controller. It can be seen from the position error graphs that the error caused by response time increases with cylinder velocity, but is almost identical for different payload cases. Furthermore, the overall system efficiency reaches the level of above 50 % (50 % - 60 % for cases of no load to half-load); while it is slightly lower in full load condition and varies in ranges 44 % - 50 % depending on velocity. It is significantly higher compared to conventional Load sensing (LS) systems, efficiency of which varies in the range of only 10 % to 20 % /10/, /11/. It is noted that the power efficiency does not greatly depend on cylinder velocity. Moreover, as it was previously explained, during lowering motion of the boom, power regenerates. Regeneration efficiency is highest with full load (level of 48 % with normal cylinder velocity and 38 - 46 % in cases of other velocities); it decreases slightly with half-load (37 - 42 %), while it is insignificant in cases with no load. It can be concluded that regeneration efficiency depends on the payload, rather than cylinder velocity.

5 Discussion

This paper is dealing with the hybrid mining loader with zonal hydraulics realized with DDH. DDH is a compact hydraulic unit with low oil volume, short pipelines and reduced potential leakage points, which brings lower environmental hazards. In addition, regenerative energy of DDH brings advantages compared to conventional systems. This leads to extension of driving range and/or working time of electric driven vehicles. However, DDH itself requires a precise selection of hydraulic components to match hydraulic cylinder dimensions, and this brings along challenges with position control as system response directly depends on the electric motor.

In this research, reference tracking of the existing system was improved by means of automatisation of the process of choosing PID controller parameters and keeping high efficiency of the system. For this purpose, a fuzzy PID controller was designed and implemented. In this paper, the positioning control in different loading conditions were implemented and verified experimentally with DDH mining loader, and thus, the goal of having a fuzzy PID controller independent from payload weight and velocity variation was reached.

However, fuzzy control also has some downsides. Its main disadvantage is that there is no general method (yet) of selecting parameters of membership functions for inputs and outputs or for defining the rules. Therefore, designing fuzzy control for each system requires excessive experimenting and testing. However, once the parameters and rules have been determined, it is fairly simple and straightforward to implement this type of control into a pre-existing model.
Despite demonstrated excellent performance and high efficiency, it is wise to note that a DDH unit comes with the price of extra electric components, such as an expensive battery, electric motors and controllers. In the future, dimensioning of components (due to unknown duty cycle), limited market of suitable and available electric components, as well as charging method and aging of battery should be considered. Regarding future research on the topic, position control of the bucket should be improved in order to increase power efficiency during the whole cycle.

6 Summary and conclusion

In order to investigate behaviour and efficiency of hybrid NRMM, a full-size prototype of a mining loader was built during the Tubridi and EL-Zon projects. The aim of this study was to improve reference tracking of the existing boom DDH system, while automating the process of choosing controller parameters and keeping the efficiency of the system as high as possible. For that purpose, a fuzzy PID controller was designed and implemented. The behaviour of the system was put to the test with three different payloads and with four distinct velocities. It is obvious that the reference tracking was significantly improved compared to the previous case when only a simple PID control was used – the steady-state error is noticeably reduced (from 34 % to around 1 %), and the response time is shortened (from 2 s to 0.3 s). Furthermore, the overall system efficiency reaches the level of above 50 % (50 % - 60 % for cases of no load to half-load); while it is slightly lower in full load condition (44 % - 50 %).

7 Acknowledgements

This research was enabled by the financial support of Tekes, the Finnish Funding Agency for Technology and Innovation (project EL-Zon) and internal funding at the Department of Mechanical Engineering at Aalto University.

8 Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>e</td>
<td>Position error</td>
<td>[mm]</td>
</tr>
<tr>
<td>\dot{e}</td>
<td>Derivative of the position error</td>
<td>[mm/s]</td>
</tr>
<tr>
<td>\epsilon(k)</td>
<td>Error between the reference and the process output signal</td>
<td>[-]</td>
</tr>
<tr>
<td>F_{boom}</td>
<td>Force acting on load pins of the boom</td>
<td>[N]</td>
</tr>
<tr>
<td>F_{bucket}</td>
<td>Force acting on load pins of the bucket</td>
<td>[N]</td>
</tr>
<tr>
<td>I_{boom}</td>
<td>Battery output current that is used by the boom motor controller</td>
<td>[A]</td>
</tr>
<tr>
<td>I_{bucket}</td>
<td>Battery output current that is used by the bucket motor controller</td>
<td>[A]</td>
</tr>
<tr>
<td>K_p</td>
<td>Proportional gain</td>
<td>[-]</td>
</tr>
<tr>
<td>K_i</td>
<td>Integral gain</td>
<td>[-]</td>
</tr>
<tr>
<td>K_d</td>
<td>Derivative gain</td>
<td>[-]</td>
</tr>
<tr>
<td>U_{bat}</td>
<td>Battery output voltage</td>
<td>[V]</td>
</tr>
<tr>
<td>\omega_{boom}</td>
<td>Velocity of the boom cylinder</td>
<td>[m/s]</td>
</tr>
<tr>
<td>\omega_{bucket}</td>
<td>Velocity of the bucket cylinder</td>
<td>[m/s]</td>
</tr>
<tr>
<td>\eta</td>
<td>Overall system efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>\eta_{reg}</td>
<td>Regeneration efficiency</td>
<td>[-]</td>
</tr>
</tbody>
</table>

References

/3/ Häminnen, H., Minav, T., Pietola, M., Replacing a constant pressure valve controlled system with a pump controlled system, In: Proceedings of the 2016 Bath/ASME Symposium on Fluid Power and Motion Control, FPMC2016, Bath, United Kingdom, Sep 7-9, 2016.
Combining Control and Monitoring in Mobile Machines: the Case of an Hydraulic Crane

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The widespread use of electro-hydraulic (EH) technology of the last decades has led to important improvements in the control features, safety and performance of hydraulic machines. However, limited work exploited the use of the EH control features for condition monitoring. This paper proposes a neural network based diagnostic algorithm, that takes advantage of the parameters of a controller developed for the case of an independent metering hydraulic system. The reference application is a truck loading crane available at the authors’ research center. The results show how the proposed methodology is effective to detect faults (the faults considered pertain to the pump, the metering valves and the cylinder), with a limited number of sensors.

Keywords: Control, Diagnostic, Independent Metering, Hydraulic Cranes, Mobile Hydraulics

Target audience: Mobile Hydraulics, System Diagnostic, Control

1 Introduction

The widespread use of electro-hydraulic (EH) technology of the last decades has led to important improvements in the control features, safety and the performance mobile applications. The hardware (electronic control unit, sensors) which is used for control functions can be exploited to implement strategies for Prognostics and Health Management (PHM), which would bring important benefits, such as increased reliability and safety, optimal condition based maintenance of the hydraulic systems.

Therefore, the integration of EH control with diagnostics/prognostics techniques can make the EH technology more appealing even for hydraulic applications traditionally reluctant to the adoption of electronic control. The integration of diagnostic and prognostic features in EH systems also enables condition based maintenance strategies, which is more effective than the classic time-based maintenance.

Researchers in the area of diagnostics have proposed several approaches, which can be divided into two main categories: model based approaches and data driven methods (Figure 1). As the words suggests, a model based approach relies on the development of a mathematical model of the system representing its health status. The comparison between the output of the model and those of the real system leads to the residual analysis, which gives a diagnosis of the system health. Good examples of this methodology can be found in [1-2]. On the other hand, data driven approaches are the most popular in research because they don’t require the derivation of a mathematical model of the machine, which is always a challenging task for complete EH systems. Data driven approaches requires a first offline step, in which a proper identification of the system features allows for the training of a classifier. Once the classifier is trained, the data driven algorithm can run online on the machine giving the diagnosis of the system through the classifier. Although a data-driven method does not require the derivation of a mathematical model of the system, it strongly relies on the availability of experimental data. This might represent an important drawback, because it might be difficult to produce experimental data in faulty conditions for training purposes. Good examples of data-driven approaches applied to hydraulic systems can be found in [3-4].

This paper contributes in the topic of combining EH control strategies with PHM, and proposes a solution for the diagnostic of the entire hydraulic system, enabling the fault detection of the main hydraulic components. The novelty of the proposed approach is that the diagnostic algorithm is not only based on sensors installed on the machines, but it also uses controller parameters as additional indicators of the health status of the machine.

Figure 1: Block diagrams of the model based approach and the data-driven approach for diagnostic algorithms.

The reference machine used to demonstrate the potential of the proposed approach is a hydraulic crane for truck applications instrumented at the authors’ research center. The hydraulic system chosen for proving the effectiveness of the proposed diagnostic method is an EH independent metering system, which is different from the open-center system that equips the commercial version of the reference machine. Independent metering systems are known for presenting control challenges, and for this reason they are particularly suitable for the proposed method which combines control features with conventional sensors for diagnostic purposes.

In [5], the authors formulated and experimentally validate a numerical model of the reference machine. A control strategy for the independent metering systems was also synthesized and the simulation results proved the effectiveness of the control technique. The controller was model based, and the uncertainties of the model parameter where handled through a PI controller. The parameters of the PI were found by means of an optimization procedure. The value of the cost functions used for the optimization were also found to be correlated to the health condition of the machine. This past work put the basis of the present effort, which proposes a diagnostic algorithm to evaluate the health status of the system. Thus, the diagnostic algorithm constitutes the element of novelty of this work. The proposed diagnostic method uses as inputs not only information coming from the sensors installed on the machine, but also information coming from the cost function used to define the controller parameters. The input parameters are then used within a neural network, which is able to provide outputs indicative of the faults. As selected faults considered for the analysis, the case of pump degradation, cylinder degradation and selected valve faults will be considered. After a proper training of the neural network, examples of significant results are shown.

2 Reference Machine

The reference machine considered in this work is a truck loading crane with a maximum capacity of 5 t, based on the commercial product Atlas 125.1 A4 (Atlas Maschinen GmbH 1996), shown in Figure 2. This machine is installed on a concrete base at the Maha Fluid Power Research Center of Purdue University. The crane arm is operated through 4 hydraulic actuators: the swing, the main boom, the outer boom and the telescopic actuator. In
this study, the proposed diagnostic strategy was applied to the outer boom actuator, whose movements involves both resistive as well as overrunning load phases.

Figure 2: Reference machine, ATLAS 125.1 A4, installed at the Maha Fluid Power Research Center

The hydraulic circuit of the reference machine is based on a post-compensated Load Sensing (LS) system, implemented through cartridge independent metering valves provided to the authors by Hydraforce. Compensators are present at each meter-in control section, and permit an easy control of the velocity of each actuator, while meter-out sections are used to counter-react overrunning loads. A simplified schematic of the system is shown in Figure 3. In more details, the hydraulic circuit can be divided into 4 main sections, the flow supply unit, which is essentially a fixed displacement pump (a gear pump of 19 cm³/rev), an unloading valve that permits to pressurize the pump to the level of the LS signal, the independent metering valves manifold and the actuator. All the actuators present in the reference machines are connected in parallel downstream the unloading valve; however, for the sake of clarity Figure 3 shows only the reference actuator. The hydraulic circuit is controlled by the Data Acquisition System (DAQ) NI cRIO-9030 equipped with only pressure sensors (of clarity Figure 3 shows only the reference actuator. The hydraulic circuit is controlled by the Data Acquisition System (DAQ) NI cRIO-9030 equipped with only pressure sensors (of clarity Figure 3 shows only the reference actuator). The hydraulic circuit is controlled by the Data Acquisition System (DAQ) NI cRIO-9030 equipped with only pressure sensors (of clarity Figure 3 shows only the reference actuator).

A complete numerical model of the machine is used, implemented with the AMESim commercial software. This model has been derived and experimentally validated in [5]. Within this work, the same model will be used to validate the proposed diagnostic algorithm, as described in the following sections.

3 Control Strategy

The controller used for the independent metering system of Figure 3 is presented in this section. The controller constitutes an essential part of the diagnostic algorithm that will be described in section 4. For this reason, although the derivation of the controller was the main subject of [5], for completeness this section describes the essential elements of the controller. However, the reader can refer to [5] for more details on the controller and its experimental validation.

3.1 Control Synthesis

The EH post compensated LS independent metering architecture of the hydraulic circuit of Figure 3 requires a proper controller to guarantee a proper and safe functioning of the machine. As widely explained in literature [5], at least two different cases need to be considered: resistive and overrunning loads.

In the case of resistive loads, the meter-out valve can be fully-opened to reduce energy consumption of the machine, and the controller can set the opening area of the meter-in orifice that works along with the compensator to eliminate possible interference with other actuators. For the case of overrunning loads, a more complex control strategy is required. The working principle of the controller can be easily explained from the simplified schematic of Figure 4, where only the extension case is considered in the case of overrunning load. The retraction case can be derived with a similar procedure.

Being the system based on LS compensators (Figure 3), the pressure drop Δp across the meter-in valve (valve 1) can be in first approximation assumed constant. For a simplistic model of the system, the compensator can be excluded, as shown in Figure 4.

The force applied at the cylinder, can be derived from the force balance at the cylinder in Equation (1):

\[ F_{cyl} = p_B \cdot A - p_A \cdot A + m_{eq} \cdot x_{cyl} \]

(1)

Where \( A \) and \( a \) are respectively the piston area and the rod side cylinder area, \( m_{eq} \) is the equivalent mass of the rod, \( x_{cyl} \) is the linear acceleration of the cylinder, \( p_A \) and \( p_B \) are respectively the pressure in the piston chamber and in the rod chamber of the cylinder and \( F_{cyl} \) is the force acting on the cylinder.

From the force balance at the cylinder, a reference pressure \( p_{Aref} \) can be calculated at the rod chamber to exactly balance the load, allowing the pressure in the piston chamber to reach zero-gauge pressure. A reference pressure \( p_{Aref} \) needs to be introduced to avoid cavitation in the piston chamber, before calculating \( p_{Aref} \) with Equation (2).

\[ p_{Aref} = p_B + \frac{A}{A_p} (p_{Aref} - p_A) \]

(2)

As mentioned before, the actuator velocity is set from the meter-in portion of the system, according to the LS principle, therefore Equation (3) needs to be always satisfied.

\[ \dot{x}_{cyl} = \frac{Q_{in}}{A} = \frac{Q_{ref}}{A} \]

(3)

Afterwards, considering the two orifice equations, the meter-out command can be easily derived. Its control law is described in Equation (4).
\[ u_{\text{out}}(t) = A_{\text{in}}^{-1} \left( \frac{\alpha}{A} \frac{\Delta p}{p_{\text{ref}} - p_r} A_{\text{out}}(u_{\text{in}}(t)) \right) \]  

Where \( A_{\text{in}}(u) \) is the area function of the meter-in valve and \( A_{\text{out}}^{-1}(u) \) is the inverse area function of the meter-out valve.

\[ CF_2 = \int \max(p_{\text{ref}} - p_{\text{out}}, 0) \]

The value of the first term of the cost function \((CF_1)\) represents how much the pressure in the piston chamber is below a reference pressure \(p_{\text{ref}}\), (a pressure level above atmospheric, such that cavitation is avoided with a certain margin).

The value of the second term of the cost function \((CF_2)\) represents how much the pressure in the cylinder (connected to the meter-in) is higher than the reference value \(p_{\text{ref}}\). Therefore, this term is created to reduce the energy consumption of the machine, having the goal to minimize it.

The weights \(\alpha_1\) and \(\alpha_2\) are needed to give a priority to the cost function related to the cavitation phenomena since if cavitation occurs in these kind of machines, safety needs to be guaranteed.

The value of the cost functions will be afterwards used for the diagnostic algorithm proposed in section 4.

4 Diagnostic Algorithm

The diagnostic algorithm is based on a neural network (NN) approach (Figure 6), defined in Equation (8).

\[ y_j = f \left( \sum_p w_p x_i \right) \]

Where \(y_j\) represent the \( j \) output of the neural network, \(x_i\) represent the \( i \) input of the neural network and \(w_p\) represents the neurons of the neural network.

The advantage of the NN approach is given by the fact that the system is considered as a black box; after a proper training the NN will be able to detect the faults in the hydraulic system.

The output \(y_j\) of the NN are defined as faults of the hydraulic system, and in this work four faults were considered, leading to four outputs of the neural network. The choice of the faults considered to prove the effectiveness of the procedure was arbitrary, although realistic fault instances were considered. In particular, the faults for the hydraulic pump and the hydraulic cylinder were considered, along with possible malfunctioning of the meter-in and meter-out valve. The valve faults are considered representative of a spool blockage instance, where the spool is not able to fully open due to impurities in the fluid or other sources of mechanical obstructions to the motion of the spool.

Each fault was simulated by changing the corresponding parameter, as shown in Table 1.

<table>
<thead>
<tr>
<th>Fault Description</th>
<th>Healthy</th>
<th>Component Failure</th>
<th>Neural Network Output</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volumetric efficiency [%]</td>
<td>0.95</td>
<td>0.55</td>
<td>0 – 1</td>
</tr>
<tr>
<td>Cylinder friction [N]</td>
<td>10000</td>
<td>0</td>
<td>0 – 1</td>
</tr>
<tr>
<td>Meter-in/Meter-out limiter [-]</td>
<td>1</td>
<td>0</td>
<td>0 – 1</td>
</tr>
</tbody>
</table>

Table 1: Parameter considered for the representation of the fault in the reference machine.

The pump fault in this work pertains to its volumetric efficiency: a 0.95 volumetric efficiency was considered for the healthy case, while a value of 0.55 indicates the component failure. The values in between 0.95 and 0.55 represent intermediate conditions in which the algorithm must be able to detect the health status (or percentage of fault) of the component. The same approach could be extended for the case of hydromechanical efficiency, although this aspect was not considered in the present study.

A similar approach was used for the case of cylinder fault. In this case the fault parameter considered is the internal friction of the cylinder. Internal cylinder leakages, not considered in this work, could represent a possible extension. The meter-in and meter-out valve fault, where artificially simulated by limiting the input command to
The 11th International Fluid Power Conference, IFP, March 19-21, 2018, Aachen, Germany

proposed in this work. The considered unfolding consists in two phases (Figure 7); the first one is the main boom extension or unfolding until the main boom reaches an angle $\phi_1$; the second phase represents the outer boom unfolding, until the outer boom reaches the angle $\phi_2$.

Figure 7: Case study.

For this work, only the second phase (outer boom extension) was considered due to a more interesting behaviour from the controller point of view. In fact, this phase presents both resistive and overrunning load conditions. An example of the acquired signal from the pressure sensors on the machine are shown in Figure 8.

The mean of the signals is then calculated and the value is fed to the neural network.

Figure 8: Example of the acquired pressure signal during a healthy operating condition of the machine (blue line) and a faulty condition of the meter-out valve as example (red line).

5.2 Neural Network Training

As shown in Figure 1, the data-driven method requires a first phase of training off-line. A total of 85 training cases are considered. 5 cases of healthy conditions, while for each fault a total of 20 cases are considered, with an increase value from 0 to 1 and step size of 0.05. A white-gaussian noise is added to simulate more realistic
conditions where other variables (environment temperature variations, sensor noise, etc.) might affect the sensor outputs.

The results from the training are shown in Figure 9. Each plot represents a single output $y_j$ from the neural network. On the horizontal axis is represented the case number, starting from a healthy condition, each fault is being analysed separately. The first twenty cases represent a linear degradation of the pump. The following cases represents the meter-in (MI) fault, followed by the cylinder fault and the meter-out (MO) fault. In blue is represented the target value, therefore the real fault value affecting the component, and in red there is the output obtained from the neural network.

To evaluate the performances of the proposed approach, the mean-squared error between the neural network output and the ideal value $i_j$ has been used in Equation (10).

$$\text{MSE}_j = \frac{1}{N} \sum_{i=0}^{N-1} (i_j - y_j)^2$$

The mean-squared error of the training phase is shown in Table 2.

<table>
<thead>
<tr>
<th></th>
<th>Pump Fault</th>
<th>Meter-in Fault</th>
<th>Cylinder Fault</th>
<th>Meter-out Fault</th>
</tr>
</thead>
<tbody>
<tr>
<td>MSE</td>
<td>2 \cdot 10^{-4}</td>
<td>0.002</td>
<td>7 \cdot 10^{-4}</td>
<td>0.009</td>
</tr>
</tbody>
</table>

Table 2: Mean-squared error of the training case.

5.3 Neural Network Validation
To validate the neural network and to simulate an online case, a different set of cases was considered. In this validation set, the fault percentage is 0, 0.25, 0.5 and 1. This set of faults is reduced in order to test the ability of the network to recognize the correct percentage of fault after being trained. Also in this case, each fault parameter values was artificially modified with the addition of white gaussian noise, in order to not compare the same exact simulation and to simulate a real application. The results are shown in Figure 10.

For the validation case, a total of 34 cases are evaluated, in which the target values are coupled and spared from 0 to 1, with the addition of gaussian noise. From the results, it can be noticed that the cylinder fault, the pump fault and the meter-in fault match the target value quite well. This is also confirmed from the mean-squared error calculation shown in Table 3.

<table>
<thead>
<tr>
<th></th>
<th>Pump Fault</th>
<th>Meter-in Fault</th>
<th>Cylinder Fault</th>
<th>Meter-out Fault</th>
</tr>
</thead>
<tbody>
<tr>
<td>MSE</td>
<td>6 \cdot 10^{-4}</td>
<td>0.004</td>
<td>0.001</td>
<td>0.015</td>
</tr>
</tbody>
</table>

Table 3: Mean-squared error of the validation case.

The detection of the meter-out fault is instead the one that has more issues, although the mean-squared error evaluation could be acceptable. This problem of the neural network for detecting the meter-out fault comes from the particular case study considered in this research. This is also confirmed in Figure 11, which shows the input fed to the neural network, for two reference faults: the meter-out one (not properly detected) and the pump one (one of the faults properly detected). It can be noticed that with an increasing pump fault level, the feature related to the pump pressure fed to the neural network sees a linear decrease. On the other hand, when the meter-out fault occurs, the pump pressure feature has a jump at the value of 0.6 as meter-out (MO) fault. This derives from the
fact that a small intensity of the fault is not affecting the system. This is mainly due to the meter-out control. When the fault has a small intensity, there is almost no effect in the acquired signals. Only when the intensity of the fault is high enough to affect the meter-out control performances, the effect of that is a big increase of the system pressure, due to the valve characteristic. Operating the machine with small loads, a small change in the input creates a big change in the pressure drop across the valve, coming from the area characteristic of the valve. This is seen also in Figure 8, where there is a big discrepancy between the healthy condition and the faulty condition in the pressures. This discrepancy is present in all the input selected for the neural network, therefore the neural network is not able to separate all the level of faults of the meter-out valve. This explains the problem of the neural network detecting the meter-out fault condition. In this case study, due to the jump of the input features to the network is not able to separate all the level of faults of the meter-out valve. This explains the problem of the neural network training algorithm to recognize the trends in the available input signals in case of faulty conditions.

The proposed method can be considered as a valuable diagnostic algorithm for mobile equipment, where a power unit (power supply), a regulation unit (proportional valves) and a user unit (hydraulic cylinders or motors) are always present, leading to a diagnosis of the main components of the hydraulic system.

7 Acknowledgements

The authors would like to thank the National Fluid Power Association (NFPA) and the Center for Compact and Efficient Fluid Power (CCEFP) for funding the project from which this work derives (grant 16MO2). The authors would also like to thank Hydraforce for having provided the independent metering valve block and for the support in the development of its numerical model and Siemens for the use of the LMS AMESim software.

6 Summary and Conclusion

This paper presents a new diagnostic algorithm able to detect faults in a EH system, using information coming from the sensors, and added information coming from the controller, such as the control input and the value of the cost functions. The reference case is a hydraulic crane, and the unfolding operation is considered as reference for the diagnostic evaluations. The proposed algorithm is synthesized considering only the outer boom actuator, which present both resistive and overrunning conditions; additionally, only certain faults (pump volumetric efficiency degradation, cylinder friction, metering valve openings) were considered. However, the proposed approach could be extended to more general cases with a proper modification of the topology of the neural network used to estimate the faults. Only three pressure sensors were utilized for the diagnostic function; two of these sensors are necessary for the proper control of the independent metering system. In this way, the use of more expensive sensors (such as flow meters or position sensors) is avoided. This is an important result achieved thanks to the capability of the neural network training algorithm to recognize the trends in the available input signals in case of faulty conditions.

Results from a detailed model of the machine shows good performance of the network. Limitation of this approach are related to the difficult prediction of the meter-out valve health condition, which cannot be predicted with a proper resolution by the neural network. While the other faults can be predicted with good accuracy at all cases, the meter-out valve fault is predicted only according to a pass/fail fashion.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>EH</td>
<td>Electro-hydraulic</td>
<td>[-]</td>
</tr>
<tr>
<td>PHM</td>
<td>Prognostic and Health Management</td>
<td>[-]</td>
</tr>
<tr>
<td>LS</td>
<td>Load-sensing</td>
<td>[-]</td>
</tr>
<tr>
<td>LSPC</td>
<td>Load-sensing post compensated</td>
<td>[-]</td>
</tr>
<tr>
<td>MI</td>
<td>Meter-in</td>
<td>[-]</td>
</tr>
<tr>
<td>MO</td>
<td>Meter-out</td>
<td>[-]</td>
</tr>
<tr>
<td>( F_{cyt} )</td>
<td>Force acting on the hydraulic cylinder</td>
<td>[N]</td>
</tr>
<tr>
<td>( p_{in} )</td>
<td>Respectively pressure at the bore side and at the rod side</td>
<td>[bar]</td>
</tr>
<tr>
<td>( p_{out} )</td>
<td>Respectively reference pressure at the bore side and at the rod side</td>
<td>[bar]</td>
</tr>
<tr>
<td>( \dot{x}<em>{cyt},\ddot{x}</em>{cyt} )</td>
<td>Respectively linear velocity and acceleration of the hydraulic cylinder</td>
<td>[m/s-m/s^2]</td>
</tr>
<tr>
<td>( Q_{in},Q_{out} )</td>
<td>Entering and leaving flow at the hydraulic cylinder ports</td>
<td>[m^3/s]</td>
</tr>
<tr>
<td>( A_{in}(x),A_{out}(x) )</td>
<td>Respectively meter-in and meter-out orifice area</td>
<td>[m^2]</td>
</tr>
<tr>
<td>( p_p )</td>
<td>Pump outlet pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>( p_T )</td>
<td>Tank pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>( u_{in},u_{out} )</td>
<td>Meter-in and meter-out command</td>
<td>[-]</td>
</tr>
<tr>
<td>( CF )</td>
<td>Cost function</td>
<td>[-]</td>
</tr>
<tr>
<td>( \alpha_1,\alpha_2 )</td>
<td>Cost function weights</td>
<td>[-]</td>
</tr>
<tr>
<td>NN</td>
<td>Neural network</td>
<td>[-]</td>
</tr>
<tr>
<td>( x_i )</td>
<td>i - Neural network input</td>
<td>[-]</td>
</tr>
<tr>
<td>( y_j )</td>
<td>j - neural network output</td>
<td>[-]</td>
</tr>
<tr>
<td>( w_k )</td>
<td>k – neural network neuron</td>
<td>[-]</td>
</tr>
<tr>
<td>( u_{in},u_{out} )</td>
<td>Respectively meter-in and meter-out command</td>
<td>[-]</td>
</tr>
</tbody>
</table>
References


Fault-tolerance Operation for Independent Metering Control Valve

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This paper focuses on the faulty issues of the independent metering valve (IMV) in mobile applications. First, typical faults are studied in a 2t excavator to analyze their negative influences. The model of the abnormal system is estimated according to the results of fault detection and diagnosis. Accordingly, a fault-tolerance controller is designed to reconfigure normal controller by the coordinate control of other parallel available valves. With the presented fault operation, the dynamic characteristic under reconfigured modes can strictly match with that of system simulation. Simulations are conducted in the excavator to verify the fault-tolerance controller.

Keywords: Independent metering valve (IMV), fault detection and diagnosis, fault tolerance control (FTC), excavator, safety and reliability

Target audience: Mobile Hydraulics

1 Introduction

Independent metering valve (IMV) is a promising hydraulic control valve compared with the conventional proportional controlled directional valve (PDV). By breaking the mechanical coupling between inlet and outlet, IMVs are usually applied to mobile hydraulic systems to improve the energy efficiency, adaptivity and controllability. In spite of these advantages, IMVs are still not widely applied in practical mobile machineries. Both Prof. Munserhoff (RWTH Aachen University) and Prof. Weber (Dresden University of Technology) consider that one of the main obstacles which prohibits it from industrial applications is the issue including reliability and safety [1-2].

Figure 1 exhibits the system hardware layouts using PDV and IMV. Because more than one valve are controlled simultaneously for one actuator, the system is unable to operate normally if any one valve encounters failures, such as spool jammed, electromagnet failure, etc.. The more valves arranged in the hydraulic system, the higher fault rate for the system will occur. Furthermore, to take advantages of IMV, many functions used to be attained by the hydro-mechanical way, such as flow or pressure control and load sensing, are moved into software with additional valves and electronic sensors. The reputations of electronics and sensors are less reliable than purely hydro-mechanical devices [3-4]. However, mobile systems have tougher safety and reliability requirements than industrial systems since an operator often experiences more hazardous conditions with the absence of complete separation from danger [5]. Therefore, it is urgent to deal with the faulty issues of IMV.

Previous works on the fault operation of IMV have concentrated on the digital hydraulic valve system [6-10]. The system is consisted of four digital flow control unit (DFCU), of which each DFCU contains five parallel connected 2-way on/off-type valves such that the combinations of on/off state for each valve determine the flow through DFCU [11]. The unique layout of DFCU extends the system with fault-tolerance functions because if part of on/off valves encounters with faults, other parallel connected valves can still work to compensate the performance degradations. However, general IMV layouts still employ four proportional control valves and the fault tolerance operation of DFCU might be not applicable to such situation.

Although the fault operation capability of individual proportional controlled valve is weaker than DFCU, when it is used in the independent metering control system, the additional multi-valve parallel layout and electronic feedbacks beyond the convention system still make it possible to deal with faults. First, the electronic sensors make the system easier to apperceive the self-status for faulty detection and diagnosis. Second, extra valves can be involved in the control system to weaken the adverse effects caused by faults. Therefore, this study focuses on the faults of valve components in IMV and aims to design a fault-tolerance control method to enhance the reliability and assure the safety in mobile applications. A special mobile application — excavator is employed in this study.

2 Studied System

A 2 t excavator using the independent metering control system is presented in Figure 2. It was presented thoroughly in [12] and [13]. A hydraulic-mechanical coupled simulation model is based on this test rig and this model has been proved that it has the ability to capture both the dynamic and static properties precisely [14]. The IMV block in this study is different to the valve block utilized in [12]. Four 2-way proportional control valves replace the dual directional valve such that the independent metering control layout has stronger universality. In order to distinguish these valves, they are indentified by “V1, V2, V3, V4” as depicted in Figure 2. The pump displacement is regulated with an electronic load sensing way, of which the pressure margin between supply and load pressures is 1.6 MPa.

Figure 2. Studied independent metering control system in excavator
3 Fault Analysis

Hydraulic valves may encounter with the problems including jammed spool, disabled electromagnet, wire disconnection, leakage, etc. Any faults in one valve will make the actuators disable to track the desired trajectory. Different failures may cause the same abnormalities. For example, both wire disconnection, disabled electromagnet and spool jammed closed will make the valve unable to open. Both valve jammed open and valve failure in a large opening displacement mean that there are large unexpected flow across this faulty valve. Therefore, the features of valve faults can be divided according to the the equivalent spool locations when the fault happens, as depicted in Table 1.

<table>
<thead>
<tr>
<th>Abnormal state</th>
<th>Equivalent spool location</th>
<th>Fault types</th>
</tr>
</thead>
<tbody>
<tr>
<td>Valve unexpected closed</td>
<td>(0, s0)</td>
<td>Case 1</td>
</tr>
<tr>
<td>Valve unexpected open</td>
<td>1</td>
<td>Case 2</td>
</tr>
<tr>
<td>Valve failure in an intermediate</td>
<td>Large opening 1</td>
<td>Case 3</td>
</tr>
<tr>
<td>displacement</td>
<td>Small opening (s0,1)</td>
<td></td>
</tr>
<tr>
<td>Valve leakage</td>
<td>(s0,1)</td>
<td>Case 4</td>
</tr>
</tbody>
</table>

Table 1. Possible fault situation of IMV (s0 represents the deadzone of spool).

Considering the reliability and safety in mobile applications, the study focuses on adverse effects of these faults on the excavator. Although there are four valve abnormal states, different locations of these faults (which valve is failure) will result in distinct faulty issues in actuators. Besides, system performances are greatly determined by the operating modes in the independent metering control system. Therefore, the influences of faults depend on not only the faulty location and type (faulty features in Table 1), but also the operating mode. Operation modes using IMV can be divided into three types according to the flow supplies: normal mode (Nor.), regeneration mode (Reg.) and float mode (F0) [12][13]. The target of mode switching is to obtain a high possible efficiency according to the system state variables. Figure 3 depicts the mode switching logic in terms of the load characteristics.

Considering both the valve faulty information and system operating modes, possible faulty issues are simulated and corresponding fault knowledges are analyzed by the boom upwards and downwards motions, as shown in Figure 4. According to Figure 3, when boom lifts upwards, the load locates in the quadrant I and the normal mode is selected. When boom lowers downwards, the load locates in the quadrant II and the float mode is selected. The influences of each faults are given as follows:

**Remark:** a. In the figure of velocity:

- - - - v (Normal)  - - - - v (Faulty)  — — x1 (Normal)  — — x2 (Faulty)

b. In the figure of pressure:

- - - - - p1 (Normal)  - - - - - p1 (Faulty)  — — — — — — — — — — — — p4 (Normal)  — — — — — — — — — — — — p4 (Faulty)

(1) Faulty Issue 1: Boom upward (Normal mode) - stoppage (deteriorated motion performances)

(2) Faulty Issue 2: Boom upward (Normal mode) - moving slowly (deteriorated motion performances)
(3) Faulty Issue 3: Boom upward (Normal mode) - reversing (dangerous behaviors)

<table>
<thead>
<tr>
<th>Fault</th>
<th>Velocity</th>
<th>Pressure</th>
<th>Knowledge</th>
</tr>
</thead>
<tbody>
<tr>
<td>V3-Case 2</td>
<td><img src="image" alt="Graph of V3-Case 2" /></td>
<td><img src="image" alt="Graph of V3-Case 2" /></td>
<td>Both $p_a$ and $p_b$ decrease sharply. $p_a \approx p_b$.</td>
</tr>
<tr>
<td>V4-Case 2</td>
<td><img src="image" alt="Graph of V4-Case 2" /></td>
<td><img src="image" alt="Graph of V4-Case 2" /></td>
<td>Both $p_a$ and $p_b$ decrease sharply. $p_a \approx p_b$.</td>
</tr>
</tbody>
</table>

(4) Faulty Issue 4: Boom upward (Normal mode) - pressure rising fast (deteriorated energy-saving performances)

<table>
<thead>
<tr>
<th>Fault</th>
<th>Velocity</th>
<th>Pressure</th>
<th>Knowledge</th>
</tr>
</thead>
<tbody>
<tr>
<td>V2-Case 1</td>
<td><img src="image" alt="Graph of V2-Case 1" /></td>
<td><img src="image" alt="Graph of V2-Case 1" /></td>
<td>$(p_a, p_b) \approx (0, 0)$. $p_b$ is close to $p_a$ and much larger than $p_b$.</td>
</tr>
<tr>
<td>V2-Case 2</td>
<td><img src="image" alt="Graph of V2-Case 2" /></td>
<td><img src="image" alt="Graph of V2-Case 2" /></td>
<td>$(p_a, p_b) \approx (0, 0)$. $p_a$ decreases and is much larger than $p_b$, obviously lower than $p_a$.</td>
</tr>
</tbody>
</table>

(5) Faulty Issue 5: Boom downward (Float mode) - dropping fast (dangerous behaviors)

<table>
<thead>
<tr>
<th>Fault</th>
<th>Velocity</th>
<th>Pressure</th>
<th>Knowledge</th>
</tr>
</thead>
<tbody>
<tr>
<td>V3-Case 2</td>
<td><img src="image" alt="Graph of V3-Case 2" /></td>
<td><img src="image" alt="Graph of V3-Case 2" /></td>
<td>$p_a \approx p_c$. $p_a$, $p_b$, and $p_c$ gradually decrease to $p_c$.</td>
</tr>
</tbody>
</table>

Summary:

1. It is seen that only by the normal controller, the independent metering control system will confront with various negative issues including deteriorated performances, incorrect motions or even dangerous behaviors when one of valves encounters faults. In this case, fault-tolerance control is required to take over the normal controller. Safety should be guaranteed at first and then system performances such as trajectory tracking and energy efficiency are expected to improve during faulty conditions.

2. The same faulty issue may be caused by different valve faulty types. For example, if the actuator continues to move slowly whatever the valve opening increases (Faulty Issue 2), there exit four possible types of valve faults. On the other hand, the same valve fault may cause different faulty issues. For example, if valve V2 encounters the failure in a small intermediate position, the actuator will move slowly (Faulty Issue 2) and simultaneously the system pressure will rise quickly to relief pressure (Faulty Issue 4), such that both trajectory tracking and energy saving performances are deteriorated.

Therefore, it is confused to distinguish the internal fault feature only by the external faulty issues. Fault detection and diagnose are necessary before the fault-tolerance control.

4 Pressure-Based Fault Detection and Diagnose

Due to the diversity of valve faults, the first target of fault-tolerance control is to identify each fault by the system states. The information of each fault contains not only faulty locations (which valve is failure), but also faulty types. At first, the following two assumptions are given:

1. All sensors are in normal conditions;
2. There is only one valve encountering a fault.

To monitor the behavior of a valve, the most commonly used methods are measurements of coil current and spool displacement. However, there exist some occasions lack of these feedbacks in mobile machinery. Meanwhile, pressure sensors quite often already exist in the independent metering control system, so pressure signals are used for fault detections.

The model-based method has been widely used in fault detection and diagnose [10]. However, the model of hydraulic system contains time-varying, nonlinear and uncertain characteristics, such that the accurate model is difficult to obtain. Complex intelligent control approaches, such as neural networks, have been used in many applications for fault detections [15]. In this paper, as the knowledge of faults have been obtained, a fault diagnose process is similar with a fault tree establishment [16]. Therefore, in terms of the faults knowledge demonstrated in Section 3, the fault-tree-based fault diagnose expert system is employed based on the pressure signals.

![Figure 5 Fault trees for faulty issues 1 and 2](image)
tree, the faulty issues analyzed in Section 3 are selected as the analysis target (top events). Then all of the possible independent factors or factor combinations (fault knowledge) that can lead directly to the top event happening are listed. At last, the fault detection system further analyzes failure cause and deduces step by step following this procedure until all basic valve faults are found out. Figure 5 shows the fault trees for faulty issues 1 and 2.

5 Fault-Tolerance Operation with Reconfigured Controller

In terms of the faulty information identified by the fault detection and diagnosis, the adversely effects caused by the fault can be quantified and the model of the faulty system is estimated to prepare for the fault-tolerance control. For a hydraulic actuator, the flow and pressure dynamics can be obtained from Equations (1) to (5):

\[ q_i = (\pm) q_1 + (\pm) q_2 \]  
\[ q_i = (\pm) q_1 + (\pm) q_2 \]  
\[ q_i = (\pm) q_1 + (\pm) q_2 \]  
\[ q_i = q_i + q_2 \]  
\[ q_i = K_q a_i A_q \sqrt{p_i} \]

where the positive and negative symbols represent the flow charged into or discharged from each cylinder port respectively. A value of zero represents that the corresponding valve is abnormal or not active with closed state.

With respect to a random fault, the system dynamic is changed by the way that one of the valve flows is abnormal compared with normal conditions. The flow equations for typical faults in Section 3 are listed as follows:

<table>
<thead>
<tr>
<th>Fault types</th>
<th>Normal conditions</th>
<th>Faulty conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boom upward-stoppage - V4, case 1</td>
<td>( q_1 = q_1 )</td>
<td>( q_3 = q_3 ), ( q_3 = 0 )</td>
</tr>
<tr>
<td>Boom upward-slow - V1, case 3</td>
<td>( q_i = q_i )</td>
<td>( q_2 = q_2 ), ( q_2 = 0 )</td>
</tr>
<tr>
<td>Boom upward-slow - V3, case 4</td>
<td>( q_i = q_i )</td>
<td>( q_1 = q_1 ), ( q_1 = 0 )</td>
</tr>
<tr>
<td>Boom upward-reverse - V3, case 3</td>
<td>( q_i = q_i )</td>
<td>( q_2 = q_2 ), ( q_2 = 0 )</td>
</tr>
<tr>
<td>Boom upward-pressure rising - V2, case 3</td>
<td>( q_i = q_i )</td>
<td>( q_1 = q_1 ), ( q_1 = 0 )</td>
</tr>
<tr>
<td>Boom downward-dropping - V3, case 2</td>
<td>( q_i = q_i )</td>
<td>( q_1 = q_1 ), ( q_1 = 0 )</td>
</tr>
</tbody>
</table>

Table 2: Equations for different fault situations (a represents a constant small spool displacement, b represents a constant large or maximum spool displacement)

The basic idea of the fault tolerance control is to compensate or even eliminate the abnormal flow such that the adverse effect on the system dynamic can be weakened. It can be achieved owing to the multi-valve parallel layout using independent metering control techniques. If one of valves is failure, the control strategy in the normal controller should be reconfigured by other available and standby valves. In order to accurately compensate adverse effects of the faulty valve, the model of the faulty system is estimated according to Table 2, such that the dynamic characteristic under reconfigured modes can strictly match with that of faulty system.

Based on the fault detection results, there are three fault tolerance approaches to address corresponding fault cases.

5.1 Reconfiguring valve control command

It is suitable for the faults caused by unexpected flow leakages, such as bypass flow leakages caused by opening of V3 in faulty issues 2 and 3. The fast dropping caused by abnormal opening of V3 also employs this method. The unexpected flows are compensated by the parallel valve V1, and of course the reference valve flow and supply flow from pump must be enlarged, as shown in Figures 6 and 7. In Figure 6, the flow equation in this faulty system is given in terms of Table 2.

\[ q_i = q_i - q_1, \quad q_1 = q_1 \]

In the fault tolerance operation, the modified reference flow across V1 (\( q_{fa,ref} \)) should be given by adding a compensation flow \( q_c \).

\[ q_{fa,ref} = q_{fa,ref} + q_c \]

In order to precisely match the leakage flow, the required compensation flow \( q_c \) should be equal to the unexpected flow across V3 (\( q_3 \)). Thus, the fault magnitude of the valve should be first estimated. Because valves V1 and V2 work normally, their spool displacements can be considered to strictly track the given control commands when ignoring the valve response time. As a consequence, the flows across V1 and V2 can be given as:

\[ q_i = K_q a_i A_q \sqrt{p_i} \]

In steady states, the inlet and outlet flows have the following rough relationship by omitting other leakages:

\[ q_i = \frac{A_{sa}}{A_{so}} q_{a,ref} \]

Then, the compensation flow can be estimated by Equations (6) to (10):

\[ q_i = K_q a_i A_q \sqrt{p_i} - q_{fa,ref} - \frac{A_{sa}}{A_{so}} K_q a_i A_q \sqrt{p_i} \]

In Figure 7, the valve V1 and pump are not active in the normal controller. For compensation the fault, they are activated to supply more flows to prevent cylinder from fast dropping. The abnormal flow across V3 is given in terms of Table 2.

\[ q_i = K_q a_i A_q \sqrt{p_i} \]
Due to the uncertainty of failure position, the flow across valve V3 is unable to be estimated by Equation (8), but it can be captured indirectly by outlet flow in terms of Equations (9) and (10) as:

\[ q_3 = \frac{A_1}{A_4} K_b (\bar{u}_{v_{in,ref}} - p_4 - p_b) \sqrt{p_s - p_b} \]

The compensation flow should be equal to the excess flow beyond the reference one caused by the unexpected large opening:

\[ q_i = \frac{A_1}{A_4} K_b (\bar{u}_{v_{in,ref}} - p_4 - p_b) \sqrt{p_s - p_b} - V_{cbr} \]

5.2 Reconfiguring valve controller

In some occasions, although the faulty valve is unable to regulate anymore due to the failure in an intermediate displacement such as the jammed spool, the inlet flows still cross this valve to drive the cylinder. The faulty event usually happens because the flow is out of control in this abnormal valve. Therefore, the flow control function can be switched to the outlet valve such that the “meter-in control” is changed to “meter-out control”. By this reconfiguration of valve controller, the cylinder is able to track the reference velocity again. Taking the fault of V1-case 3 during boom lifting for instance, as shown in Figure 8, the original control command is given as:

\[ x_{v1,c} = \frac{v_{ref}}{A_1} \frac{1}{\sqrt{p_s - p_b}} \frac{1}{K_b} (p_b - p_s) \]

By the fault tolerance control, the control commands are regiven to the valve V2 and pump when omitting the leakages in the two components. In this case, the system also has the capability to track the desired motion and the meter-in valve is only considered as a fixed orifice.

\[ x_{v2,c} = \frac{v_{ref}}{A_1} \frac{1}{\sqrt{p_s - p_b}} \frac{1}{K_b} (p_b - p_s) \]

\[ u_b = v_{ref} K_b \frac{A_4}{A_1} \bar{u}_{v_{in,ref}} \]

5.3 Reconfiguring operating mode

If one valve is disable in the closed state, then it blocks the required flows into or out from the cylinder. What’s more, this faulty valve could not be used any more unless it is replaced offline by a new one. Online fault tolerance control can still make the system working, but with deteriorated energy-saving performances. It requires a reconfiguration of operating modes. As shown in Figure 9, when V2 is faulty to close, the cylinder must be stopped. Then un-activated valve V4 opens and the operating mode is changed from normal one to regeneration one. By this mode switching, the cylinder continues to move and track the reference velocity under the fault of V2, but the systempressure will increase due to the differential connection of two cylinder ports.

6 Verification of Fault Tolerance Control

Figure 10 exhibits the results of fault tolerance control for Figure 6 (Boom upward slow – V3, case 3). With the presented flow compensation approach, the velocity and supply pressure are close to those in the normal condition. Oscillations occur when the fault tolerance controller is introduced into the system, and the oscillations are attenuated quickly to steady state. It is seen that supply flow and valve opening of V1 are both enlarged to compensate the unexpected leakage flow across V3. Figure 11 exhibits the results of fault tolerance control for Figure 7 (Boom dropping fast – V3, case 2). With the presented flow compensation approach, the system no longer drops fast and the velocity is close to the normal condition. It is achieved by the activation of V1 and the pump. It is seen that flows supplied by pump again and cross V1 into the piston chamber to prevent the cylinder from dropping.
The cylinder moves quickly again compared with the faulty condition. abnormal position of V1 is too small to restrict the flow rate. In spite of this, the adverse effect is weakened the present ed

Figure 12 exhibits the results of fault tolerance control for

The 11th International Fluid Power Conference, 11. IFK, March 19-21, 2018, Aachen, Germany

$q_s/\text{L·min}^{-1}$

$v/\text{mm·s}^{-1}$

Figure 10. The simulation results by FTC for Figure 6

(a) Cylinder velocity              (b) Supply pressure

Figure 11. The simulation results by FTC for Figure 7

(c) Cylinder velocity              (d) Supply pressure

Figure 12. The simulation results by FTC for Figure 8

(a) Cylinder velocity              (b) Supply pressure

(c) Supply flow

(d) Valve V1 opening

7 Conclusion and Future work

A fault-tolerance operation system for IMV is presented parallel with the normal controller. Fault tree analysis is employed to establish a fault diagnosis algorithm based on the pressure signals, which has the capability to search the causes of faults online. Then, the paper designs a fault-tolerance controller to reconfigure original control modes by other available and standby valves. The simulation on the excavator demonstrates that under different faults of valves, the system can continue tasks with the proposed fault-tolerance controller, and the safety can be guaranteed with little degradation in motion and energy-saving performances.

Future work will concentrate on the verification of the fault-tolerance operation system by multi-actuator movements. Also, some loading conditions such as digging and cutting by the excavator should be considered.

8 Acknowledgements

This work was supported by the National Natural Science Foundation of China (Grant No.51705152 and No.U1509204), Jiangxi Provincial Natural Science Foundation (Grant No.20161BAB216133)

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_h$</td>
<td>Head side area of cylinder</td>
<td>$[\text{m}^2]$</td>
</tr>
<tr>
<td>$A_r$</td>
<td>Rod side area of cylinder</td>
<td>$[\text{m}^2]$</td>
</tr>
<tr>
<td>$K_p$</td>
<td>Proportion coefficient</td>
<td>$1$</td>
</tr>
<tr>
<td>$K_v$</td>
<td>Orifice flow conductivity coefficient</td>
<td>$[\text{m}^3/(\text{s·Pa}^{1/2})]$</td>
</tr>
<tr>
<td>$n_o$</td>
<td>Rotational speed of pump</td>
<td>$[\text{r/min}]$</td>
</tr>
<tr>
<td>$p_s$</td>
<td>Pressure in head side chamber</td>
<td>$[\text{Pa}]$</td>
</tr>
<tr>
<td>$p_{r_s}$</td>
<td>Pressure in rod side chamber</td>
<td>$[\text{Pa}]$</td>
</tr>
<tr>
<td>$p_{r_p}$</td>
<td>Pressure margin between pump and load</td>
<td>$[\text{Pa}]$</td>
</tr>
<tr>
<td>$p_d$</td>
<td>Drain line pressure</td>
<td>$[\text{Pa}]$</td>
</tr>
<tr>
<td>$p_p$</td>
<td>Pump supply pressure</td>
<td>$[\text{Pa}]$</td>
</tr>
</tbody>
</table>
\[ q_a \] Head side chamber flow of cylinder [m^3/s]
\[ q_b \] Rod side chamber flow of cylinder [m^3/s]
\[ q_c \] Compensation flow by fault-tolerance control [m^3/s]
\[ q_i \] Flows for different valve, i=1,2,3,4 [m^3/s]
\[ q_s \] Pump supply flow [m^3/s]
\[ u_i \] Control signals for different valve, i=1,2,3,4 [v]
\[ u_{i,max} \] Maximum control signals for different valve, i=1,2,3,4 [v]
\[ u_p \] Control signal for pump [v]
\[ x_{i,j} \] Spool displacement for different valve, i=1,2,3,4 [m]
\[ v \] Cylinder velocity [m/s]
\[ V_a \] Chamber volume of cylinder head side [m^3/r]
\[ V_b \] Chamber volume of cylinder rod side [m^3/r]
\[ V_p \] Pump displacement [m^3]
\[ \beta_e \] Oil elasticity modulus [Pa]

References

Efficiency studies on double pump supply units

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In this paper three concepts of double pump supply units are presented and compared to a conventional variable displacement pump as reference. These supply units consist of two off-the-shelf pumps in a parallel arrangement and they are meant to perform like a continuously variable source of flow rate. In order to evaluate possible energy savings of the supply units, their efficiency characteristics are firstly computed in a steady-state simulation but also examined on a test bench. By means of a semi-synthetic load profile for an exemplary application, the annual savings of the systems are calculated in comparison to the reference pump. Moreover, a rating system for the systems’ complexity is shown and applied to the three concepts in order to judge the trade-off between efficiency and complexity. The studies show that the more complex concepts provide higher saving potentials than simpler systems, but the interdependence may come unpredictably in some cases.

Keywords: double pump, efficiency, auxiliary pump, boost

Target audience: Mobile Hydraulics

1 Introduction

Depending on the range of operation, the hydraulic power demand in mobile machines may heavily vary from application to application. In well-equipped machines the hydraulic power supply is carried out by variable displacement pumps which are dimensioned for peak pressure and/or flow rate in order to cover a wide range of different working profiles. Since this peak power is required rarely, the pump operates at poor efficiency quite often. Hence, a double pump approach is examined, in which two smaller pumps replace one conventional (large) variable displacement pump. In this paper, three different concepts of double pump systems for an exemplary application are shown. The concepts are evaluated with regard to their complexity by a rating system, which is presented briefly. In the next step, the concepts are both simulated and built up on a test rig in order to estimate their efficiency characteristics. Finally the complexity and the savings of the exemplary application are contrasted with each other to show, which concept offers the best cost-benefit ratio.

2 Exemplary application

As exemplary application, a tractor of the mid-range segment with 100 kW is chosen, because those machines operate in manifold operating cases with appreciable hydraulic power demand. Furthermore, such machines are often equipped with Load-Sensing-controlled pumps. Since the tractor is only used as example to get an idea of a realistic load profile, no real measured data is used within the project, but a semi-synthetic load profile is created. This load profile is based on open data [1, 2, 3]. The time slice of each operating point has a great impact on the overall energy consumption and should be carefully evaluated when enhancing the system’s efficiency. The relative time slices of the considered operating points are shown in Figure 1. It can be seen, that the importance (derived from the time slices) of the operating points are heavily unequally distributed in the field. Approximately 45 % of the operating time, the hydraulic system is in standby mode, near the coordinate origin. About 15 % of the time, the system works at its maximum pressure, but with small flow rates. The maximum flow rate (> 90 %) is needed in only 11 % of the operating time.

Tractors in the performance class of 100 kW may work up to 1200 hours per year. In the studies, the annual operating time is defined to 1000 hours per year. The common pressure level of agricultural machines amounts to approximately 200 bar. Load-sensing pumps need to apply an additional control pressure difference of 15 – 30 bar. The maximum flow rate of those tractors is between 50 and 170 l/min [4]. Hence, the maximum flow rate in this project is limited to 90 l/min. Thus, the system has a hydraulic peak power of 30 kW.

3 Conceptual design

The central approach is to split up the displacement of a (large) variable pump into two (smaller) units. By this, the displacement, which is not needed temporary, can be deactivated by switching off one of the pumps in a parallel arrangement. An increase of the working pressure by serial connection of those two pumps is not considered, since in the field of mobile machines there are already high-pressure pumps available to cope with the demand. In [5] it is described, that by means of combining different components a high number of different concepts and architectures is conceivable. However, the focus will be on the three concepts, which are shown below.

Concept 1

The first system consists of two variable displacement pumps, which are driven by the same shaft (Figure 2). The shaft can be split by a clutch, thus the secondary pump can be deactivated mechanically and there are no drag losses in the pump. When the primary pump is fully swivelled out, the clutch engages and activates the secondary pump. A check valve inhibits leakage across the secondary pump when it is deactivated. The displacement of the pumps can be distributed freely on the two pumps. To ensure that the primary pump stays at full displacement, its control pressure difference is increased by several bars towards the secondary pump control pressure difference.
4 Rating system for complexity

To evaluate the costs and technical effort of the concepts, a rating system was developed. The rating system takes costs for components, packaging, assembly, maintenance and interfaces into account. The result of the rating is a quality factor, which allows a relative evaluation of the costs compared to the functional benefits.

4.1 Approach for the rating system

The rating scheme is divided into two parts. The first one evaluates the complexity of a component itself. The second part considers the peripheral devices and, if necessary, control technologies. The modelling of both parts is based on the power level of the components, which is defined by the maximal required flow and effort. Flow and effort. Here, both flow and effort are divided into eight classes.

4.2 Complexity factors for the concepts

The concepts, which were presented in chapter 3, are rated with the rating system. In the following, the results of the rating system will be shown. Since the displacements of the pumps in the concepts are not defined, there will be a result for the maximum size of the primary pump, the maximum size of the secondary pump and for the components, which were used at the test bench. Furthermore, the complexity of the reference system, which consists of a single variable pump and its control, will be shown as comparison.

4.2.1 Complexity of concept 1

The complexity factors for the first concept and the reference system can be seen in Figure 4. As expected, the reference system with only one variable displacement pump leads to the lowest effort with a complexity factor of 2610. The concept with the largest secondary variable pump (concept 1, max.) has a higher factor than the same concept with a smaller secondary pump. The reason for this is, that the clutch, which connects the secondary pump with the driving shaft, has to transmit a higher torque if the pump is larger. Thus, the clutch has to be dimensioned more robust, too. According to the Figure, the version with a small secondary pump (concept 1, min.) would lead to the lowest complexity. However, in this case the efficiency of the primary pump in operating points of a small flow rate would not be much better than the efficiency of the reference pump. On the test
The focus of the project is efficiency studies on the different supply systems. Therefore a simulation model was built up with LMS Amesim. As criterion the efficiency of the whole supply unit and the power losses are used. The simulation model is based on generic efficiency diagrams of the single components. The diagrams are not based on measured data of the used machines, but they are derived from real characteristic diagrams of different pumps of the same type, which are normalized. Because the characteristics do only take the losses of the rotation kit into account, but not the leakage of the pump controller, those losses were calculated by a pressure-dependent approach of Kögl /6/.

The assumed characteristic diagram of the reference pump is depicted in Figure 7 at a constant speed of 2200 rpm. It represents the characteristics of an axial piston pump in swash plate design. At 85-100% of the peak power, there is a plateau with small gradients in flow rate and pressure. With smaller flow rate and smaller pressure, the gradients increase.

5 Simulation

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In concept 3, the secondary pump is a constant pump and its maximum displacement is limited to 50 % of the total displacement of both pumps. Since a constant pump’s design is less sophisticated (e.g. gear pumps) and they do not need a controller, the complexity of all variants is smaller than the complexity of the other both concepts, see Figure 6. On the test bench, the concept with the maximum displacement of the constant pump was realised. Hence, there is no difference between the two bars on the right hand side. The complexity factor is $\Phi_{\text{test bench}} = 3268$. With regard to the reference system, the complexity of the test bench system of the first concept is 44 % higher, the second concept 31 % higher and of the third concept it is only 12 % higher.

In Figure 6 the same diagram for the supply units of concept 1-3 are shown, but the shaft speed of the pumps is different from the speed in Figure 7 due to the fact that each supply unit has to be able to deliver the same maximum flow. The displacements of the simulated pumps, which are also used at the test bench, can be seen in

Figure 4: Complexity factor for different versions of concept 1

Figure 5: Complexity factor for different versions of concept 2

Figure 6: Complexity factor for different versions of concept 3

Figure 7: Simulated efficiency of the reference pump at speed of $n = 2200$ rpm
Table 1. For all concepts a common characteristic of a double pump supply unit can be seen, because there is an obvious decline in the efficiency when the maximum flow rate of the primary pump is reached.

In the first concept, the coupling point is reached at 65 l/min. For smaller flow rates, the efficiency characteristics are typical for a swash plate pump because only the primary pump is on duty. When the secondary pump is activated, the efficiency decreases about 6-9 percentage points, but increases with higher flow rates up to 89 % again. Compared to the reference pump (in the same operating points), the first concept has a higher efficiency. For small power levels (<3 kW), the power losses can be reduced by 5 kW, which is a saving of 42 percentage points. The savings are decreasing with higher flow rate and smaller pressure, at the coupling point the losses are reduced by 2 kW, which are approximately 10 percentage points. For higher flow rates, when both pumps are working, the difference in the power losses of the reference system and the system of concept 1 is very small, but the double pump system still has a higher efficiency than the reference system.

In comparison to the first concept, the efficiencies of the second concept are lower at small flow rates, because the secondary pump produces drag losses when it is not active. After reaching the coupling point, the efficiencies are the same. Because of the drag losses, the savings of concept 2 compared to the reference pump reach a maximum of only 20 percentage points at small flow rates. At the coupling point, there are savings of 10 percentage points again and for higher flow rates, the savings conform to the savings of concept 1.

On the right hand side in Figure 8 the efficiency diagram for concept 3 is shown. Since the speed of the pump is bigger, since the efficiency of the secondary pump, which is modelled as a gear pump, is smaller than the efficiency of the corresponding piston pump of concept 1 and 2. At small flow rates, there are significant savings between 2 and 4 kW, which are again decreasing with higher flow rate and lower pressure. If both pumps are working, there are only very small savings at very low pressures. At higher pressures, the concept has even higher losses than the reference system, so it should not be used, if the peak power of the system is required very often.

<table>
<thead>
<tr>
<th>Concept</th>
<th>Pump</th>
<th>Number (Figure 9)</th>
<th>Max. displacement in cm³</th>
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<tr>
<td>Reference</td>
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<td>71</td>
</tr>
<tr>
<td>1</td>
<td>Primary</td>
<td>P3</td>
<td>23</td>
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<td>Secondary</td>
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<td>3</td>
<td>Primary</td>
<td>P3</td>
<td>23</td>
</tr>
<tr>
<td>4</td>
<td>Secondary</td>
<td>P4</td>
<td>20</td>
</tr>
</tbody>
</table>

Table 1: Displacement of the pumps, which were simulated and used at the test bench

6 Bench tests

To investigate the double pump supply units and to validate the simulation results, a test bench was built up. The test bench consists of three variable piston pumps in swash plate design and an internal gear pump, which are connected to one common hydraulic line. Since the available pumps do not offer a drive through, a power shift gear box is used in order to (de-)activate the pumps dynamically. The simplified structure of the test bench is shown in Figure 9. This contribution focusses on the results of the measurement and not on the test bench and the experimental procedure.
therefore at poor efficiency. However, at higher flow rates (> 90 l/min), the disadvantages of the double pump system may probably become smaller, because the efficiency of the used secondary pump will increase very fast when displacement is increasing.

In concept 3 (Figure 10, right), the efficiency characteristic with the drop in the middle of the diagram is less distinct than in the simulation (Figure 8). One reason for this is, that due to the smaller speed, the coupling point is located at a flow rate of approximately 45 l/min, but there was no measurement at this level. In the lower diagram, the difference of the power losses of the reference pump and the double pump system (concept 3) show, that the new system reduces the losses only at small flow rates if only the primary pump is performing. At high requirement which demands the secondary pump to work as well, the reference pump works more efficient.

**7 Results**

To evaluate the three double pump concepts, it is necessary to merge the efficiency studies, the load profile of the exemplary application and the complexity rating. Firstly, the energy losses \( E_p \) of the system are calculated out of the power losses \( P_e \) considering the above mentioned annual working period \( T_a = 1000 \text{ h} \) according to Equation (4).

\[
E_p(Ap, Q) = \int P_e(Ap, Q) dT_a \cdot T_a = P_e(Ap, Q) \cdot \frac{\int (Ap, Q) - T_a}{\sum Q^2 Ap^2 Q} - T_a
\]  

(4)

If the energy losses are normalized with the losses of the reference system, the benefit (savings) and the costs (complexity) can be contrasted. The result are shown in Figure 11. It is obvious, that the concepts with clutches have significant smaller power losses than the reference system or concept 2. The reason for this is that the operating points at low small rates occur much more often than points with high flow rates (see Figure 1). This is advantageous for all double pump systems, because the smaller primary pumps work with a higher efficiency at those flow rates than the reference pump. However, in concept 2 this advantage is reduced by the secondary pump, which is connected to the drive shaft permanently which generates drag losses. With only 2 % savings, the second concept has no notable effect on the efficiency, but it is 30 % more complex than the reference system. The other double pump systems benefit from the clutch, since they are driven in operating points with small flow rates very often. The secondary pump is only used occasionally to boost additional flow rate.

The big saving potential of concept 1 of more than 60 % is outstanding, but the complexity of the system is very high, because two variable pumps and an externally operated clutch are needed. A good balance between complexity and energy savings can be realised with the third concept. The savings amount to 30 %, while the system is only 12 % more complex than the reference system.

**Figure 11: Normalized complexity and energy losses of the double pump supply units compared to the reference pump**

As a result of the project it can be recorded, that those double pump supply units, which offer dynamical engaging of the secondary pumps, may decrease the energy losses of applications with heterogeneous load profiles significantly. The increased complexity of the system has to be evaluated individually for every system and its load profile. A method to do so was shortly presented in chapter 4.

**8 Summary and Conclusion**

Since hydraulic systems of mobile machines have to provide a certain corner power, which is rarely requested, those systems work at poor efficiency for a huge amount of time. Hence, in this paper three different concepts for double pump systems of an exemplary application of a tractor are discussed and evaluated. The concepts are rated with regard to their complexity by an approach, which takes into account both, the needed components and the necessary peripheral devices into account. To evaluate the possible energy savings, a simulation model was built up and the results are shown. It can be derived, that the double pump systems generate less power losses than the reference system with a single variable displacement pump. To validate the results, a test bench was built up and the measurements are presented briefly. The savings of the measurement are a little smaller than the simulated savings, in some operating points the double pump systems are even more inefficient than the reference system. On the other hand the qualitative behaviour of the simulation can be confirmed by the measurements. Finally, the savings for the exemplary application and the complexity are contrasted with each other. This comparison shows, that (for the investigated concepts) more complex systems lead to higher saving potentials and that one has to evaluate in each case, which benefit justifies which effort.
9 Acknowledgements

The content of this contribution is mainly based on the research project “Effiziente Versorgungseinheit für mobilhydraulische Systeme”, FKM-Nr. 703270. The authors would like to thank the Research Fund for Fluid Power of the VDMA and the belonging companies for supporting the project.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( a_f )</td>
<td>Factor for the asymptote in Gompertz-function</td>
<td>[-]</td>
</tr>
<tr>
<td>( b_f )</td>
<td>Base complexity of technology</td>
<td>[-]</td>
</tr>
<tr>
<td>( c_f )</td>
<td>Growth rate in Gompertz-function</td>
<td>[-]</td>
</tr>
<tr>
<td>( e_{pot} )</td>
<td>Effort class</td>
<td>[-]</td>
</tr>
<tr>
<td>( e_{str} )</td>
<td>Flow class</td>
<td>[-]</td>
</tr>
<tr>
<td>( E_{v} )</td>
<td>Energy losses</td>
<td>[J]</td>
</tr>
<tr>
<td>( k_{pot} )</td>
<td>Exponent for effort</td>
<td>[-]</td>
</tr>
<tr>
<td>( k_{str} )</td>
<td>Exponent for flow</td>
<td>[-]</td>
</tr>
<tr>
<td>( h_{tech} )</td>
<td>Factor for technology</td>
<td>[-]</td>
</tr>
<tr>
<td>( P_{v} )</td>
<td>Power losses</td>
<td>[W]</td>
</tr>
<tr>
<td>( Q )</td>
<td>Flow rate</td>
<td>[l/min]</td>
</tr>
<tr>
<td>( t )</td>
<td>Time</td>
<td>[s]</td>
</tr>
<tr>
<td>( T_{a} )</td>
<td>Annual working time</td>
<td>[h]</td>
</tr>
<tr>
<td>( \Delta p )</td>
<td>Pressure difference</td>
<td>[bar]</td>
</tr>
<tr>
<td>( \Phi_i )</td>
<td>Complexity factor</td>
<td>[-]</td>
</tr>
<tr>
<td>( \Phi_K )</td>
<td>Component factor</td>
<td>[-]</td>
</tr>
<tr>
<td>( \Phi_s )</td>
<td>Interface factor</td>
<td>[-]</td>
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</tbody>
</table>

References

Hydraulic excavator is widely used in the construction field, due to their small size to power ratio and big actuation forces. However, due to large throttling loss and gravitational potential wasting, its energy efficiency is very low, which is even lower than 10%. This paper aims to improve the energy efficiency of the hydraulic excavator by reducing throttling loss and regenerating potential energy directly based on a novel pump controlled system. The system under consideration utilizes a newly designed asymmetric pump which has three ports, the two are connected to the hydraulic cylinder, the other is connected to an accumulator. Thus, this system can regenerate the potential energy directly and can match the unequal flow rates of the single rod cylinder basically. Furthermore, working performances of the excavator boom system with the asymmetric pump and independent metering circuit are studied comparatively. Results show that, compared with an independent metering circuit, the electric power consumption during the boom going up can be reduced by 56%.

**Keywords:** Hydraulic excavator, high energy efficiency, asymmetric pump controlled, energy recovery

**Target audience:** Mobile Hydraulics, Mining Industry, Design Process

1 Introduction

Nowadays, hydraulic excavator is widely used in the construction field, and there are millions in use in the world. However, the energy efficiency of the whole excavator is poor, which mainly caused by the low thermal efficiency of the fuel engine[22], large throttling loss[3-4] and kinetic energy and potential energy loss of the working device[5]. Hence, increasing the energy efficiency of excavators has become a hot topic for the manufacturers and researchers.

In terms of engine energy efficiency increasing, a conventional approach is to adjust the engine speed according to the load condition[19]. The other is the diesel engine cylinder deactivation technology[50]. However, these two approaches can only improve the economy of the diesel engine in light load. An approach to comprehensively improve the engine efficiency is to add an auxiliary power unit to the system, called the hybrid technology[11-15]. Furthermore, a relatively thorough approach is to use some accumulators to separate the engine from the actuators, allowing optimal engine operation independent of the current power demand[11-17]. And also, the engine can be replaced with an electric motor to eliminate emissions completely[30].

Conventional valve controlled systems usually feature low energy efficiency due to the inherent throttling losses and the simultaneous actuation of the valves’ meter-in and meter-out control edges. Therefore, some theses attempt to use an independent metering system to replace the four-side valves to deal with this problem. And this is also one of the research hotspots in valve controlled systems[17-30]. And also, to the mobile machine, a single pump was often used to drive more than one actuators which work under different pressure level, thus large throttling loss was needed to control the cylinders velocities. Thus, the throttling loss cannot be avoided in the valve controlled system. Therefore, there are many years’ efforts to develop hydraulic systems without throttling losses. Pump controlled systems also called displacement controlled system, can eliminate throttling loss completely and have proven themselves in practice for a long time[19-26]. However, once the linear drives are to be used in the form of differential cylinders, it becomes difficult to connect the asymmetric flow rates of the consumer with the balanced volume flow of the pump.

For hydraulic excavator, usually its boom weight is heavier than the load. When the boom moves downwards, it does not need to supply power to drive the boom cylinder down, and it needs to balance the gravity force of the working device by controlling the pressure in the cylinder rodless chamber. During this process, the potential energy of the working device is consumed on the control valve and converted into heat. Therefore, it is necessary to recover the potential energy to reduce the fuel consumption. As a result, much research has been conducted in the field of energy saving mobile hydraulics. And current research can be divided into electrical recovery and hydraulic recovery according to their storage style. For the electrical recovery solutions, an electric generator/motor is used to recover the potential energy and maybe an accumulator was used to decrease the installed power of the electric generator/motor by decoupling the boom lowering and power generation process. And then, the stored electric energy can be used to drive the main pump during another process[21-24]. For the hydraulic recovery, a hydraulic motor/pump is used to control the cylinder velocity and an accumulator is used to store the energy. And then, during another process, the hydraulic motor can be used for auxiliary driving the main pump[20-27]. And also, there is another way to control the cylinder with a hydraulic transformer[28]. However, many energy conversion links will cut down the energy recuperation efficiency, and the installed power is large. That is to say, the energy recovery efficiency is not high enough and many devices need to be added to the hydraulic excavator, which makes it complicated and expensive.

In this paper, an innovative pump controlled architecture with a newly designed asymmetric pump which can efficiently recover the potential energy of the working device is put forward. Innovative research and major contribution of this article are that the high energy efficiency and simple reusing can be achieved with only the newly designed asymmetric pump without long energy conversion links and complex recovery circuit.

This paper is organized as follows. Firstly, the system structure and working principle of the single rod cylinder controlled by asymmetric pumps is given in Section II. In Section III, experiments are carried out to prove the performances of the system, and the energy efficiency of the new system is compared with a separate metering in and separate metering out system. A conclusion is given in Section V.

2 Asymmetric pump controlled single rod cylinder system

Recently, a single pump controlled single rod cylinder in the study region is one of the hot topics in pump controlled systems due to its simple structure, low costs, and big output force. In these systems, there exist unequal flow rates at two ports of the cylinder due to its asymmetric structure. And when a conventional pump is used to control this type cylinder, either a deficient or excess flow rate is always formed in the closed circuit.

Hence researches have been focused on the compensation method of the unequal flow rates and on improving the stability of the system. This paper proposes a novel compensation solution based on a newly designed asymmetric pump and the system energy efficiency can be improved without much control efforts and additional element.

2.1 System structure

Our team has introduced the principle of asymmetric valve controlled asymmetric cylinder system to the pump controlled single rod cylinder system based on a newly designed asymmetric pump. The principle of asymmetric pump controlled single rod cylinder is given in Fig.1. A servo motor is used to regulate the drive speed of the newly designed asymmetric fixed displacement pump. The ports A1 and B1 of the pump are directly connected to the single rod cylinder, and port C1 is connected to an accumulator directly and to a tank through a check valve. Two pressure relief valves are utilized to limit the operating pressure, respectively.
During the cylinder extension process, the pump sucks oil from the cylinder rod chamber through port B1, and from tank or accumulator through port C1. Then the pump discharges oil to the cylinder rodless chamber. During the cylinder retraction process, the pump sucks oil from the cylinder rodless chamber through port A1. Then the pump discharges oil to the cylinder rod chamber and to the accumulator through port B1 and C1.

\( n \) is the rationing speed of the pump, \( v \) is the velocity of the cylinder, \( A_s \) and \( A_d \) represent the area of the rodless and rod chamber of the cylinder. \( D_s, D_u \) and \( D_c \) are the displacements of the pump ports \( A_1, B_1 \) and \( C_1 \). When the system is used to control the boom cylinder, the control cavity is the rodless chamber. The cylinder velocity can be written as \( v = D_s/n/A_s \). As the displacement ratio is designed to equal to the area ratio of the rodless and rod chamber of the cylinder, the flow rates in and out of the pump and cylinder can match to each other basically.

### 2.2 Asymmetric pump

Fig.2 gives the working principle and a photograph of the valve plate and cylinder block of the new designed asymmetric pump. As shown in Fig.2, it can be seen that there are four assignment windows on the valve plate, named A, B, C and D, where A and C are in the same circle with a radius of \( R_1 \) and B and D on one with \( R_2 \). Both windows A and B are connected to each other by the port \( A_1 \) on the pump end shell cover. Both windows C and D are one-to-one correspondence with the pump ports of B and C. There are 10 plunger chambers that are divided into two groups averagely. At the bottom of the cylinder block, there are an inner annular array and an outer annular array. The pitch radiuses of those two annular arrays match with the slots of A, B, C and D on the valve plate. As shown in Fig. 2(b), the plunger chambers are identified as \( S_i \), corresponding to the outer annular array, and those five pistons suck and discharge the oil only from the outer annular array. Such benefits from the asymmetric structure of the flow distribution make the flow rates ratio of the three ports of the pump be 1:0.5:0.5. And an area ratio of the single rod cylinder can be matched to change the flow rate ratio by adjusting the piston diameters or \( R_1 \) and \( R_2 \). Benefiting from such asymmetric structure of the pump, the flow rates of the cylinder and the pump can be balanced basically.

### 2.3 Theoretical analysis

According to the flow continuity equation of hydraulic cylinder, the flow of the cylinder can be formulated as Eq.1 and Eq.2.

\[
q_i = A_s v + \frac{V_l}{\beta_s} \frac{dp_s}{dt} \tag{1}
\]

\[
A_v v = q_i - \frac{V_l}{\beta_s} \frac{dp_s}{dt} \tag{2}
\]

During the working period, the hydraulic cylinder is under the action of hydraulic pressure and load force, according to the force balance equation, the dynamic equation of the hydraulic cylinder can be shown in Eq.3.

\[
p_s A_k - p_b A_k = m d^2 v/dt^2 + F \tag{3}
\]

The pump is under the action of hydraulic load pressure, accumulator pressure and torque regulated by the electric motor, according to the torque balance equation, the dynamic equation of the hydraulic pump can be shown in Eq.4.

\[
T_p = \frac{d}{dt} \left[ B_n \pi \frac{1}{2} (p_i D_k - p_b D_k - p_b D_b) \right] \tag{4}
\]

With the asymmetric pump controlled boom cylinder system, the pressure in the rod chamber, \( p_b \) can be assumed as zero. And for the hydraulic cylinder with an area ratio of 1:2, the displacement ratio of the three port of the
3 Experimental system

In order to provide compared data about working performance and energy efficiency, the test of boom cylinder with the independent metering system driven by an inverter motor is implemented first. Fig.3 gives the principle of the experimental system controlled by independent metering valves.

Fig.3 Boom cylinder controlled by separate metering system

In Fig.3, the system in which boom cylinder, arm cylinder and swing motor are controlled by independent metering system driven by an inverter motor. An electro hydraulic axial piston variable displacement pump which the pressure and flow are continuously tunable is used in this system. There are additional instruments such as displacement sensors in the actuators, pressure sensors installed to detect the pressure inside the actuators and pump ports, the power sensors and rotational speed sensor on the motor to detect the electric power and rotational speed. The control concepts are being realized by the hardware in the loop computer control system ds1103.

The proposed system works as below: the controller takes in the inputs of the joystick and the measured quantities such as pressures, powers, displacements, flow and speed of the motor. And then the controller analysis the demand automatically and output the voltage signals directly applied to the converter motor, pump and valves according to the set strategy [6].

After the test of the boom cylinder controlled by independent metering system, the asymmetric pump controlled arm cylinder test rig is constructed, as shown in Fig.4.

3.1 Working performance and energy efficiency

3.1.1 With independent metering system

The main task of the boom cylinder is to lift the working mechanism and load to the specified position. Its operational characteristics are that the load condition during the extension process is resistance mainly, and which is overrunning mainly during the retraction process.

In the process of frequent retraction, the gravitational potential energy of the working mechanism is dissipated into heat in the control valves, and the pump still outputs the flow to the cylinder during the overrunning retraction process. A simple way to reduce the energy consumption is using the flow regeneration technology. In this paper, when the boom cylinder extends, the inlet and outlet oil valves open fully to reduce throttling loss. And then the regeneration technology is used when the boom cylinder retracts.

The pressures of the system and position of the boom cylinder with respect to time are presented in Fig.6 (a). It can be seen that the pressure of the pump, \( p_{\text{Pump}} \), is about 13.0 MPa and the pressure in the cylinder rodless cavity, \( p_{\text{Cavity}} \), is about 10.1 MPa and which is 0.8 MPa in the rod cavity. Thus, the pressure loss of the system, \( \Delta p \), is...
about 2.9 MPa. By referring to the results, during boom cylinder extension process, the pump pressure tends to oscillate which is similar to the load sensing system. And, there is a sudden decrease pressure during the startup stage of the cylinder retraction.

Fig.6 (b) gives the electric power and energy input to the inverter, when the boom cylinder velocity is about 100 mm/s. It can be seen that the electric power input to the inverter, $P_e$, is about 14.5 kW during the boom cylinder extending the process, and which is 2.3 kW during the cylinder retraction and non-working process. During a complete extension and retraction process, the energy input to the inverter, $E_e$, is about 64.2 kJ.

When the boom cylinder velocity is about 100 mm/s, the control signal is zero, the speed of the motor is set to zero and the hydraulic cylinder does not move. It can be seen that when the speed of the servo motor is zero, the cylinder displacement goes down gradually due to the leakage.

3.1.2 Simple asymmetric pump controlled without energy recovery

Fig.7 gives the measured cylinder displacement, pump pressure, electric power, and energy input to the inverter, and the hydraulic energy output of the pump, when the pump port T is connected to the oil tank. During 0 ~ 2 s, 9 ~ 14 s and 17.8 ~ 20 s, the control signal is zero, the speed of the motor is set to zero and the hydraulic cylinder does not move. It can be seen that when the speed of the servo motor is zero, the cylinder displacement goes down gradually due to the leakage.

As shown in Fig.8 (a), it can be seen that, during the extension process, the pressure of pump port A and cylinder rodless cavity, $p_{rodless}$, is about 10.4 MPa and which is 1.2 MPa in the rod cavity. And during the working process, the pressure in the cylinder is respectively stability.

Fig.8 (b) gives the electric power and energy input to the inverter when the boom cylinder velocity is about 40 mm/s. It can be seen that the electric power input to the inverter, $P_e$, is about 9.24 kW during the boom cylinder extending the process, and which is 3 W during the cylinder retraction and non-working process. Compared to the independent metering system, the power consumption can be reduced about 2.0 kW during cylinder retraction with the asymmetric pump controlled system.

3.1.3 Asymmetric pump controlled with energy recovery

As shown in Fig.8 (a), it can be seen that, during the extension process, the energy input to the inverter, $E_e$, is about 31.0 kJ and which is 32.1 kJ during a complete extension and retraction process. The energy consumption per stroke during the extension process can be calculated as $\Delta E_e/\Delta x$, which is about 0.14 kJ/mm. And it is about 0.20 kJ/mm with the independent metering system. Thus, it can be concluded that the energy saving ratio is about 30%.

3.2 Energy consumption compared under different velocity

As shown in Fig.8 (a), it can be seen that, during the extension process, the energy input to the inverter, $E_e$, is about 16.2 kJ and which is 18.3 kJ during a complete extension and retraction process. The energy consumption per stroke during the extension process is about 0.07 kJ/mm. And it is about 0.16 kJ/mm with the independent metering system. Thus, it can be concluded that the energy saving ratio is about 56%.

Fig.9 gives the comparison of energy consumption per stroke under different velocities with pump and valve controlled system. The results show that the energy consumption of the two system decreases as the velocity increases. The energy efficiency increasing of the electric motor is the main cause of this phenomenon. Compared to the independent metering system, the asymmetric pump controlled system was 50% more efficient.
during the extension process. With the Figs.7-9, the estimated 41% of the potential energy could be regenerated and the utilization efficiency reaches to 90%.

**Fig.9 Energy consumption under different velocities**

### 4 Conclusion

Due to large throttling loss and gravitational potential wasting, the energy efficiency of the hydraulic excavator is low, which is even lower than 10%. This paper aims to improve the energy efficiency of the hydraulic excavator by reducing throttling loss and regenerating potential energy directly. The system under consideration utilizes a newly designed asymmetric pump which has three ports, the two are connected to the hydraulic cylinder, the other is connected to an accumulator. Thus, this system can regenerate the potential energy directly without throttling loss and the utilization efficiency reaches to 90% which is very high than an existed project.

And compared with an independent metering circuit driven by a convert motor, the working performance of the excavator boom system with the asymmetric pump is stable. Compared with the independent metering circuit, the electric power consumption during the boom going up can be reduced by more than 40%. And if the energy efficiency of the power source is considered, during the whole process of the boom cylinder working, the energy consumption can be reduced by 50%.

### 5 Acknowledgements

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This work was supported by the National Natural Science Foundation of China (Grant Nos. U1510206 and 51575374).

### Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A_r</td>
<td>Area of the Cylinder Rodless</td>
<td>[m²]</td>
</tr>
<tr>
<td>A_b</td>
<td>Area of the Cylinder Rod</td>
<td>[m²]</td>
</tr>
<tr>
<td>B_v</td>
<td>Viscous Damping Coefficient</td>
<td>[Ns/m]</td>
</tr>
<tr>
<td>B_m</td>
<td>Rotational Viscous Damping Coefficient</td>
<td>[Nm/rad]</td>
</tr>
<tr>
<td>D_a</td>
<td>Displacements of the Pump Ports A_1</td>
<td>[m]</td>
</tr>
<tr>
<td>D_b</td>
<td>Displacements of the Pump Ports B_1</td>
<td>[m]</td>
</tr>
<tr>
<td>D_c</td>
<td>Displacements of the Pump Ports C_1</td>
<td>[m]</td>
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</tbody>
</table>

### References


Entrainment of free water into hydraulic systems through the rod sealing

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Water in oil-based hydraulic systems is a source for many machinery failures. It accounts for up to 20% of the life expectancy failures and even before that, it impacts the expected performance negatively /1/. Water can enter a hydraulic system in various ways. In this article, the entry through the dynamic seal of the rod is investigated. After a brief description of the damage mechanisms of water in a hydraulic system, the theory of the entrainment is explained. The test bench is then described to study the effect. Finally, entrainment results for two test fluids (oil and water) are presented and compared to the theory.

Keywords: Rod Sealing, Water, Contamination, Reynolds Equation
Target audience: Science Community, Sealing manufactures

1 Damage in hydraulic systems by water

This section gives an overview of the damaging mechanisms of water in hydraulic systems. In addition to the damage to the pressure medium, the used components, made from different materials, are impacted by water.

1.1 Pressure medium ageing

The ageing process of pressure media used in hydraulics in presence of water occurs due to two mechanisms: oxidation, which occurs in mineral oil based fluids and hydrolysis, which takes place in ester based fluids. The effect of water plays the role of a catalyst and accelerates the process. Due to their relatively high electronegativity, oxygen ions can dissociate a hydrogen atom from a hydrocarbon molecule. This produces a hydrocarbon radical that forms a hydrocarbon peroxide radical with an oxygen atom. This is very reactive and attacks other molecules. The further elimination of a hydrogen atom results in a hydrocarbon hydroperoxide and a new hydrocarbyl radical. The formation of more and more radicals keeps the chain reaction going on resulting in products such as alcohols, aldehydes, ketones and acids.

The process of hydrolysis is displayed in Figure 1.

![Figure 1: Hydrolysis on a 3-fold ester /2/, /3/](image1)

The two-valued alcohol group (shown on the left, encircled) has only a low hydrolytic stability. Water dissolves this site and forms an alcohol. The separated molecule residue becomes an acid (in the picture on the right). As a result, acidification of the system can occur, due to which metallic components can be corrosively attacked.

1.2 Seal Damage

Seals in hydraulic systems, which have the task of preventing leakage, come into direct contact with the fluid and react with it, so they can be attacked by the oil’s water contamination /4/. In /4/ the influence of water in oil on weight change of polyamide sealing elements after a defined time under the influence of different oil types and various humidity ranges is investigated. It is shown that with dry oil (0% water content), the sealing elements lose weight. Since at higher content levels water diffuses out of the oil into the sealing material, a weight gain is observed. That could squeeze the seal out of its installation groove and is commonly known as gap extrusion.

1.3 Damage to metallic components

In /4/ the influence of water on metallic components with regard to corrosion is investigated. There is a risk of corrosion particularly at locations where water can settle, for example at the bottom of the vessel. In valves or in narrow sealing gaps (e.g., in axial piston pumps) there is the risk that corrosion products lead to a clamping of the components and thus to dangerous or at least unwanted behaviour of the system.

2 Theory of Blok

The sealing mechanism of rod seals is commonly described by Blok’s theory /5/. On the basis of this theory, fluid can be entrained into the hydraulic system via the rod seal. The basics and the resulting entrainment mechanism are explained below.

In the case of dynamic seals, a fluid-filled sealing gap (gap height h usually <1 μm) is formed between rod and seal. The pressure distribution in the fluid gap corresponds to the previous pressing in the gap, due to the preloading of the seal during assembly and the system pressure. The mechanism of returning the fluid is called dynamic sealing. The calculation of rod seals is based on the “inverse-Reynolds theory”. Rod seals are dynamically sealed when the oil volume being pulled out during the extension of the cylinder is smaller than the oil volume that could theoretically be dragged back into the cylinder during retraction. The drag volume depends on the rod diameter and the lubricant film height. For the purpose of designing rod seals, the lubrication film heights for inward and outward travel are calculated and compared. For dynamic tightness, the theoretical retraction film height must be larger than the extension film height.

The Reynolds equation, applied to the gap between seal and rod, takes the form as given in Equation (1). It couples the spatial pressure gradient \( \frac{\partial p}{\partial x} \) with the viscosity \( \eta \), the rod velocity \( u_R \) and the gap height \( h \). \( h_0 \) is the lubricating film height at the point of the pressure maximum at which the spatial gradient equals zero.

\[
\frac{h^2}{\eta} \frac{\partial p}{\partial x} - 6 \cdot \eta \cdot u_R \cdot (h - h_0) = 0
\]  (1)

Rod seals are pressed against the rod by the preload and the applied system pressure when the system is not moving. The resulting stress is supported by the lubrication film on the rod via the fluid pressure. Furthermore, the pressure in the lubrication film normal to the piston rod is assumed to be constant /5/, /6/. For the calculation of the lubricating film thickness, therefore, the fluid pressure is equated with the stress distribution previously determined by experiments or FEM calculations. A typical pressure profile as well as the velocity distribution in the lubrication gap is shown in Figure 1.
The system pressure \( p_{slip} \) is applied to one side of the seal. Due to the geometry of the sealing element, the pressure rises very steeply (steeper than shown in the figure, since otherwise the relevant points could not be represented). When a maximum is reached, the pressure drops to ambient pressure \( p_{LH} \). The pressure increase prevents leakage of the fluid. The best sealing is obtained with a triangular pressure profile with the maximum of the pressure distribution where the pressure driven fluid velocity is 0 and its distribution therefore represented. When a maximum is reached, the pressure drops to ambient pressure (Equation (2)).

\[
\eta \frac{\partial^2 p}{\partial x^2} + \frac{\partial}{\partial x} \left( 3h^2 \frac{\partial p}{\partial x} - 6\eta \frac{\partial h}{\partial x} \right) = 0
\]

(2)

In an inflection point of the pressure distribution (where the greatest change in pressure is observed) the first term of the left side of the equation equals 0. To fulfill the equation the difference in the brackets has to equal 0 as well. Rearranging the equation reveals the gap height underneath the inflection point of the pressure distribution (Equation (3)).

\[
h_a = \sqrt{\frac{2\eta}{9\eta - 6\eta \frac{\partial p}{\partial x}}} h_a^* \]

(3)

Inserting \( h_a \) back into Equation (1) and evaluating it at the same spot delivers the gap height \( h_a^* \) underneath the maximum of the pressure distribution where the pressure driven fluid velocity is 0 and its distribution therefore linear (Equation (4)).

\[
h_a^* = \sqrt{\frac{8}{9\eta - 6\eta \frac{\partial p}{\partial x}}} h_a
\]

(4)

Assuming that the volume remains constant, \( h_a \) on the rod can easily be derived by applying volume consistency (Equation (5)).

\[
h_a = \frac{1}{2} h_a^* = \frac{2}{9\eta - 6\eta \frac{\partial p}{\partial x}} h_a
\]

(5)

The entrainment potential can be quantified as a difference volume between retraction and extension based on the presented mechanism. This is calculated according to Equation (6). Index \( a \) and \( A \) stand for outward travel, \( e \) and \( E \) for inward, the same theory applies to both directions.

\[
\Delta V = \pi dh \left( \frac{u_a}{\sqrt{\frac{\partial p}{\partial x} \frac{\partial h}{\partial x}}} - \frac{u_e}{\sqrt{\frac{\partial p}{\partial x} \frac{\partial h}{\partial x}}} \right)
\]

(6)

If \( \Delta V \) is equal to or greater than 0, which means that more volume can be entrained through the seal than travels out before, the sealing is called tight.

When fluid is applied to the outside of the seal, e.g. water on the rod, it can be entrained when the rod is retracted. This is particularly problematic in mobile hydraulic machines, which are also operated under rain and partly under water.

Water has a different viscosity than hydraulic oil and will therefore be dragged differently. The ratio of the entrained volumes, assuming that the pressure distribution and the speed of the rod remains the same, is given in Equation (7). It is furthermore assumed, that the absolute oil film height during extension is negligible compared to the water film height during entrainment.

\[
\frac{\Delta V_{\text{Water}}}{\Delta V_{\text{Oil}}} = \sqrt{\frac{\eta_{\text{Water}}}{\eta_{\text{Oil}}}}
\]

(7)

A similar ratio is found in /7/.

Using literature data for viscosity of water and oil at different temperatures, the expected relation can be plotted as shown in Figure 3.

![Figure 3: Expected relation for entrained water vs. entrained oil](image)

For an average temperature range of hydraulic systems, the relation ranges from 10% up to 16.5%.

The theory given is only valid if the oil layer, which adheres to the rod when it is extended; remains adhered to it during the entire time. When supplied with water, it will lie on top of the oil film with no influence on it.

3 Test bench

In this section, the developed test bench to measure entrained water through the rod seal is presented. Furthermore, a proof of concept of the system is conducted.

3.1 Test chamber

At the Institute for Fluid Power Drives and Controls (IFAS), a test bench was set up to investigate the entrainment of free water. The test bench is shown in Figure 4.
3.2.1 Validation of the pressure dependency

In order to verify the suitability of the measuring system, the relationship between chamber pressure and displacement of the piston was checked. The theoretical relation is given in Equation (8) /9/:

\[
\Delta p = \frac{E' F F}{V_0} \cdot \Delta V = \frac{E' F F}{V_0} \cdot A \cdot \Delta x
\]

\(\Delta p\) is the pressure rise due to an additional volume \(\Delta V\) which is squeezed into the original volume \(V_0\). \(E' F F\) is the corrected bulk modulus of the fluid. In this case \(\Delta V\) can be calculated as a function of the displacement \(\Delta x\) of the piston and its cross sectional area \(A\). \(V_0\) and \(E' F F\) are considered constant which leads to a linear relation between \(\Delta x\) and \(\Delta p\). Water has a negligible influence on the bulk modulus as it is only present in small fractions (see Figure 3) and its immiscibility with oil.

The piston is displaced by increasing the pressure on the back thus the resulting pressure in the chamber along with the actual displacement of the piston are measured. The results are shown in Figure 6.

![Figure 6: Measured relation between piston displacement and chamber pressure](image)

As shown in the Figure there is a good linearity between piston displacement and chamber pressure. In addition the constancy of back pressure in respect to piston displacement was evaluated.

3.3 Temperature dependency

Both results, the linearity between piston displacement and chamber pressure as well as the constant back pressure, lead to the conclusion, that the entrainment sensor is well suited for the purpose.

3.2 Entrainment Sensor

The entrained volume is measured with a self-developed sensor which is shown in Figure 5. The hydraulic part is drawn in greater detail to show relevant components (red box).

![Figure 5: Entrainment sensor](image)

It consists of a piston, which is sealed by a rod seal. The space in front of the piston is connected to the chamber. A LVDT is attached to the piston measuring the displacement. Entrained water volume \(\Delta V\) displaces the piston and can be calculated with the piston diameter. In order to maintain a constant pressure \(p_{\text{back}}\) in the chamber, pressure \(p_{\text{back}}\) is applied to the reverse side of the piston. This pressure is provided by a 4-liter hydro-accumulator, which is almost completely empty. The displaced volume of the piston is taken up into the accumulator and the pressure remains constant during operation.
During tests, an increase in oil temperature may occur due to the change in ambient temperature, frictional heat or other circumstances. As a result, the oil expands and, especially in the present case, this would lead to a movement of the piston and thus falsify the measurement results. In the following section, the temperature-dependent displacement effect of the sensor’s piston is explained. Then a method is presented which is used to neglect the effect due to recalculation. Finally, the results of the heating measurement which are necessary for the parameterisation of the method are presented. This is done when the rod is at a standstill.

3.3.1 Temperature induced expansion of volume

With a rise in temperature most materials expand their geometric dimension. For liquids, this effect is expressed in an increase of volume. In the present case, this would lead to a displacement of the piston and would thus falsify the measurement results.

In order to calculate this effect, measurements in dependency of temperature were carried out. The temperature is increased slowly, and the displacement of the piston is recorded. In the experiments, the temperature-dependent displacement can thus be subtracted from the measurement signal, and the entrained volume can be determined according to Equation (9).

\[ \Delta V_{\text{entrained}} = \Delta V_{\text{measured}} - \Delta V_{\text{therm}} = A_{\text{piston}} \cdot (\Delta \alpha_{\text{measured}} - \Delta \alpha_{\text{therm}}) \]  

\( A_{\text{piston}} \) represents the cross sectional area of the piston and \( \Delta \alpha \) the displacement of the piston (measured and due to thermal effects). It is assumed, that the inside volume of the chamber remains constant during testing and that only the volume expands. As the chamber system is closed and state of the art sealing is applied, it is furthermore assumed, that the oil mass in the chamber remains constant.

According to \( /9/ \) the change in volume of a mineral oil \( \Delta V \) can be calculated using Equation (10).

\[ \Delta V_{\text{therm}} = V_0 \cdot \gamma \cdot \Delta \alpha = A_{\text{piston}} \cdot \Delta \alpha_{\text{therm}} \]  

\( V_0 \) is the original Volume, \( \gamma \) the expansion coefficient (here \( 7 \cdot 10^{-6}\text{K}^{-1} \)) and \( \Delta \alpha \) the deviation of the temperature from a reference temperature. Rearranging of Equation (10) gives a relation between an increase in temperature and the resulting displacement (Equation (11)).

\[ \frac{V_0 \cdot \gamma}{A_{\text{piston}}} \cdot \Delta \alpha = \Delta \alpha_{\text{therm}} \]  

Derivation of Equation (11) yields the displacement gradient with respect to the temperature.

Furthermore Equation (12) applies:

\[ \frac{\theta}{\theta} = \frac{\Delta T}{\Delta T} = \frac{\Delta x}{\Delta x} \]  

Putting the design (given in Table 1) and fluid parameters into this Equation the numerical relation between \( \Delta \alpha \) and \( \Delta \alpha_{\text{therm}} \) is derived (Equation (13)).

\[ \frac{5.47 \text{ mm}}{K} \cdot \Delta \theta = \Delta \alpha_{\text{therm}} \]  

\begin{table}[h]
\centering
\begin{tabular}{|c|c|}
\hline
\( V_0 \) & 220921.7 mm\(^3\) \\
\hline
\( A \) & 28.27 mm\(^2\) \\
\hline
\end{tabular}
\caption{Test bench design parameters}
\end{table}

This means that an increase in temperature of 1 K will theoretically result in a displacement of 5.47 mm of the piston.

Piston displacement due to a temperature change can be compensated by measuring the temperature of the fluid, calculating the resulting change and subtracting it from the measured displacement \( \Delta \alpha_{\text{measured}} \) (Equation (14)).

\[ \Delta \alpha_{\text{therm}} = \Delta \alpha_{\text{measured}} - \Delta \alpha_{\text{therm}} = \Delta \alpha_{\text{measured}} - \frac{5.47 \text{ mm}}{K} \cdot \Delta \theta \]  

3.3.2 Heating tests

To verify Equation (13) heating tests are carried out in small increments of about 2°C in the temperature ranges of the actual measurements. The rod is in stand still. The temperature change is thereby only caused by heat conduction and therefore requires a certain time to adjust itself. During the entrainment tests, the oil is moved through the rod and the temperature of the oil is assumed to be homogeneous. Therefore, conclusions can be drawn with the results of the heating tests on the temperature-induced volume increase during the entrainment tests.

Figure 8 shows the result of the heating test. The piston stroke of the entrainment sensor over the entire investigated temperature range would exceed the maximum stroke. Therefore, the piston is returned to the initial position for each measurement and only the relative changes are considered.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure8.png}
\caption{Temperature induced sensor displacement for different temperature ranges}
\end{figure}

It can be seen that the determined gradients deviate from the theoretical determined value. Reasons for this could be, for example, the non-linear expansion behavior of the sealing materials. Furthermore, the original oil volume can differ due to manufacturing inaccuracies. In addition, the expansion of the chamber itself can lead to a smaller gradient than determined before. As the last and most likely cause, it can be assumed that the heat coefficient used in the calculation differs from the actual.

The mean gradient determined is \( 3.50 \text{ mm} / \text{K} \) with a standard deviation of 3.96%. This can be attributed to the resolution of the used sensors. The temperature sensors resolution is 0.1°K, which corresponds to an inaccuracy of 5% at a temperature range of around 2°C. The displacement measurement system, which measures the displacement of the piston, has a resolution of 0.075 mm, a further inaccuracy of 0.9% in the measured range. Overall accuracy therefore accounts to up to 5.9% as the errors are connected serially. The mean gradient was used to correct the influence of the temperature change during the entrainment measurements.

4 Entrainment Measurements

Subsequent, the results of the intake measurements are explained, which have been corrected for the temperature influence. Experiments are carried out with oil and with water separately, first oil and second water is used as test fluid. The pressure is 30 bar and the velocity of the rod 175 mm/s. In addition, the temperature is varied.

4.1.1 Measurement results

The raw travel signals of the piston displacement and the raw data of the temperature have discontinuities due to Stick-Slip of the entrainment sensor piston as well as due to discrete data increments of the temperature sensor and the used LVDT. Therefore, the measurement data for the displacement and the temperature were
approximated by means of a second degree polynomial before the temperature compensation. Figure 8 shows the raw and the approximated signal for the temperature (left) and for the piston displacement (right).

Figure 9: Raw data and approximated function, temperature left, displacement right

It is obvious, that there is good alignment between the approximated curve with the raw data of the displacement and a sufficient alignment of the temperature data. In the following, the approximated data is used for further calculations. A total of three measurement series are conducted for each measuring point. The data for each point is then approximated, temperature compensated and finally averaged.

Figure 10 shows the results of entrained fluid for different temperatures at a pressure of 30 bar. The red line represents the tests with oil as test fluid whereas the blue stands for water as test fluid.

Figure 10: Entrainment of oil vs. water at different temperatures and 30 bar

Furthermore, the standard deviations of the averaged and compensated measurement series were calculated and are given in Table 2.

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Standard deviation Water</th>
<th>Standard deviation Oil</th>
</tr>
</thead>
<tbody>
<tr>
<td>24.9°C</td>
<td>10.4%</td>
<td>10.6%</td>
</tr>
<tr>
<td>33.2°C</td>
<td>3.65%</td>
<td>1.47%</td>
</tr>
<tr>
<td>41.3°C</td>
<td>98.0%</td>
<td>91.8%</td>
</tr>
</tbody>
</table>

Table 2: Measurement standard deviations

For the highest temperature of 41.3°C, no significant entrainment could be measured, which simultaneously results in a very large standard deviation of the individual measuring series.

For the measurement at 24.9°C it can be seen that oil is dragged into the system. In the case of water, the entry is negative; meaning that fluid is discharged from the system. About 600 µl of oil are entrained over a total stroke of 100 m and some 200 µl is discharged when water is supplied.

At a temperature of 33.2°C, the intake curves are similar to the previous temperature. Approximately 100 µl of oil are drawn in over a stroke of 100 m whereas 300 µl are discharged when supplied with water.

4.1.2 Comparison and discussion of entrained fluid volume results

Linear ratios can be obtained for the linear entrainment curve at 24.9°C and 33.2°C. This is not the case for the temperature of 41.3°C, since the draw-in equals zero. The average ratios are given in Table 3. The value of the theoretical ratio from Figure 3 is also given.

<table>
<thead>
<tr>
<th>Temperature</th>
<th>( \frac{\Delta V_{\text{Water}}}{\Delta V_{\text{Oil}}} ) measured</th>
<th>( \frac{\Delta V_{\text{Water}}}{\Delta V_{\text{Oil}}} ) theoretical</th>
</tr>
</thead>
<tbody>
<tr>
<td>24.9°C</td>
<td>-0.366</td>
<td>0.108</td>
</tr>
<tr>
<td>33.2°C</td>
<td>-3.064</td>
<td>0.119</td>
</tr>
</tbody>
</table>

Table 3: Comparison of the measured and theoretical entrained volume ratios

It is clear that the ratios are very different from each other. Therefore, it can be assumed that the theory explained in chapter 2 is not applicable in the case of water entrainment.

A possible explanation for fluid volume exiting the chamber when external water is supplied may be that the oil film adhering to the rod during extension is detached in contact with water and rises due to the difference in density and is transported away from the sealing gap accordingly. The only lubricant then is water, which has a lower viscosity and is only slightly entrained back in. To ensure the measurement results other possibilities of rod-wetting will also be implemented and tested in the future.

5 Summary and conclusion

In this article, the water entrainment potential across piston rod seals in hydraulic cylinders was discussed. First, the theory of Blok was applied to the case of water and necessary corrections for the viscosity were explained. Afterwards, the test bench was explained and the sensitivity and suitability were examined. Furthermore, the method for correcting the temperature change during the measurement was presented. Subsequently, the measurement results of the intake for oil and water were displayed and discussed. It was found that the measured intake volumes of water are negative over the stroke. In total, it was possible to measure entrainment of oil over a stroke length of 100 m of up to 600 µl and a loss of fluid volume in the chamber of up to 300 µl when suppling water at 30 bar chamber pressure.

It can be stated that the experimental results presented above do not correspond to that of the theoretical predictions. The most likely explanation is that the oil film attached to the rod is detached when contacted with water due to density differences. Water is then the only lubricant in the sealing gap which has a lower viscosity which leads to a smaller intake film height than the height during extension. Over all, fluid volume exits the chamber which has been measured.

6 Acknowledgements

The authors thank the Research Association for Fluid Power of the German Engineering Federation VDMA for its financial support. Special gratitude is expressed to the participating companies and their representatives in the accompanying industrial committee for their advisory and technical support.
Nomenclature

<table>
<thead>
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<th>Variable</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$A$</td>
<td>Cross-sectional area</td>
<td>[mm$^2$]</td>
</tr>
<tr>
<td>$d$</td>
<td>Rod diameter</td>
<td>[mm]</td>
</tr>
<tr>
<td>$E'_p$</td>
<td>Bulk modulus</td>
<td>[bar]</td>
</tr>
<tr>
<td>$h$</td>
<td>Gap height</td>
<td>[μm]</td>
</tr>
<tr>
<td>$H$</td>
<td>Stroke</td>
<td>[m]</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>$\Delta p$</td>
<td>Pressure difference</td>
<td>[bar]</td>
</tr>
<tr>
<td>$\Gamma$</td>
<td>Temperature</td>
<td>[°C]</td>
</tr>
<tr>
<td>$u$</td>
<td>Velocity</td>
<td>[mm/s]</td>
</tr>
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<td>$V_0$</td>
<td>Original volume</td>
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</tr>
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<td>$\Delta V$</td>
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<td>[μl]</td>
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<td>Dynamic viscosity</td>
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</tr>
<tr>
<td>$\gamma$</td>
<td>Expansion coefficient</td>
<td>[K$^{-1}$]</td>
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</tbody>
</table>

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Development of an interface between a plunger and an eccentric running track for a low-speed seawater pump


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The DOT concept for offshore wind energy is a seawater hydraulic network where turbines are directly coupled to a centralized hydro-power platform. The essential missing component is a low-speed hydraulic pump that uses seawater as its hydraulic medium. This low speed hydraulic pump is currently being designed and tested by DOT, where novel machine components have been developed. This paper describes the development of an interface between an oval running track which is used as eccentricity to actuate a hydraulic piston. Several approaches have been performed as well as prototyping and validation steps. These steps as well as the design approach are presented in this work.

Keywords: Fluid power networks, positive displacement pump, Eccentricity, crank shaft replacement.

Target audience: (Sea)water Hydraulics, Offshore Industry, Design Process, Machine design.

1 Introduction

Advancements in wind turbine design mean records in terms of turbine size and power capacity continue to be broken. As blade length increases, the nominal rotation speed of wind turbines decreases asymptotically, and torque increases exponentially. In current industries were robust machines are used to handle large torques, hydraulic systems are usually preferred. Hence, the case for a compact hydraulic power train for a wind turbine is prevalent.

The Delft Offshore Turbine (DOT) concept makes use of a seawater hydraulic network where every wind turbine rotor is directly coupled to a positive displacement pump and electricity generation is centralized, creating an offshore hydro power plant /1/. Key design drivers are mass reduction and minimizing maintenance. For much of the DOT concept, existing technology can be used as a basis. This also applies partially for the hydraulic drive train. Hydraulic piping and hydro power plant stations are mature technologies, and therefore do not need to be specifically developed. The essential missing component for the DOT drive train system is a pump that makes use of seawater as its hydraulic fluid. In previous studies /1/, it was determined that a positive displacement pump is the best option for the application in terms of torque to mass ratio, mechanical and volumetric efficiency. There is however no commercially available low-speed and high-torque positive displacement pump able to handle seawater as its hydraulic medium. Hence the decision was made in the spring of 2016 to develop such a machine inhouse. This paper focuses on an essential interface inside this pump: the conversion from rotary drive motion to linear piston motion. Two particular design solutions are described, including their prototyping and performance.

2 Design challenge

Previous work on the DOT approach made use of an existing second hand V44 Vestas wind turbine with a rated power output of 600 kW. The turbine was retrofitted with back-to-back oil and water hydraulic circuits to fundamentally prove the DOT approach at full scale /2-3/. Since the V44 Vestas was already available as a prototyping platform the choice was made to design the pump using the V44 Vestas nacelle geometry as design start condition. This restriction on the pump radius proved to be a major design challenge, so much so that the design pressure had to be restricted to 160 Bar. Also, other components in the hydraulic loop proved to be a limiting factor, thus restricting the design pressure to 160 Bar. A high-level overview of the DOT piston pump design can be seen in Figure 1. Lowering the design pressure meant lowering the input torque. As part of a follow-up project, a second pump segment is added to double the torque and make the pump design modular. Hence, the nominal volume flow of 1600 L/min and rated input torque 280 kNm are divided over the two modular segments.

Figure 1: System architecture of one of the modular pump segments. The architecture can be divided into its three main interfaces. The transmission converts the input of the wind turbine into a higher rotational velocity. The cylinder interface converts an eccentric motion into a translation.

Another method to keep the pump dimensions within the restrictions was to increase the rotational speed of the wind turbine shaft from 28 RPM at max speed to 600 RPM by means of gearbox. This transmission is used to actuate an oval that is used as eccentric running track. This eccentric running track, called a wave generator ring (WGR), can be seen in Figure 2. The oval creates two strokes during every rotation, causing an effective speed of cylinder actuation at 1200 RPM. Hence, by lowering the design pressure, introducing a gear ratio and eventually adding a second module, the pump design is made sufficiently compact. The remaining challenge is the interface between the wave generator ring and the to-be actuated piston.

Figure 2: Overview of the drive and the placement of the wave generator ring.
3 Method

In this section the different approaches used to create the drive to plunger interface are discussed. The constraint on the pump dimensions meant that only a select number of design approaches could fit. Because of the nature of the problem, three approaches were originally discussed as possible solutions. These are sliding, rolling and fluid bearing interfaces. Of these approaches, both the sliding and rolling bearing interfaces are the most well-known and therefore have a lower risk during development. Both these approaches were designed and analysed. Both the sliding and rolling bearing interface design will be discussed.

3.1 Sliding bearing interface

A sliding bearing is a basic commonly used tribological interface, found in a significant amount of machines /4/. As such, it proved to have great potential as the interface between the WGR and piston. To actuate the piston by as much of a pure translation as possible, an additional interface was designed between the WGR and piston. This interface consists of a combination of rollers, circular flexure and sliding bearing. These will be called inner rollers, waveband and slider bearing from this point onwards. The basic layout of this configuration can be seen in Figure 3.

Figure 3: Sliding surface bearing approach. By adding additional rollers and a so-called waveband, the piston is actuated through almost pure translation.

By adding this additional rollers and flexure, a conversion from rotation to translation could be conceived. This created the following additional challenges:

- The design of a large single flexure with a deflection of 50 mm.
- Design of inner rollers which are subjected to high load cases up to 45 kN.
- Design of a stable sliding bearing with a lifetime of 6 months.

Since the dimensions were limited, the max size of the sliding bearing and its possible sacrificial material was therefore also limited. These main objectives were the goal of detail designing of this principle.

3.1.1 Detail design: sliding bearing

The limiting dimensions meant only 10 mm of sacrificial material was available for the sliding bearing. With the goal of 6-month operational lifetime, this means a wear rate of the sliding bearing of 2.3 μm per hour. This is at operational conditions at 1200 RPM/160 Bar. To obtain a low wear rate with a sliding bearing, the PV-value has to be low. Because of the configuration of the waveband and sliding bearing, there still exists an oscillating horizontal motion between the waveband and bearing. This oscillating motion occurs over a distance of 50 mm, with an average velocity of 1.87 m/s. Next to this is the horizontal stroke of the cylinder caused by the deflection of the waveband. To obtain the desired flow, with the chosen cylinder dimensions, this stroke is equal to 50 mm. The geometry of the sliding bearing is designed such as to maximize the amount of surface area, therefore lowering the contact pressure on the bearing with the waveband. The surface area of the final slider bearing is equal to 785 mm². With this surface area the trajectory made by the sliding bearing as well as its final design can be seen in Figure 4.

Figure 4: (A) The basic geometrical dimensions of the sliding bearing where h=10mm, b=150mm and l=70mm. The rounded corners cause the total surface area to be 9500 mm². (B) The piston and its stroke being equal to 50mm and its horizontal displacement being equal

Figure 5: (A) The FEM modelling of the inner roller with a pre-compression of 0.2 mm. Its occurring Von Mises stress is 545 MPa. (B) The FEM model of the waveband at 50 mm deflection. Its maximum occurring Von Mises stress is 910 MPa.

Since the waveband and inner rollers are fundamentally circular flexures, their occurring stresses can be predicted with high accuracy using a finite element approach. Both were modelled using Comsol Multiphysics with a geometrical nonlinear approach to determine its compliance behaviour. Both were modelled with a linear elastic material model using a Youngs modulus of 210E9 Pa. To transfer the motion as well as the forces from the wave generator ring to the piston, the inner rollers had to be pre-compressed. 0.2 mm compression created the desired friction between these components. The load cases and resulting Von Mises stresses of the individual modelling of both the waveband and inner rollers can be seen in Figure 5. The load capacity of the waveband at max deflection
of 50 mm is equal to 3000 N, which can be used as a validation of the FEM model. The material used to produce both components is heat treated 100Cr6 with a yield stress of 2500 MPa.

3.2 Roller bearing interface

Another fundamental approach to solving this interface problem is by using a roller bearing. The main limitation on the choice of bearing for the cylinder interface was the available dimensions. In this case a maximum design space height of 50 mm was available for the roller bearing. This has significantly limits the bearing diameter. The main advantage of this kind of interface is that the contact area is limited to a line contact, and therefore no additional mechanism is needed to convert the rotation into translation as was done for the sliding bearing. Roller bearings have also been widely investigated and lifetime calculations are in general reliable, and described in the ISO 281:2007 norm /6/.

3.2.1 Detail design: roller bearing

The choice was made to make use of the SKF NNCF 5004 CV, which is the largest double row cylindrical bearing available able to fit within the design space. This bearing has a basic dynamic load rating of 52.3 kN with a reference speed of 8500 RPM. The final cylinder interface design can be seen in Figure 6.

\[ L_{10h} = \frac{10^6}{60 \cdot n} \left( \frac{C}{P} \right)^{3/2} \]

Which is a basic life rating indication at 90% reliability /8/ in hours. Here, \( C \) equals the basic dynamic load rating of the bearing in Pa, \( P \) is equal to the load per bearing in Pa and \( n \) equals the rotational velocity of the bearing in RPM. The calculated lifetime at previously mentioned conditions is 40 hours, which is an accepted risk at this point of pump development. It however also directly shows the severe limitation of the roller bearing interface. The axial locking system was added to minimize the effect of axial rotations, which could drastically increase the load on a single bearing thereby shortening its limited lifetime even further.

4 Results

Both interfaces were prototyped either partially to test critical components, or in full. The results of their performance will be discussed in this section. In all cases no additional lubrication is used. The environment where this interface is part of will be constantly contaminated with seawater and as such, no stable lubrication can be used. The objective is therefore to find a interface that also functions without the use of lubrication.

4.1 Sliding bearing interface

For the sliding bearing interface, the following three cases were identified as being critical to its performance. These are as follows:

- Lifetime of the sliding bearing/waveband interface.
- Lifetime of the waveband
- Lifetime of the inner roller

These three cases will therefor all be discussed.

4.1.1 Sliding bearing material

To determine the wear rate of the sliding bearing/waveband interface a pin-on-disk measurement /4/ was performed for several high potential materials to determine their wear rate, change in friction coefficient and temperature change. The test setup can be seen in Figure 7. Two materials, AS PC04 and ZL1500T, were tested in a scaled loading condition. ZL1500T is a modified PEEK material, while AS PC04 is a carbon based composite material.

![Figure 6: Configuration of the roller bearing on top of the piston. The axial locking system prevents rotation around the axis.](image)

At 160 bar, the static load on the piston is 45.2 kN, assuming an equally distributed load on each roller bearing of 22.6 kN. Taking into account the dimensions of the WGR as well was the dimensions of the roller bearing gives a rotational velocity of 8100 RPM. An indication of bearing lifetime was calculated using the following equation:

The pin-on-disk measurement was performed for 90 minutes with a sample rate of 5 Hz to gain some insight on the desired properties. A load of 350 N was applied on the test samples with a surface of 6.25 mm\(^2\) to obtain an equal load condition as the full scale sliding bearing. The displacement measurements were filtered using a low-pass Butterworth filter. The disk is made from 100Cr6 with a hardness of 45 HRC and a roughness of 0.3 Ra. The friction coefficient was determined by the balance of load case and output of the torque sensor. The result of this measurement can be seen in Figure 8. The main issue with both materials is that the temperature does not reach a steady state temperature, which is destructive for sliding bearings /6/. The wear rate for the PC04 is also significant, causing it to fail in terms of lifetime. The main cause of this is the high velocity.

![Figure 7: (A) Pin-on-disk test setup used to determine the wear properties of the sliding bearing materials. A 100Cr6 disk is used as running track. The constant force mechanism guarantees that the pin is loaded with a constant load](image)
The 11th International Fluid Power Conference, IFK, March 19-21, 2018, Aachen, Germany

4.1.3 Inner roller lifetime

The inner roller lifetime is greatly dependent on its load case and the occurring stress level. Because the absolute deflection of the inner rollers is significantly smaller than for the waveband, it can still be modelled as a geometric linear model. Therefore, its lifetime is further investigated by modelling the inner rollers in the entire system with a finite element approach. The resulting load case can be seen in Figure 11.

Although the Von Mises stresses are significantly higher but still acceptable, it's the circumferential stresses causing the very limiting lifetime. The circumferential stress range occurring at the inner rollers exceeds 2000 MPa. Since these are not contact stresses, this is not an allowable stress range. This severely limits their lifetime and therefore not making it a suitable mechanism for the desired loading conditions. What should also be noted is that the pressure on the sliding bearings is significantly lower with only a max of 27.64 MPa, instead of the design value of 56 MPa. This is caused by the compliance of the waveband and inner rollers.
4.2 Roller bearing interface

Since the roller bearings are standardized components and the lifetime calculations determined a very limited but critical lifetime, the choice was made to make a simple test setup to validate this. The test setup can be seen in Figure 12.

Figure 12: (A) Test setup used to test performance of the cylindrical roller bearing. (B) close up of running surface and the method of actuating the bearing onto the running surface. (C) wear damage caused by the cylindrical bearing on the running surf.

The test setup is designed such that both the pressure on the roller bearings as well as the revolution velocity of the WGR can be achieved. The Running track is made from 100Cr6 with a hardness of 58 HRC. The hydraulic piston is used to apply an external load up to 50,000 N.

The roller bearing was initially loaded at 5000 N, which is 10% of max capacity and the running track was slowly increased to its max velocity of 18 m/s which corresponds to an actuation velocity of 900 RPM of the running track. Every interval increase of 100 RPM was performed for 300 seconds. It was found that at a load of 5000 N with a rotational velocity of 900 RPM, the bearing both caused destructive damage to the internal cylindrical rollers of the bearing. This was seen by steel dust being emitted from the internal rollers, meaning destructive failure of the inner cylinders. Internally the roller bearing was not up to the specified requirements and performed worse than its expected calculated lifetime. Additionally, the loads were so significant that permanent damage was caused on the running surface as seen in figure 12.C.

5 Conclusion & outlook

DOT is developing a low-speed positive displacement pump for seawater. Arguably the most critical interface of this pump is where the conversion from rotary drive motion to linear plunger motion takes place. Two solutions for this interface have been developed and tested, namely specific types of slider and roller bearings. For the slider bearing both the speed and stress were too large. For the roller bearing the load also proved far too great for the allowable bearing diameter.

By not having a rigid body mechanism between the wave generator ring and piston, the piston is subjected to a shear load. The effects of this phenomena on the piston and its lifetime will be discussed in future work.

Since both solutions have proven to be inadequate an alternative is required. Therefore the next step is to develop a hydrostatic bearing /8/. Although such a bearing would introduce additional volumetric losses, mechanical losses are less, and wear is minimal. A significant challenge for this particular hydrostatic bearing is the required material compliance to follow the shape of the ellipsoidal running track.

6 Acknowledgements

The research presented in this paper was part of the DOT500 ONT project, which was conducted by DOT in collaboration with the TU Delft and executed with funding received from the Ministerie van Economische zaken via TKI Wind op Zee, Topsector Energie.

References


Free gas in a hydraulic system is usually accompanied by negative aspects. Currently available models usually underestimate degassing at liquid-gas interfaces that are exposed to fluid flows, which is the most relevant degassing mechanism in hydraulic systems. Therefore, a new approach for physical modelling of bubble formation at liquid-gas interfaces is presented. Based on recent findings on diffusion-driven nucleation a simple model to calculate the mass fraction of gas being set free in a hydraulic fluid is derived. This approach is experimentally validated and could be implemented in available calculation tools.

Keywords: Cavitation, oil hydraulics, degassing, diffusion-driven nucleation

Target audience: Mobile Hydraulics, Plant Design, Numerical Simulation

1 Introduction

The formation of gas bubbles in hydraulic fluids is usually accompanied by negative aspects. Dispersed gas opposes the secure and efficient operation of hydraulic machines. Some possible consequences of dissolved gas in hydraulic systems are a higher compliance, speed up of oil ageing, cavitation and deterioration of heat flux. As the hydraulic fluid is an essential and often underestimated machine element, it is an engineer’s task to make this element calculable.

Degassing is mainly driven by three different mechanisms:

1. permeability of connecting elements,
2. inclusion of air during assembly, and
3. degassing of the working liquid.

By using construction guidelines and norms developers, facility designers and users are able to reduce negative aspects. While the first two mechanisms can be traced back to incorrect assembly or faulty constructions, degassing is a physical effect which cannot be avoided. Thus, degassing needs to be considered in an early design phase of components and facilities. Faulty design due to insufficient modelling of degassing processes in numerical simulations is usually linked to time- and cost-intensive iteration loops in the development processes. The advantage of numerical simulations becomes a disadvantage due to the insufficient quality of the results.

In order to predict the dynamic performance of the hydraulic system calculation methods are necessary. Currently available models describing degassing processes in hydraulic systems are based on 0D-modelling. They include empirically determined parameters, which need to be calibrated for each use case. These models are state of the art and widely used in the industrial context, even though they do not take degassing processes into account. Aside from 0D-modelling the application of commercial solvers (CFD) including cavitation models is possible. Yet, they do not provide validated models describing the degassing process.

1.1 Preconditions for cavitation and degassing

Cavitation occurs when the pressure falls below a critical pressure, called Blake’s threshold pressure. Blake’s threshold pressure is derived from a stability analysis of a spherical gas bubble. This bubble is called a nucleus. The analysis leads to an eigenvalue problem and finally Blake’s threshold pressure, cf. /19,24/. The derived pressure is always lower than the vapour pressure of the fluid. Concluding from this, fluids are able to absorb tensile stresses. Thus, the necessary requirements for cavitation to occur are a pressure level below the critical pressure and the presence of a nucleus, cf. Figure 1.

In a degassing process the critical pressure is the saturation pressure of the fluid. If the local pressure is decreased below the saturation pressure, the local concentration of dissolved gas is higher than the gas concentration in an equilibrium. The fluid is supersaturated. Based on Henry’s law the pressure in the fluid and the Henry coefficient, one can conclude that degassing processes only occur when the supersaturation of the fluid is greater than zero. Thus, is a necessary condition for degassing. As for the sufficient conditions, there are two possibilities. First, a sufficient supersaturation to form free bubbles in the liquid from the metastable phase due to the movement of gas molecules, which is called homogeneous nucleation theory. Usually this theory is of minor importance in technical applications, as the necessary supersaturation of can not be reached. The second possibility is the existence of a liquid-gas interface which enables molecular mass transport. This has been investigated at TU Darmstadt in the last years. In this case the entrapment of gas in minor crevices, so-called surface nuclei, is of importance.

It is important to note, that in contrast to cavitation, vapour pressure is of minor relevance for degassing processes. There are attempts to include both degassing and cavitation in a unified theory. Yet, there are doubts whether cavitation can be a real cause of degassing or whether it acts as a reinforcement. The time step limiting degassing is the diffusion rate. This also needs to be considered in the theory of /14/.

1.2 Diffusion-driven nucleation

The necessary condition for degassing is a supersaturated fluid and the presence of a liquid-gas interface. In hydraulic components there are usually no free (liquid) surfaces, so the degassing potential is low. Yet, nuclei within the liquid and gas entrapped on surfaces cannot be avoided in technical applications. This is where degassing takes place. The nuclei contribute to the process in different ways. Dispersed gas bubbles and free floating particles are carried by the fluid, so they reside only for a short period of time in the locally supersaturated section. Hence, the amount of gas which can diffuse into the nucleus is limited. The degassing potential depends on the number of nuclei and their specific surface.

Consequently, degassing processes can be mainly observed at gas cavities in the walls limiting the flow, cf. Figures 1 and 2. Gas cavities in cavities and cracks or attached to surface roughness elements, steps or drill holes serve as surface nuclei and allow the diffusion of gas that is solved in the liquid. The surface nuclei grow and free bubbles detach when a critical size is reached, cf. Figure 1. The critical size depends on the geometry of the corresponding surface nucleus. The bubble detachment triggers the self-exiting process. In technical applications “new” supersaturated fluid constantly streams along the surface nuclei, so the process continues for a long time – in fact as long as the liquid is supersaturated. Only when the flow comes to rest, e.g. after turning off the hydraulic machine, an equilibrium will be reached and the degassing stops.
The diffusion mass flux of solved gas into the surface nuclei and the following detachment of free gas bubbles is called diffusion-driven nucleation. It represents the most important mechanism of degassing in technical applications. The surface of the “produced” gas bubbles serve as additional liquid-gas interfaces. Free bubbles may also attach to roughness elements or steps and thus can act as surface nuclei. Thus, a large amount of gas can be set free. Figure 2 shows typical nucleation sites one can find in technical applications (drill holes, roughness elements) and their corresponding generic model (surface nuclei, steps).

There are only few scientific articles that focus on the degassing processes in technical fluids. Bubble growth and detachment has been mainly investigated in quiescent liquids, cf. /15-17/. Yet, in technical applications the impact of the flow area on the mass flux and bubble detachment needs to be taken into consideration. Therefore, Peters and Honza /22/ created an experiment which allowed the investigation of nucleation in crevices. Blind holes with an inner diameter of 0.6 mm and 0.8 mm serve as nucleation sites. The authors found out that the nucleation rate, i.e. the frequency of bubble detachment, depends on the shear rate at the wall. Yet, they did not formulate a functional dependency. Nucleation rates of 1 Hz to 10 Hz were measured. Groß and Pelz enhanced the experimental setup from Peters and Honza to investigate nucleation and bubble detachment in more detail. They managed to reach nucleation rates of up to 1000 Hz in silicon oil, cf. /8-10/.

2 Modelling of degassing processes in technical fluids

In most cases it is argued that diffusion processes are too slow to have an impact on cavitation phenomena. This holds true for the bubble collapse. During bubble formation, diffusion-driven processes have to be taken into consideration, as Groß and Pelz showed /11, 20/. In this paper a new approach to estimate the molecular mass transport being set free in a fluid in motion is presented, cf. /7/.

Fick’s first law of diffusion states that the mass flux of gas being dissolved in a liquid and transported across the phase boundary layer is given by

$$ m = -D \frac{\partial c}{\partial y} + \text{v} \cdot \nabla c = \nabla \cdot (D \nabla c), $$

where $M$ is the molecular mass of the gas, $D$ the diffusion coefficient, $c$ the concentration of the gas solved in the liquid and $\text{v}$ the vector normal to the surface $A$. Hence, the mass flux is proportional to the concentration gradient at the boundary layer. In order to obtain a high mass flux there needs to be either a large surface area or a high concentration gradient on the boundary layer. Since the concentration gradient cannot be measured directly, its extent is probably underestimated in most cases, leading to the misjudgement of diffusion processes as described before.

The concentration field in the liquid satisfies the advection diffusion equation

$$ \frac{\partial c}{\partial t} + \text{v} \cdot \nabla c = \nabla \cdot (D \nabla c), $$

which yields the concentration gradient. Figure 3 shows two common configurations for degassing in hydraulic systems. The concentration of dissolved gas in the liquid is $c_w$. The concentration determined by the local pressure at the liquid-gas interface is $c_a$. Hence, the supersaturation can be calculated with equation (2).

On the left side of figure 3 a surface nucleus with an inner diameter $d$ is shown. In this case the diameter is the characteristic length. If the surface nuclei are much smaller than the length of the components (pipe diameter, gap height) the velocity field $u(y)$ can be linearized near the wall, $u(y) = \gamma y$, with wall shear rate $\gamma$. This approximation is also valid for turbulent flows, if the concentration boundary layer lies within the viscous sublayer of the flow. Usually the concentration boundary layer is much smaller than the boundary layer of the flow, so the condition is usually satisfied (high Schmidt-Numbers $Sc = \nu D \gg 10^3$).
The results of the degassing mass flux (13)
\[ \frac{\dot{m}}{\dot{m}_o} = \frac{Q}{\rho_u M} = \frac{Q}{\rho_u h M}. \] (8)

\( Q \) represents the flow rate, \( \rho_u \) the saturation pressure of the liquid and \( c_o \) the concentration of the dissolved gas in the liquid. The ratio \( \zeta \) describes the mass fraction of the dissolved gas being set free while flowing through the respective component. The volume fraction can be determined from the mass fraction. It depends on the local pressure \( p_a \) and the saturation pressure \( p_c \). The volume fraction characterizes the amount of air in relation to the volume of the liquid and is given by
\[ \varphi = \zeta \Lambda \frac{p_a}{p_c}. \] (9)

Here \( \Lambda = R T \cdot \chi \) is the dimensionless solubility (\( \chi \) is the Ostwald coefficient) with the universal gas constant \( R \) and the temperature \( T \). In the following the solutions for a boundary layer overflown by a homogeneous velocity field is used, cf. equation (7). The solution is an upper boundary of the degassing mass flux. The considered boundary layer has the length \( l \) and the depth \( b \).

The degassing max flux
\[ \dot{m} = \frac{2}{\sqrt{\pi}} c_o M DB \left( \frac{U}{Q} \right)^{1/2} = \frac{2}{\sqrt{\pi}} c_o M DB \rho_u^{1/2}. \] (10)
yields the mass flux ratio
\[ \zeta = \frac{2}{\sqrt{\pi}} \frac{\Lambda \left( \frac{DB b^3}{Q^2} \right)^{1/2}}{\sqrt{\pi} \chi + 1} \quad \text{and the volume flux ratio} \]
\[ \varphi = \frac{2}{\sqrt{\pi}} \Lambda \left( \frac{DB b^3}{Q^2} \right)^{1/2} = \frac{2}{\sqrt{\pi}} \chi \frac{DB b^3}{Q} \rho_u^{1/2}. \] (12)

Mass fraction and volume fraction increase with increasing supersaturation \( \zeta \) and flow velocity \( U \). An increasing volume flux decreases both parameters. Usually there is a functional dependency between the flow velocity near the liquid-gas interface \( U_o \) and the volume flux \( Q \).

If \( b \) is the circumference of a circular flow cross section (e.g. a pipe or nozzle), so the whole circumference surface is covered by a liquid-gas interface, the volume flux is given by \( Q = Ub^2/(4 \pi) \).

Inserted in equations (11) and (12) yields
\[ \zeta = \frac{2}{\sqrt{\pi}} \Lambda \left( \frac{DB b^3}{Q^2} \right)^{1/2} \quad \text{and} \quad \varphi = \frac{2}{\sqrt{\pi}} \Lambda \left( \frac{DB b^3}{Q^2} \right)^{1/2}. \] (13)

On first sight it may be remarkable, that mass fraction and volume fraction of the free gas decrease with increasing flow velocity, respectively Péclet-Number. For a constant volume flow \( Q \propto U b^2 \) the mass fraction and volume fraction only depend on the supersaturation \( \zeta \), the diffusion coefficient \( D \) and the characteristic length of the boundary layer \( l \). Additionally, the volume fraction depends on the solubility \( \Lambda \).

The supersaturation \( \zeta \) itself is also a function of the flow velocity, as an increasing flow velocity decreases the local static pressure and consequently the local saturation concentration \( c_o \). The ratio \( l/b \) in equation (14) can be
Degassing of fluids in technical systems is an important objective due to its extensive negative impacts on the efficient and secure operation of hydraulic components. Currently available modelling approaches enable the estimation of degassing in tanks but fail when it comes to formation of bubbles near boundary layers. The formation of bubbles near boundary layers between liquid and gas, so-called nucleation sites, is the most common and therefore most relevant degassing mechanism in hydraulic systems.

The present paper presents new findings on degassing in tanks but fail when it comes to formation of bubbles near boundary layers. The formation of bubbles near boundary layers between liquid and gas, so-called nucleation sites, is the most common and therefore most relevant degassing mechanism in hydraulic systems. The present paper presents new findings on degassing in hydraulic systems and highlights the relevance for technical applications. Based on the presented degassing model calculations of the mass flux ratio and volume flux ratio may be performed. Contrary to available 0D-models no empirical parameters are used. If the model is implemented by software developers, future design calculations may be performed in more detail. The application of the degassing model optimizes systems understanding and enables the development of optimized components and systems.
### Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Dimension</th>
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<tr>
<td>A</td>
<td>Surface area of the boundary layer</td>
<td>( L^2 )</td>
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<tr>
<td>b</td>
<td>Depth of the problem</td>
<td>( L )</td>
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<tr>
<td>c</td>
<td>Gas concentration of the fluid</td>
<td>( L^2 N^{-1} )</td>
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<tr>
<td>( c_p )</td>
<td>Gas concentration at the boundary layer</td>
<td>( L^2 N^{-1} )</td>
</tr>
<tr>
<td>( c_s )</td>
<td>Saturation concentration of the gas in the liquid</td>
<td>( L^2 N^{-1} )</td>
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<td>( c_m )</td>
<td>Gas concentration in the surrounding fluid</td>
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<tr>
<td>D</td>
<td>Pipe diameter</td>
<td>( L )</td>
</tr>
<tr>
<td>( D )</td>
<td>Diffusion coefficient</td>
<td>( L^2 T^{-1} )</td>
</tr>
<tr>
<td>d</td>
<td>Diameter of the crevice</td>
<td>( L )</td>
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<tr>
<td>( H )</td>
<td>Henry-Coefficient</td>
<td>( L^3 M^{-1} Y^2 N )</td>
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<td>I</td>
<td>Characteristic length of the boundary layer</td>
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<td>M</td>
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<td>( p_s )</td>
<td>Saturation pressure</td>
<td>( L^3 MT^{-2} )</td>
</tr>
<tr>
<td>( P_m )</td>
<td>Pressure of the liquid</td>
<td>( L^3 MT^{-2} )</td>
</tr>
<tr>
<td>Q</td>
<td>Volume flux</td>
<td>( M^3 T^{-1} )</td>
</tr>
<tr>
<td>( R )</td>
<td>Universal gas constant</td>
<td>( L^3 MT^{-2} N^0 )</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
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</tr>
<tr>
<td>( t )</td>
<td>Time</td>
<td>( T )</td>
</tr>
<tr>
<td>( \vec{n} )</td>
<td>Velocity field</td>
<td>( LT^{-1} )</td>
</tr>
<tr>
<td>U</td>
<td>Velocity homogenous flow</td>
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</tr>
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</tr>
<tr>
<td>( \gamma )</td>
<td>Coordinate</td>
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</tr>
<tr>
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<td>( \nu )</td>
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</tr>
<tr>
<td>( \xi )</td>
<td>Mass fraction</td>
<td>1</td>
</tr>
</tbody>
</table>

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An Approach to Wear Simulation of Hydrostatic Drives

to Improve the Availability of Mobile Machines

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Wear in swashplate type axial piston pumps mainly occurs in three tribological contact pairs. These are swashplate-slipper, piston-cylinder and cylinder-block-valveplate. This article focuses on a simulation model, based on the approach of Archard and Fleischer, to predict the wear in the piston-cylinder contact. Besides general geometric data, the exact piston and cylinder contours and the wear-induced material removal over time are taken into account. A special focus in the simulation is on the investigation of the dependency of the viscosity of the hydraulic fluid on the wear. First results from test runs demonstrate a good correspondence between the simulation and measured wear on a test bench.

Keywords: swashplate type axial piston pump, piston-cylinder contact, fluid film, wear simulation, influence of viscosity, experiment

Target audience: pumps, tribology, mobile machines

1 Introduction and Motivation

The availability of mobile machines is gaining importance due to seasonal operation. An important business sector in this respect is municipal service. An unforeseen failure and the related downtime of vehicles in the high season is associated, among other effects, with a reduced fulfilment of summer and winter road maintenance. Consequences are significant disabilities and increased hazards in public road traffic. One of the best-known vehicle representatives in this sector is the Mercedes-Benz Unimog as an equipment carrier. An increased system pressure up to 500 bar in the hydrostatic drive system of the Unimog /16/, the operating points selected for volumetric efficiency improvement (high swashplate angles) and the application-driven temperatures lead to a higher load to the installed swashplate type axial piston pump (STAPP), in particular to the piston-cylinder contact pairs of the pump.

Operating the machine until standstill and reacting to the machine failure is a rather suboptimal approach. If maintenance intervals were adapted to the current machine states and planned times of use of the machine, unplanned failures could be minimised. The current machine state can be determined in two ways /2/ /6/:

- condition-based maintenance: machine state is analysed on the basis of state data
- load-based maintenance: machine state is predicted using the suitable models to calculate the damage out of past loads.

A major advantage of the load-based maintenance strategy is the possibility of an estimation of the machine condition, long time before a change in the state of the system is measurable. This allows the operator of a machine to schedule maintenance timetables at an early stage and to adjust them to times of deployment. This article focuses on a simulation model to predict the wear in the piston-cylinder contact for static operating points and is therefore a contribution to the use of load-based maintenance in vehicles with hydrostatic drive systems in the future.

2 Current State of Research

2.1 Lubrication Gap Simulation

The calculation of tribological contacts in STAPPs has been in the focus of many researches since the works of Kolk /20/, Renius /27/ and Böinghoff /1/. One of the most important factors for describing leakage, friction and wear intensity is the lubrication gap width (LGW) at every position in a tribological contact pair. The thickness of the fluid film can be affected by many coefficients like pressure, fluid-flow, temperature, pump speed, surface as well as the dimension and material properties of the friction pair. The available computational power has led to the fact that complex, numerically based models have become increasingly important for simulating these lubrication gaps (LGs) of STAPPs in the last two decades. The simulation programs CASPAR /21/ /24/ /25/ /29/, PUMA /5/ /9/ /19/ /28/ /30/ and SiKoBu /14/ are representatives for the simulation of the piston-cylinder contact. All these simulation programs calculate the state values in the lubricating gap, e.g. LGWs, pressure fields and friction conditions with respect to the above-mentioned influencing variables.

In this paper, Gels’ simulation program SiKoBu is used for all simulations of the LG between the piston and the cylinder, i.e. local pressure, deformations, local LGW.

2.2 Wear Simulation in Hydraulic Pumps

In the past some research work has already concerned with the wear investigation of hydrostatic pumps. Most of these projects have had the goal of improving the wear behavior of the components by means of geometrical, manufacturing or material-technical changes in the contact pairs /14/ /7/ /23/ /10/. In a current research project /13/, Ivantsyn sets different operating points of a STAPP into relation to the friction and temperature behavior and thus determines unfavorable operating points of the hydrostatic unit.

The mentioned experiments are focused on LGWs, contact pressures, heat generation, elastic and plastic deformation due to the fluid-structure interactions. These quantities are linked to wear volume but the dependence on wear mechanism is complex to establish. Therefore, laws of wear have been developed to predict wear and correlate them to experimental results. Nevertheless, wear is rarely simulated in STAPPs or analogue systems. In/3/ Chang proposes a wear model based on Archard and implemented testing to conduct comparison validation. Further on, Ma developed a wear model for the contact pair swashplate-slipper for a STAPP /22/. This model bases on the abrasive wear model of Stolarski and Zou for lubricated contact points, which is, however, only conditionally valid from experience /18/. Similar simulation approaches can be found for the cylinder-block-valveplate contact of a STAPP /31/ and for gear pumps /12/ /13/.

This literature review shows that few wear analyses are performed on real components of STAPPs. More generally, there is a lack of qualitative analyses aiming to understand wear mechanisms. A better knowledge of the mechanisms enables to choose more accurately wear prediction models. To date, the authors are not aware of any publications on wear models for the piston-cylinder contact.

Therefore, this paper attempts to use known wear theories according to Archard /26/ and Fleischer /1/ on the piston-cylinder contact and to validate these by means of tests. The focus of this paper is to expand the validated and approved model of Gels /14/ by incorporating a wear model, which is able to describe local amount of wear due to predetermined operating points over time.

3 Overview of the Simulation Scheme

The parameters working pressure $p_{\text{high}}$, pump speed $n_{\text{pump}}$, swashplate angles $\alpha$, fluid properties (kinematic viscosity $\nu$, fluid temperature $T$) and the contours of the cylinder $c_{\text{meas,star}}$ and the piston $P_{\text{meas,star}}$ are...
forwarded to and processed by SiKoBu. SiKoBu calculates the local pressure, the LGW and friction forces $F_{\text{friction},i}$ for each fluid node $i$ and each time step of one revolution. The simulation discretises the cylinder and uses the result of SiKoBu to calculate the wear height due to material removal $a_i$ of any node at every time step using the wear models explained in 3.2.

As long as the simulation height of each node stays under a determined boundary level, the material decrease due to wear of $1$ revolution is added to the total wear height $a_i$. If the wear height at one node exceeds the boundary level $a_{\text{max,bd}}$, the resulting contour of the cylinder $C_{\text{sim}}$ is being processed in SiKoBu. By this approach, if the simulation reaches or exceeds the total wear height of one or more nodes exceeds a defined maximum value, the simulation ends and the new cylinder contour $C_{\text{sim,end}}$ as well as the total wear volume $V_{V,tot}$ are calculated.

Figure 1 shows the basic structure of the developed simulation model.

The main parts of the simulation program are described in more detail below.

### 3.1 SiKoBu – Lubrication Gap Simulation

SiKoBu /14/ is used in this work to calculate the kinematics and the forces occurring in the LG of the piston-cylinder contact. For this purpose, several revolutions of the piston are simulated for one piston-cylinder contact. The piston is subdivided into so-called disk-notes (DN) and the lubricating film into lubricating-film-nodes (LFN) as well as lubricating-film-elements (LFE), which connect individual LFNs to one another.

The main parts of the simulation program are described in more detail below.

#### 3.2 Analytic Wear Models

For the simulation described in this paper two different wear models are used. The model of Fleischer is used to calculate the wear due to liquid friction. To calculate the wear due to solid friction the model of Archard was used. The normal contact force $F_{\text{contact},i}$ needed for the model of Archard is easier to describe than the friction force $F_{\text{friction},i}$ needed in the model of Fleischer since the friction coefficient is not relevant.

The wear model of Archard was originally developed to calculate the wear due to adhesion but is also effectively used in modelling of abrasive wear. It uses the proportionality of the wear volume $V_{V}$ to the load and the inverse of the hardness of the softer material $H$. The load is calculated as the product of the normal force $F_{N,\text{contact},i}$ and the friction path $s_R$ /32/:

$$V_{V,i} = k_{ad} F_{N,\text{contact},i} s_R^{-1}$$

The wear coefficient $k_{ad}$ cannot be calculated theoretically but has to be determined experimentally /4/. It depends on the material combination of the friction bodies /26/.

The wear model of Fleischer is an energetic wear model. It bases on the idea, that some energy is needed to form wear particles. According to Fleischer energy is supplied to the surface at each friction contact. Most of the energy dissipates into heat, while the rest of the energy effects lattice imperfections /15/. When the accumulated energy of the lattice imperfections reaches a critical level, wear particles will be built /11/:

$$V_{V,i} = \frac{1}{e_r^{*}} F_{\text{friction},i} s_R$$

A theoretical determination of the apparent density of friction energy $e_r^{*}$ is not possible. Like the wear coefficient of Archard, it has to be determined experimentally /4/.

The model of Archard is used to calculate the wear due to solid state friction. Fleischer’s model is used to calculate the wear due to liquid friction. Since solid state friction has a much higher influence on the wear volume than liquid friction, the coefficient for liquid friction is set to a value known from literature ($e_r^{*} = 10^{9}$ J mm$^{-3}$ mm$^{-1}$ /4/) and only the coefficient for solid friction was determined experimentally in this research project. Both approaches base on the simplified assumption of a particle-free hydraulic fluid.

#### 3.3 Geometric Measurement

The creation of a realistic image of the cylinder contour is of great importance for the accuracy of the simulation results. With the aid of different geometrical measuring methods, a three-dimensional image of the cylinder is therefore created. For the derivation of the contour, 14 different measuring records are available for each cylinder:

- straightness measurements at four line positions which are 90 degrees offset to one another
- circularity measurements at five depths of the cylinder. These measurements are mainly located in the most wear-intensive points. Two measurements each at the outer edges of the cylinder (8%, 16%, 84% and 92% cylinder depth) and one at the centre of the cylinder (50% cylinder depth)
- Gaussian diameters measurements at five depths of the cylinder.
Because the measurements base on different measuring methods, the individual measurements, as mentioned above, are aligned with one another by a least squares fit. In addition, a 3D geometry with spatially equidistantly distributed points is calculated using the triangulation-based natural neighbor interpolation of the measurements. Figure 2 shows the derived contour in an enlarged and consumed view of one exemplary cylinder, highlighting the underlying measurements (black) and the geometry from the interpolation (collared). 0 % depth marks the side of the cylinder at the swashplate, 100 % the side towards the valveplate. In addition to the cylinder contour, however, the piston contour also has a considerable influence on the friction and wear conditions in this contact pair. In order to derive a suitable geometry for this component, the following measurements are carried out:

- straightness measurements at four line positions, which are 90 degrees offset to one another
- Gaussian diameters measurements at three depths of the piston.

Due to the fact, that over a longer period of time the piston is uniformly loaded by its own rotation over its entire surface area at the same depth, the measurement of the circularity is omitted. In addition, the rotating position of the piston at a specified point of time would not be unambiguously identifiable without installation of further measurement technology into the STAPP and thus cannot be mapped for a simulation over several operating hours.

Further, to use the geometries as an input for the simulation, in Section 6 the measurements are used to validate the simulation models. To determine the wear coefficient $k_{\text{ad}}$, cf. formula 1, the factor was determined by means of the method of least-significant error squares, so that the difference between the measured wear volume and the simulated wear volume of all the cylinders was minimised.

## 4 Simulation Results

### 4.1 Points of High Wear Intensity

The simulation results show that the highest points of wear are located at the front and the rear of the cylinder. This meets the expectations, since the highest wear occurs when piston and cylinder have solids contact. When the piston tips in the cylinder due to pressure forces, solids contact results at the rear and the front of the cylinder.

As shown in figure 3, wear at the side of the swashplate (0 % depth) is higher than wear at the side of the valveplate (100 % depth). The starting contour for the simulation is the representation of a standard cylinder without a significant rounding of the outer edges.

### 4.2 Effect Analysis

An effect analysis was executed to examine the effect of the parameters viscosity, hydraulic pump speed, pressure on the high pressure side of the pump and swash plate angle. The parameters were changed gradually and the wear volume was simulated. Eight stages of viscosity and three stages of pump speed, pressure and swashplate angle were simulated which makes a total of seventy two stages.

Table 1 shows the values of the parameters that were taken for the effect analysis.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Stage 1</th>
<th>Stage 2</th>
<th>Stage 3</th>
<th>Stage 4</th>
<th>Stage 5</th>
<th>Stage 6</th>
<th>Stage 7</th>
<th>Stage 8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kinematic viscosity $\nu$ [cSt]</td>
<td>1</td>
<td>3</td>
<td>5</td>
<td>7</td>
<td>10</td>
<td>20</td>
<td>30</td>
<td>40</td>
</tr>
<tr>
<td>Pump speed $n_{\text{pump}}$ [rpm]</td>
<td>900</td>
<td>1500</td>
<td>2100</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Pressure on high pressure side of the pump $p_{\text{pump}}$ [bar]</td>
<td>100</td>
<td>300</td>
<td>500</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Swashplate angle $\alpha$</td>
<td>9</td>
<td>15</td>
<td>21</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Figure 4 shows the results of the effect analysis for a kinematic viscosity $\nu = 1$ cSt and the various values for the other parameters. It can be seen that a higher pressure results in a higher wear volume. This is caused by the increased pressure forces, which lead to a higher probability of solids contact. An increase of the pump speed also results in an increase wear volume since the friction path is larger. Similar results are given for the swashplate angle. A bigger swashplate angle results in a higher wear volume. Not only do the individual parameters effect the wear volume, but also it strengthens the effect, that other parameters have on the wear volume. The swashplate angle for example has a higher effect on the wear volume, if the pressure is high and viscosity is low. Similar effects can be monitored for all of the parameters. The wear volume is given relatively to the highest wear volume reached in this simulation.
Figure 4: Exemplary effect analysis for a kinematic viscosity $\nu = 1$ cSt

The effect of the viscosity can be seen in figure 5.

Figure 5: Effect of fluid viscosity to wear intensity

A higher viscosity usually results in fewer wear volume. Is the viscosity too low, the hydrostatic bearing pressure cannot absorb the lateral force and the piston has to be supported by solid contact with the cylinder. An exception of the effect of the viscosity can be seen for high pump speed. The wear volume decreases for increasing viscosity until a minimum is reached and the wear volume increases again for higher viscosities. A possible explanation is the effect of the liquid friction, which increases with rising viscosity. Higher viscosity results in less solids contact. When the minimum is reached, the solids contact and thereby the solid state friction do not decrease anymore, but the liquid friction increases. Therefor the total of solid state friction and liquid friction increases.

5 Test Bench

A Unimog device of the Euro 5 generation, loaded by a four-wheel-acoustic roller dynamometer, is used as a test set-up to compare simulation results to caused wear on a test bench, see figure 6. This test set-up was chosen for the reason that loads that are derived from the field measurement data can be imprinted on the STAPP and that the rest of the drive train is not removed from the system with all the accompanying influences [8].

In addition to the necessary geometric measurement data described in section 3.3, further measuring systems are installed in the vehicle, which can be used during operation for condition-based maintenance. These include volume flow measurements installed in the main circuit (MC) for detecting the efficiency losses in the hydraulic circuit, as well as particle monitors in the flushing circuit (FC) for detecting the cleanliness level and roughly estimating a possible wear volume in a time interval.

6 Validation of the Simulation Model

In order to trust the simulation results, certain operating conditions will be validated using geometric measurements of one exemplary cylinder of the unit. The notation according to table 2 is selected for the respective TRs.

<table>
<thead>
<tr>
<th></th>
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<th></th>
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<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>2</td>
<td>465</td>
<td>19</td>
<td>1500</td>
<td>21</td>
<td>11.0</td>
<td>71</td>
<td>344</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>3</td>
<td>465</td>
<td>19</td>
<td>1500</td>
<td>20-21</td>
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<td>75</td>
<td>348</td>
</tr>
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<td>3</td>
<td>4</td>
<td>465</td>
<td>19</td>
<td>1500</td>
<td>21</td>
<td>4</td>
<td>75</td>
<td>348</td>
</tr>
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<td>4</td>
<td>5</td>
<td>465</td>
<td>19</td>
<td>1500</td>
<td>21</td>
<td>2.75</td>
<td>71</td>
<td>344</td>
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<td>5</td>
<td>6</td>
<td>465</td>
<td>19</td>
<td>900</td>
<td>21</td>
<td>2.75</td>
<td>71</td>
<td>344</td>
</tr>
</tbody>
</table>
As an example, TR No. 4 is examined in detail.

Figure 7 shows the comparison between the simulation (brown with leftward-pointing triangles) and the measurements (measurement before the TR: yellow with squares; measurement after the TR: violet with diamonds) on the four straightness records. The relative deviation from a standard cylinder $\Delta r_i$ is plotted on the ordinate, the relative depth of the cylinder is plotted on the abscissa. The relative deviation from the standard cylinder is defined as the difference from the measured radius of the cylinder $r_{\text{meas},i}$ and the radius according to the manufacturing dimensions $r_{\text{man},i}$ multiplied by a uniform scaling factor $z_{\text{scale}}$, see formula 3.

$$\Delta r_i = (r_{\text{meas},i} - r_{\text{man},i}) \cdot z_{\text{scale}}$$  (3)

The 75 % circumference plot shows the most wear-resistant position at the outermost point of the cylinder (depth 0 %). This can be explained by the kinematics of the mechanism and the acting forces in the high pressure area. In the range between 0 - 50 %, no wear-related material removal is visible over the entire cylinder length in this simulation, as well as in reality.

For the circularity measurements it can be seen, that the qualitative comparison between simulation and reality is in a good agreement, see Figure 8. At 75 % of the circumference, the cylinder has the highest wear in the required depth regions. Again, measurement and simulation confirm very low wear at the side of the valveplate (see depth 84 and 92 %).

Figure 8: Circularity measurements for different depths of the cylinder

For a more detailed investigation, the influence of different viscosities on the wear behavior is hereafter shown and discussed with reference to the above-described cylinder for the most wear-intensive regions, see figures 9 and 10:

- straightness measurement at 75 % of the circumference: 0 % to 50 % depth of the cylinder
- circularity measurement at 8 % of the depth: 25 % to 100 % of the circumference of the cylinder

The results from the TRs with near-typical viscosities show a relatively good overlap with the wear removal from the simulation. Hardly any wear was observed in the simulation as well as in the measurement. The frictional state at this time is largely on the side of liquid friction, but a solid contact rarely occurs. Consequently, the wear removal is very small. The comparison of measurement No. 1 and No. 2 suggests a slight displacement of the angular axis. The curves were not shifted to each other for the purpose of preserving scientific traceability.

With decreasing viscosity during the TRs, there is a stronger wear pattern with regard to the maximum wear at the outer edge of the cylinder, but also with respect towards the inside of the cylinder. On average, the comparison of the results of the simulation and the straightness measurements after the TRs shows similar wear characteristics. At some points the simulation predicts a higher wear on the cylinder than the measurement shows. This can be seen, in particular, in the simulated higher gradients of the trumpet-shaped contour of the cylinder. An incorrect deformation of the piston in the simulation, as well as the self-rotation of the piston in the cylinder, can be the cause of this increased material removal. Both parameters can unfortunately not be investigated under the current experimental setup.

At a circumference between 50 and 100 % the accuracy of the model is very good. The wear removal is predicted quite precisely, especially for decreasing viscosities; nevertheless, in the range between 50 and 75 % it can be seen
that the simulation’s outcome is shifted by a few degrees to the measurement results. This indicates an inaccurate representation of the piston pressure profile in the simulation and must be further investigated in the future.

In the further cycle, the material removal does not proceed any further due to a certain running-in state and thus is not observed any further in the theoretical view at this time. It is assumed that due to an imbalance in the engine or the transmission and the internal combustion engine connected upstream of the STAPP, wear occurs due to an interaction of the inertia and acceleration effects of the piston, as well as the lack of supporting structure. The simulation cannot depict this cause of wear and therefore cannot explain the occurring physical effects. In the further cycle, the material removal does not proceed any further due to a certain running-in state and this is not observed any further in the theoretical view at this time.

In TR No. 3, in the range between 25 and 50 % of the circumference, deviations of material removal are visible. It is assumed that due to an imbalance in the engine or the transmission and the internal combustion engine connected upstream of the STAPP, wear occurs due to an interaction of the inertia and acceleration effects of the piston, as well as the lack of supporting structure. The simulation cannot depict this cause of wear and therefore does not explain the occurring physical effects. In the further cycle, the material removal does not proceed any further due to a certain running-in state and this is not observed any further in the theoretical view at this time.

TR No. 5 shows a higher wear rate than the simulation predicted. The reason for this is suspected because of the increasing temperature field of the fluid film due to the slow speed of the pump, so that the lower cylinder axial speed and thus the reduced fluid flow in the LG.

For an evaluation of the quality of the simulation approach, a statistical evaluation of the relative deviations $\Delta C$ between simulation $C_{\text{sim,end}}$ and measurement $C_{\text{meas,end}}$ at each individual point of the cylinder is carried out, see formula 4.

$$ \Delta C = (C_{\text{sim,end}} - C_{\text{meas,end}}) \cdot z_{\text{scale}} $$

The evaluation is made using the median and the 5 % and 95 % quantiles. The median specifies the limit value at which exactly 50 % of all values are larger and 50 % of the values are smaller. 90 % of the deviations are between the 5 % and the 95 % quantile. The two most-wear-intensive areas (0 - 25 % depth and 75 - 100 % depth – both over the entire circumference) are analysed separately. Table 3 shows the evaluation for the respective measurement times.

<table>
<thead>
<tr>
<th>TR No.</th>
<th>0 - 25 % cylinder depth</th>
<th>75 - 100 % cylinder depth</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Median [%]</td>
<td>5 %</td>
</tr>
<tr>
<td>1</td>
<td>-1.8</td>
<td>-4.8</td>
</tr>
<tr>
<td>2</td>
<td>1.6</td>
<td>-2.8</td>
</tr>
<tr>
<td>3</td>
<td>0</td>
<td>-15.8</td>
</tr>
<tr>
<td>4</td>
<td>0.5</td>
<td>-3.4</td>
</tr>
<tr>
<td>5</td>
<td>-0.9</td>
<td>-7.7</td>
</tr>
</tbody>
</table>

The results from the comparisons of the simulated and measured surface contours are also reflected in the statistical evaluation of the model. Positive medians indicate a too high prognosis of wear, as for example TR No. 2 and No. 4, negative medians, e.g. No. 5, to a too low predicted wear. For the TRs No. 1, No. 2 and No. 4, 90 % of the data (range between the two quantiles) lie in a tolerance window of 10% of the deviation from the standard cylinder, defined in formula 3. The large tolerance band in the third experiment results from the unexpected effect of wear in the investigation range of 25 - 50% of the circumference at a low cylinder depth. This is confirmed by the absolutely considered extremes. The too low predicted wear for the last TR leads to the high negative deviation of the simulation model.

In general, it can be summarised that the simulation shows meaningful tendencies compared to the measurement. The wear behaviour as a function of the operating parameters, in particular of the viscosity, allows a comparison and a good qualitative statement about the wear behaviour. For a more accurate depiction of the wear and its causes, further measuring techniques, for example gap width and temperature sensors, are required.

### 7 Influence of Lower Viscosities and the Transferability to Typical Operating Conditions

Both in the influence analysis, cf. chapter 4.2, as well as in the validation of the simulation results, cf. chapter 6, it has been found that the wear increases exponentially with decreasing viscosity, cf. figure 4. Due to the TR time of 15 hours per TR, no wear has been noted for the operation typical viscosity. This also confirms the results from the simulation. Due to the deviations between measurement and simulation in the area of higher viscosities, the simulation tends to be too low. The Archard model was originally developed for solid contact friction and has its strengths here. The mixed friction area and the resulting wear at individual nodes could be described even better by adaptive wear coefficients depending on the viscosity, see formula 5.
# Summary and Conclusion

The call for reduced downtimes and higher availabilities of mobile machines makes monitoring and failure prediction of technical systems more important. This paper shows a research of a wear simulation for the piston-cylinder contact of a swashplate type axial piston pump in order to describe the state of these components.

After a brief introduction to previous works in this area, a method of the used wear simulation is introduced and exemplary results are shown. In an effect analysis the system variables pump speed, pressure, swashplate angle and the viscosity are related to the wear volume. The validation of the simulation results, by using the measuring results of a full-vehicle test bench, show a relatively accurate match in the wear-intensive points of this contact pair.

To reduce the mistakes made by combining different measurement setups by an interpolation method, it would be useful to perform 3D surface roughness measurements to characterise the worn surfaces. An adaptation of the simulation model to a non-isothermal simulation of the lubrication gap would explain temperature-heating effects due to low pump speed. Equipping a test set-up with a corresponding expanding measuring technique (lubrication gap width and temperature measuring sensors) would be necessary.

# Acknowledgements

The authors would like to appreciate the support of MOBIMA e.V.. Some of the results were produced under a cooperation project supported by this institution.

## Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a_{new,i})</td>
<td>Material removal for fluid node (i) for one turn</td>
<td>[μm]</td>
</tr>
<tr>
<td>(a_{wear,bd})</td>
<td>Boundary level of wear in simulation</td>
<td>[μm]</td>
</tr>
<tr>
<td>(a_i)</td>
<td>Total material removal for fluid node (i)</td>
<td>[μm]</td>
</tr>
<tr>
<td>(e_i)</td>
<td>Apparent density of friction energy</td>
<td>[J/mm²]</td>
</tr>
<tr>
<td>(k_1, k_2)</td>
<td>Statistic wear coefficient for expended Archard’s wear model</td>
<td>[-]</td>
</tr>
<tr>
<td>(k_{ad})</td>
<td>Wear coefficient for Archard’s wear model</td>
<td>[-]</td>
</tr>
<tr>
<td>(n_{pump})</td>
<td>Hydraulic pump speed</td>
<td>[rpm]</td>
</tr>
<tr>
<td>(P_{high})</td>
<td>Pressure on temporary high pressure side</td>
<td>[bar]</td>
</tr>
<tr>
<td>(P_{low})</td>
<td>Pressure on temporary low pressure side</td>
<td>[bar]</td>
</tr>
<tr>
<td>(r_{man,i})</td>
<td>Radius of the cylinder according to the manufacturing dimensions</td>
<td>[mm]</td>
</tr>
<tr>
<td>(r_{meas,i})</td>
<td>Measured radius of the cylinder</td>
<td>[mm]</td>
</tr>
<tr>
<td>(s_0)</td>
<td>Friction path</td>
<td>[mm]</td>
</tr>
<tr>
<td>(t_{sim,job})</td>
<td>Time to be simulated</td>
<td>[h]</td>
</tr>
<tr>
<td>(v_{40/C})</td>
<td>Kinematic viscosity for 40°C temperature</td>
<td>[cSt]</td>
</tr>
<tr>
<td>(z_{scale})</td>
<td>Uniform scaling factor</td>
<td>[1/mm]</td>
</tr>
<tr>
<td>(C_{meas,end})</td>
<td>Contour of the cylinder after the test run</td>
<td>[mm]</td>
</tr>
<tr>
<td>(C_{meas,start})</td>
<td>Start contour of the cylinder</td>
<td>[mm]</td>
</tr>
<tr>
<td>(C_{sim,end})</td>
<td>Interpolated contour of the cylinder after simulation</td>
<td>[mm]</td>
</tr>
<tr>
<td>(C_{sim})</td>
<td>Interpolated contour of the cylinder for use in simulation</td>
<td>[mm]</td>
</tr>
<tr>
<td>(\Delta C)</td>
<td>Relative deviation of the cylinder contour between simulation and measurement</td>
<td>[mm]</td>
</tr>
<tr>
<td>(F_{friction,i})</td>
<td>Friction force for fluid node (i)</td>
<td>[N]</td>
</tr>
<tr>
<td>(F_{n,contact,i})</td>
<td>Normal contact force for fluid node (i)</td>
<td>[N]</td>
</tr>
<tr>
<td>(P_{meas,start})</td>
<td>Start contour of the piston</td>
<td>[mm]</td>
</tr>
<tr>
<td>(P_{sim})</td>
<td>Interpolated contour of the piston for use in simulation</td>
<td>[mm]</td>
</tr>
<tr>
<td>(H)</td>
<td>Hardness of material</td>
<td>[N/mm²]</td>
</tr>
<tr>
<td>(T)</td>
<td>Temperature</td>
<td>[°C</td>
</tr>
<tr>
<td>(V_{\gamma,\text{tot}})</td>
<td>Accumulated total wear volume</td>
<td>[mm³]</td>
</tr>
<tr>
<td>(V_{\gamma,i})</td>
<td>Wear volume for fluid node (i)</td>
<td>[mm³]</td>
</tr>
<tr>
<td>(\alpha)</td>
<td>Swashplate angle</td>
<td>[rad]</td>
</tr>
</tbody>
</table>

## References


Investigation of Laser surface texturing for Integrated PV (pressure×velocity)-value-decreased Retainer in an EHA Pump

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An integrated retainer which wraps the slippers and rotates with them is assembled in an Electro-hydrostatic actuator (EHA) pump, to eliminate the high linear velocity at slipper bottoms and to diminish the PV (pressure×velocity) value of the contact area. The impacts of laser surface texturing on the performances of the high-speed rotating retainer are investigated by conducting the CFD simulation of the flow inside several micro-dimples and the experiments on an EHA pump prototype. Wear marks are observed and the dimples with an area ratio of 16.4% are found to improve the volumetric and mechanical efficiencies of the prototype by up to 7.4% at the speeds of 6000~10000 rpm.

Keywords: Electro-hydrostatic actuator pumps, laser surface texturing, integrated retainer, CFD simulation, sliding wear
Target audience: Aerospace Applications, Surface Modifications

1 Introduction

More Electric Aircrafts (MEAs) like the Airbus A380, replace partial hydraulic circuits with Electro-hydrostatic actuator (EHA) networks, due to their prominent advantages such as efficiency improvement, weight and space savings /1, 2/. The hydraulic power supply of an EHA is a high-speed and small-sized pump, and enhancing its reliability for extending the service life of MEAs still remains an area of strong research. The EHA pump utilized in aircrafts in this study is a double-direction variable displacement axial piston pump. It is worth noting that there is a new sliding interface instead of the slipper/swash plate interface due to the unconventional structure of the retainer as shown in Figure 1.

![Figure 1: The integrated retainer in the EHA pump.](image1)

The special retainer made of steel alloys is composed of three parts which are an upper plate, a side wall and a bottom plate. They are stacked up layer by layer and fastened together by nine screws. The nine slippers are wrapped in the retainer and drive it to rotate at the same angular speed with themselves. A retainer guide as well as the slippers at the high-pressure side presses the bottom plate towards a bearing plate which is made of copper alloys and fixed to the swash plate by a pin. The damping orifices of the bottom plate lead the pressurized fluid from the oil pockets of the slippers to those of the bottom plate, generating hydrodynamic and hydrostatic forces acting on it. Compared to the total bottom area of the slippers and the high linear velocities at the slipper bottoms in the traditional structure, the large contact area between the bottom and bearing plates, and the micro sliding motions of the slippers relative to the bottom plate, diminish their PV (pressure×velocity) values. However, this structure also has demerits of occupying more space and having a higher moment of inertia, resulting in dynamic unbalance and a slower response to a speed change.

The EHA pump is characterized as the high speed up to 20000 rpm and the broad range of operating conditions /3/. With the continued demand upon the EHA pump for higher speeds and pressures, the inertial forces of piston-slipper assemblies, the inclined swash plate and the bias load due to the pressure distribution jointly contribute to excessive retainer tilt. Under those conditions, the components are prone to be abraded, burned and even deformed as shown in Figure 2, resulting in a substantial decrease in the pump efficiency.

![Figure 2: Failed retainer’s bottom plate and bearing plates.](image2)

The laser surface texturing (LST) technique as a viable option of changing surface topographies has many advantages like reserving lubricant, trapping wear particles and increasing the load-bearing capacity /4/. This technique has already succeeded in improving the performances of various industrial applications, such as mechanical seals /5, 6/, bearings /7, 8/ and piston rings /9, 10/. Studies on applying the LST to industrial hydraulic power units have been devoted to improving the tribological performances of pistons and valve plates /11-15/. Since the effects of a given surface microstructure design are sensitive to the operating conditions, surface textures capable of reducing both the friction and leakage of the lubricating gaps need to be specifically designed for high-speed EHA pumps.

The present contribution compares the effects of the rectangle-shaped micro-dimples with different area ratios in the retainer’s bottom plate/bearing plate interface by means of the CFD analysis. The k-ε turbulence model was selected to simulate the flow inside several micro-dimples. Then, the surface texture pattern which shows the highest load-bearing capacity is fabricated on the bottom plate and tested on an EHA pump prototype in first trials. The pump performance parameters and wear marks of the component surfaces are compared with those of the untextured bottom plate. Using the simulation and experiment methods to optimize the texture geometry and arrangement will be part of future work.

2 CFD simulation of surface textures

2.1 Simulation model

The geometrical parameters of the rectangle-shaped micro-dimples intended to be fabricated on the retainer’s bottom plate are shown in Figure 3. The dimple area ratio $r_d$ is calculated using Equation (1), where $A_d$ is the total area of the micro-dimples and $A_t$ is the total contact area between the bottom and bearing plates.

$$r_d = \frac{A_d}{A_t}$$

(1)
A three-dimensional turbulence model based on the \(k-\varepsilon\) model considering the cavitation phenomenon and the wall roughness effects is solved by using commercial CFD simulation software ANSYS FLUENT. The properties of the 15# hydraulic oil used in the simulation as well as the experiments are listed in Table 1. According to the quality inspection report provided by the supplier (Kunlun Lubricant), the oil density was measured at the temperature of 20 °C. The film thickness is held constant and set equal to 1 μm which approaches the level of the surface roughness, in order to evaluate the effects of LST under poor lubrication conditions. Because of the extremely thin oil film, it will spend too much computational time if the entire friction area between the bottom and bearing plates is generated a high-quality mesh. Thus, only a small area as shown in Figure 3 is simulated. The studied area covers several micro-dimples to consider the interaction between neighbouring dimples.

### Table 1: Properties of the 15# hydraulic oil used in the simulation and experiments.

<table>
<thead>
<tr>
<th></th>
<th>Kinematic viscosity (mm(^2)/s)</th>
<th>Density (kg/m(^3))</th>
<th>Flash point (°C)</th>
<th>Pour point (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>40 °C</td>
<td>13.2</td>
<td>100 °C</td>
<td>20 °C</td>
<td>837.4</td>
</tr>
<tr>
<td></td>
<td>4.90</td>
<td>20 °C</td>
<td>83 °C</td>
<td>82</td>
</tr>
<tr>
<td>13.2</td>
<td>4.90</td>
<td>837.4</td>
<td>82</td>
<td>-60</td>
</tr>
</tbody>
</table>

The Schnerr and Sauer cavitation model /16/ is used and the vaporization pressure is specified as the deaeration pressure of dissolved air which is assumed to be 6650 Pa /17/ because most of cavitation is due to dissolved air at low pressure /18/. The bubble number density is set as \(10^{10}/m^3\) which was estimated using the expression of the vapor volume fraction /16/. The case drain pressure of 0.06 MPa is imposed on the inlet and outlet surfaces. The untextured wall moves with a constant translational speed and the textured wall is stationary. Besides, the roughness parameters are set for the wall boundary conditions. A roughness height of 2 μm due to the thermal damage caused by the laser processing is imposed on the bottoms of the micro-dimples, while the rest walls have a roughness height of 0.2 μm. The roughness constant is kept at the default value. The Coupled algorithm is used for the pressure–velocity coupling and the PRESTO! (PREssure STaggering Option) discretization scheme is adopted for interpolation of pressure gradient. Moreover, the QUICK scheme is applied for the momentum and volume fraction equations.

Aiming to investigate the impacts of the dimple area ratio, textured surfaces with area ratios of 16.4%, 24.6%, 32.8%, 41% and 49% are simulated at the high speed of 10000 rpm (equal to the linear velocity of 30 m/s with a turning radius of 28 mm).

### 2.2 Simulation results

The static pressure contours of the textured wall and the cross section in an X-Z plane are shown in Figure 4. The pressure build-up in the converging regions and the pressure drop in the diverging regions are noticeable. The high linear velocity leads to cavitation in the diverging regions of all the dimples. As the dimples are laid out more compactly, the pressure increment becomes smaller and the cavitation area is enlarged, because the hydrodynamic effect of one dimple is weakened by its adjacent dimple. The flow velocity contours of the cross section are shown in Figure 5. Each dimple has an eddy near the untextured wall and the velocity fields of the dimples in one lubricating gap are quite similar before the area ratio exceeds 24.6%. In the lubricating gap with a high area ratio, the eddy region is enlarged as the dimple approaches the outlet surface.
The viscous friction force and load-bearing capacity of the untextured wall are depicted in Figure 6. The textured surfaces with an area ratio of more than 24.6% reduce the bearing capability compared to that of the untextured surface, and the area ratio of 16.4% generates the highest bearing load in this case. It is also found that the higher area ratio provides a higher average gap height, resulting in a lower shear gradient, so that the shear stress decreases almost linearly as the area ratio increases.

3 Experiments of textured retainer

3.1 Preparation of retainer

The rectangle-shaped micro-dimples with an area ratio of 16.4% generate the highest bearing load among the five area ratios in the simulation, so that they were fabricated on the retainer’s bottom plate using a Q-switched Nd:YVO4 laser with a wavelength of 355 nm, a maximum output power of 5 W and a frequency of 30 kHz. The textured bottom plate and the dimple profiles are shown in Figure 7. Molten materials are accumulated around the dimple rims and the height of the bulges is about 8 microns. The bulges were not removed to investigate their influences on the wear behaviour. The surface roughness of the dimple bottom was measured using a confocal laser scanning microscope (CLSM). The arithmetical mean deviation of the assessed area (Ra) is about 2 microns. There were also trenches with a depth of about 5 microns at the bottom edges. Before test, the textured retainer was cleaned in an ultrasonic acetone bath to remove the contaminants.

3.2 Experimental procedure

The EHA pump prototype and the high-speed test rig are shown in Figure 8. The prototype is driven by a servo motor which can be continuously adjusted from 0 rpm to 16000 rpm. The pressure and temperature values at the outlet, inlet and case drain ports of the prototype are obtained from the sensors mounted on the manifold which is connected to the ports. The total leakage of the prototype is measured by a flowmeter. The working fluid is pumped to the prototype inlet by a charge pump. Two proportional relief valves locate on the delivery lines of the charge pump and the prototype, separately.

As shown in Figure 9, the textured retainer reduces the leakage by 0.88–1.17 mL/r and the output torque by 0.55–0.68 Nm in first trials. The growth ratios of the pump efficiencies are calculated using Equation (2), where Value is the volumetric or mechanical efficiencies. As illustrated in Figure 10, both the volumetric and mechanical efficiencies of the prototype equipped with the textured retainer are higher than the untextured one at all the speeds, but the growth ratios become small as the speed increases.

$$\text{Ratio} = \frac{\text{Value}_{\text{Textured}} - \text{Value}_{\text{Untextured}}}{\text{Value}_{\text{Untextured}}}$$ (2)
3.3.2 Surface examinations

The surface topography of the tested bottom and bearing plates was observed using the CLSM and the micrographs are shown in Figure 11. The wear marks at the high-pressure side are more obvious than those at the low-pressure side on the bearing plate sliding against the untextured retainer (abbreviated as the UT bearing plate in the following contents). Meanwhile, the wear marks on the bearing plate which slid against the textured retainer (abbreviated as the T bearing plate in the following contents) are relatively uniformly distributed. It can be deduced that the untextured retainer excessively tilts towards the high-pressure side at high speeds, and the surface texture with an area ratio of 16.4% could generate additional hydrodynamic pressure to prevent the retainer from excessively tilting. Therefore, the lower leakage was obtained with the textured retainer.

In addition, the area covered by black oxides of the T bearing plate is larger than that of the UT bearing plate. Black oxides and transferred brass were found on the textured retainer surface, while the untextured retainer hardly showed any of them. Much material of the T bearing plate was worn off and large amounts of brass accumulated inside the dimples. The above phenomena were consistent with the energy dispersive X-ray spectroscopy (EDS) results in Figure 12. Besides obvious scratches were found on the two retainer’s bottom plates, the bulges around the dimple rims were flattened. It can be concluded that at high speeds with the outlet pressure of 15 MPa, the bulges aggravate the wear of the components in the prototype so that the component surfaces were burned by the high temperature caused by the solid contact.
In the future, the effects of the bulges around the dimple rim will be studied thoroughly. More systematic investigations using the CFD technique and experiments on the EHA pump will be carried out to obtain a more suitable surface texture design.

5 Acknowledgements

The authors would like to thank the Shanghai Fermi Laser Technology Co., Ltd. for providing the laser surface texturing services. The research was financially supported by National Basic Research Program of China (973 Program) (No. 2014CB046403) and National Natural Science Foundation of China (No. 51605425).

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Development of a rotary pneumatic transformer

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Pneumatic drives are widely used in industrial applications. As the energy demand of production systems becomes more and more important, nowadays, many users favour a reduction of the general supply pressure to save energy. Nevertheless, some applications afford compact and powerful drives. To serve these demands, an energy efficient local pressure boosting is necessary. Today, linear pressure boosters based on double-piston cylinders are used to fulfil this task. The paper proposes a novel concept based on pneumatic radial piston motors. The new concept features a radial piston compressor, which is driven by a radial piston motor. The paper shows simulation data as well as a validation by experimental investigations of a working model of the new booster. Different configurations of the booster are examined for a range of driving pressures and pressure ratios. The experimental results are compared to a standard pneumatic booster subsequently.

Keywords: Pneumatics, Energy Efficiency, Pressure Booster

Target audience: Pneumatics, Automation

1 Introduction

Pneumatic drives are used in a great variety of applications, especially in manufacturing and process engineering. Since compressed air is an expensive form of energy, recent developments show a tendency to lower the overall system pressure in compressed air systems. This improvement should lead to lower energy consumption in the compressor while larger drives are needed to provide equivalent driving forces /1/. Despite that, some applications require compact drives and, therefore, a higher driving pressure. To ensure the applicability of these drives while maintaining the energy efficiency it is suitable to implement pressure boosters.

Current concepts for pressure boosters feature double piston linear transformers which have a very limited efficiency. To enhance the efficiency of pneumatic transformers, the paper proposes a novel concept featuring rotating units. The new transformer comprises of a pneumatic radial piston motor which drives a radial piston compressor. Similar concepts using pumps and motors are well known by hydraulic systems /1-3/. These hydraulic transformers are used, e.g., for energy recovery in mobile or stationary systems. Hydraulic transformers are often built using axial piston machines. In these units, the separation between high and low pressure and the valve function are realised using a valve plate. Due to the low viscosity of air, control using a valve plate is not possible as the leakage through the gaps would cause very high losses which negatively impact the efficiency of the booster. Therefore, an actuation using switching and check valves is proposed.

1.1 Standard Booster

Standard pneumatic boosters that are used in industrial applications today are designed as double piston cylinders, as shown in figure 1. The double piston fulfils an alternating stroke from one side to the other. In each cycle, one of the inner chambers is used for the compression of the output air. One driving chamber and the other inner chamber work as driving chambers for the compression. The second driving chamber is exhausted to the environment. The ratio between input and output mass flow as well as between input and output pressure is defined by the ratio between the piston areas used for driving and compression, respectively. The switching valve is actuated pneumatically which is not shown in the figure for readability.

Most boosters offer the possibility to reduce the driving pressure in the driving chambers. Due to the force equilibrium at the piston the output pressure is decreased. As the air in the outer driving chambers is exhausted to the environment at the end of each cycle, a lower driving pressure in these chambers also increases the efficiency of the booster because less air is filled into the chamber. Standard double piston boosters emit a lot of noise due to the nearly undamped end stops of the piston. The efficiency of an exemplary piston booster is examined in chapter 4 of the paper.

2 Novel booster concept

Figure 2 shows a schematic of the novel design. The booster consists of a pneumatic radial piston motor shown on the left hand side and a radial piston compressor shown on the right.

The transformer takes in air at supply pressure \( p_{\text{sup}} \) in order to compress it to a higher outlet pressure \( p_{\text{out}} \). The energy needed for the compression process is supplied by the pneumatic motor shown on the left side. Each
piston is actuated by two switching valves. The compressor (right hand side) is controlled by two check valves per cylinder connecting the chamber to the supply and outlet, respectively. For clarity reasons, the valves are depicted for one cylinder of both motor and compressor only.

Additionally, the p-V-diagrams of the motor and compressor are shown in figure 2. Due to the actuation of the motor by means of switching valves, an adjustment of the motor characteristics is possible. A variation of the piston’s radial position at which the connection to the air supply is cut off influences the air mass fed to the motor. Earlier switching off does not only lead to lower air consumption and therewith higher efficiency but also reduces the mean driving torque of the motor and therewith the maximum output pressure of the compressor which is directly proportional to the mean torque.

### 3 Simulation model

To establish the main influences on the efficiency of the novel pneumatic booster, a lumped parameter simulation model was developed. The following pages present the mathematical description of the booster, its implementation in the simulation environment and some sample results of the simulation study.

#### 3.1 Mathematical description

To understand the working principle and the main influences on the exergy efficiency of the booster, a mathematical description of the kinematics, thermodynamics and the effect of the dead volume attached to each cylinder is of great importance. This description is shown in paragraphs 3.1.1 to 3.1.3.

#### 3.1.1 Kinematic model

Figure 2 shows a schematic of one cylinder including the crank drive which converts the linear movement of the piston into a rotational movement of the shaft. The conversion from rotational movement to linear movement is mainly dependent on the push rod ratio \( \lambda_s = \frac{\ell_{\text{rod}}}{\ell_{\text{crank}}} \).

\[
\ell_{\text{crank}} = \frac{1}{4 \lambda_s \left( 1 - \cos \phi + \frac{1}{2} \lambda_s \cdot \sin (2 \phi) \right)}
\]

\[
\dot{s} = \frac{\dot{\phi}}{\omega} = \frac{d\ell_{\text{rod}}}{d\phi} = \frac{\ell_{\text{crank}} \cdot \dot{s}}{\lambda_s \cdot \sin (2 \phi)}
\]

The torque \( \tau_{\text{press}} \) produced by the pressure force \( F_s = \Delta p \cdot A_{\text{piston}} \) can then be calculated using equation (3) with \( y = \arcsin \left( \frac{\ell_{\text{crank}}}{\ell_{\text{rod}}} \cdot \sin \phi \right) \).

\[
\tau_{\text{press}} = F_s \cdot \ell_{\text{crank}} = \frac{\cos (90 - \phi - y)}{\cos y} \cdot F_s \cdot \ell_{\text{crank}}
\]

The rotational acceleration \( \dot{\phi} \) is then calculated from the conservation of angular momentum using equation (4) with the sum of driving torque produced by the motor, load torque applied by the compressor and the torque losses \( M_{\text{loss}} \) caused by friction.

\[
\dot{j_s} \cdot \dot{\phi} = \sum_{j=1}^{n} M_{\text{press},j} - \sum_{j=1}^{n} M_{\text{press},j} - M_{\text{loss}}
\]

### 3.1.2 Thermodynamic modeling

Equation (5) delivers the calculation of the pressure change in each cylinder chamber. The total pressure change is calculated as the sum of the pressure change due to mass transfer into and out of the chamber \( \Delta p_{\text{in}} \), the pressure change \( \Delta p_{\text{out}} \) caused by heat exchange between air and cylinder walls and the pressure change \( \Delta p_{\phi} \) due to the volume change of the cylinder chamber in the cause of the rotational movement. \( /5/ \)

\[
\dot{p} = \dot{p}_{\text{in}} + \dot{p}_{\text{out}} + \dot{p}_{\phi}
\]

To examine the efficiency of the booster, the thermodynamic concept of exergy is used. Exergy describes the ability to conduct work of different forms of energy. The exergy fed to a pneumatic system can be calculated using equation (6). This includes the compressed air mass flow as well as the electrical energy fed e.g. to the valves. \( /6/ \)

\[
\Delta E_{\text{ex}} = \Delta m_{\text{air}} \left( h_{\text{air}} - h_{\text{amb}} \right) + \Delta T_{\text{amb}} \left( s_{\text{amb}} - s_{\text{amb}} \right) + W_{\text{el,up}}
\]

As described in \( /7/ \), an analysis using exergy instead of energy is advantageous for pneumatic systems because the influence of different losses on the overall efficiency can be quantified more easily. Additionally, effects that have a small influence on the working ability of a system can be neglected during an optimisation of automation systems.

The exergy efficiency is then defined according to equation (7) as the ratio between the output exergy \( \Delta E_{\text{out}} \) and the input exergy \( \Delta E_{\text{in}} \) over a certain period of time. The total energy consumed by the valves is around 6000 J for one cycle shown in chapter 4. Due to its small amount in relation to the exergy contained in the compressed air mass flow the electrical energy \( W_{\text{el,up}} \) is neglected in the following.

\[
\zeta = \frac{\Delta E_{\text{out}}}{\Delta E_{\text{in}}} = \frac{\Delta m_{\text{air,up}} \left( h_{\text{air,up}} - h_{\text{amb}} \right) + \Delta T_{\text{amb}} \left( s_{\text{amb}} - s_{\text{amb}} \right)}{\Delta m_{\text{air,up}} \left( h_{\text{air,up}} - h_{\text{amb}} \right) + \Delta T_{\text{amb}} \left( s_{\text{amb}} - s_{\text{amb}} \right)}
\]

If an isothermal change of state is assumed, equation (7) can be simplified to equation (8). This assumption is valid for the described booster if cooling the air to ambient temperature in the accumulator is taken into account.

\[
\zeta = \frac{\Delta E_{\text{out}}}{\Delta E_{\text{in}}} = \frac{\Delta m_{\text{air,up}} \ln \left( \frac{P_{\text{up}}}{P_{\text{amb}}} \right)}{\Delta m_{\text{air,up}} \ln \left( \frac{P_{\text{up}}}{P_{\text{amb}}} \right)}
\]

In the study, the exergy efficiency is calculated according to the definition presented in equation (8).

### 3.1.3 Influence of the dead volume

The dead volumes of compressor and motor have a major influence on the maximum efficiency which can be obtained by the booster. Therefore, a mathematical model of this influence is presented as follows.

Without consideration of leakage, the mass flow \( \dot{m}_{\text{sup}} \) fed to the booster can be calculated as the sum of the theoretical input mass flows into motor and compressor according to equation (9).
\[ m_{\text{sup}} = m_{\text{thc,mot}} + m_{\text{thc,com}} \]  

(9)

As the motor cylinder chambers are completely exhausted at the end of each rotational cycle, the theoretical input mass flow (without leakage losses) to the motor can be calculated by the chamber pressure at the inner dead center (IDC).

\[ m_{\text{thc,mot}} = V_{\text{cyl,mt}} \cdot \frac{p_{\text{yl,mt}} + p_{\text{yl,cmt}}}{\rho_{\text{yl,mt}}} \cdot n \]  

(10)

To calculate the theoretical output mass flow, the density at output pressure is needed. \( V_{\text{cyl,cmt}} \) describes the part of the chamber volume which needs to be compressed to reach the output pressure. The dead volume \( V_{\text{dead,cmt}} \) influences the size of \( V_{\text{cyl,cmt}} \) and therewith the efficiency of the booster. It is only filled with compressed air once, since it is not exhausted to the environment.

\[ m_{\text{thc,com}} = V_{\text{cyl,cmt}} \cdot \frac{p_{\text{yl,cmt}} - p_{\text{yl,cmt,sup}}}{\rho_{\text{yl,com}}} \cdot n \]  

(11)

Equation (12) describes the calculation of the volume \( V_{\text{yl,com}} \) at which the output pressure is reached. Transforming this equation delivers equation (13) for the compression volume \( V_{\text{cyl,com}} \).

\[ \frac{p_{\text{yl,com}}}{p_{\text{yl,cmt}}} = \left( \frac{V_{\text{cyl,cmt}}}{V_{\text{cyl,com}} + V_{\text{dead,com}}} \right)^n \]  

(12)

\[ V_{\text{cyl,com}} = V_{\text{cyl,cmt}} \left( 1 - \frac{p_{\text{yl,cmt}}}{p_{\text{yl,com}}} \right)^n \]  

(13)

Inserting equation (13) in equation (11) leads to equation (14) for the theoretical output mass flow of the compressor in dependence of the inverted pressure ratio \( \frac{p_{\text{yl,com}}}{p_{\text{yl,cmt}}} \).

\[ m_{\text{out,thc,com}} = \left( \frac{V_{\text{cyl,cmt}}}{V_{\text{cyl,com}} + V_{\text{dead,com}}} \right)^n + V_{\text{dead,com}} \cdot \left( \frac{p_{\text{yl,cmt}}}{p_{\text{yl,com}}} \right)^n - 1 \right) \cdot \rho_{\text{yl,com}} \cdot n \]  

(14)

Figure 4 shows the influence of the compressor dead volume on the output mass flow for both isothermal (\( n = 1 \)) and isentropic (\( n = 1.4 \)) compression. The dead volume has been normalised to the cylinder volume whereas the output mass flow was normalised to the value which is reached with no dead volume for each pressure ratio.

It is obvious, that even a relatively small dead volume of only 10 % of the chamber volume reduces the output mass flow and, therewith, the efficiency of the booster by more than 10 % for an isothermal compression with a pressure ratio of 2.5. In the functional model presented in chapter 4, off-the-shelf-components were used. This leads to a relatively large percentage of dead volume in comparison to the cylinder chamber volume. In case of the small compressor with 22 mm piston diameter, the dead volume is nearly one quarter of the chamber volume.

In this case, the maximum output mass flow is reduced by up to one third.

### 3.2 Implementation

An excerpt of the simulation model implemented in DSHplus is shown in Figure 5. Both the motor and the compressor are modelled using a combination of a pneumatic cylinder and a thrust crank.

The torque delivered by the thrust cranks works as an input to a common shaft which comprises the inertia of the booster. Additionally, velocity dependent friction is calculated in the model of the rotating mass. The rotational position and velocity are then fed back to the thrust cranks where they are converted to the radial position and velocity of the cylinders where the piston force and the chamber pressure are calculated. The specific cylinder model used in the simulations does not include friction losses.

### 3.3 Simulation results

In the following, some sample results with high influence on the design of rotary pneumatic boosters are shown. Results towards exergy efficiency are presented for different configurations that were analysed in the study.

#### 3.3.1 Exergy efficiency

Different configurations of motor and compressor geometry were included in the research. In all variations a five cylinder motor was implemented in combination with a three or five cylinder compressor. For both motor and compressor, the piston diameter, the crank and rod length were varied on the basis of off-the-shelf pneumatic radial piston motors. The piston diameter values considered here are 22 mm, 32 mm and 44 mm respectively.

Table 1 gives an overview over some simulation results for the exergy efficiency for a configuration comprising of a five cylinder motor in combination with a three cylinder compressor with 32 mm piston diameter each.

<table>
<thead>
<tr>
<th>( p_{\text{sup}} ) (bar)</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 bar</td>
<td>51.5 %</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>5 bar</td>
<td>56.8 %</td>
<td>40.4 %</td>
<td>36.6 %</td>
<td>-</td>
</tr>
<tr>
<td>6 bar</td>
<td>71.1 %</td>
<td>65.0 %</td>
<td>52.8 %</td>
<td>39.8 %</td>
</tr>
</tbody>
</table>

The results show a maximum efficiency of over 70 %. In this case, only a small pressure ratio of 7/6 was reached. For higher pressure ratio values, the efficiency is lower, yet an efficiency of more than 50 % is possible.
Variation of the piston diameter shows an increase in efficiency for larger pistons due to the lower impact of friction in comparison to the driving torque and especially to the lower losses caused by the dead volume. As the dead volume stays constant for all variations, its percentage of the cylinder chamber volume is lower and, therefore, a higher output mass flow can be realised.

4 Functional model
To validate the simulation results and to show the potential of the new transformer, a functional model was built. The experimental setup, the functional model itself and some experimental results are shown in the following.

4.1 Experimental setup
Figure 6 shows a schematic of the test rig used to investigate the pneumatic booster. The high pressure air is delivered into an accumulator with a volume of 5 l. To simulate pneumatic consumers, a valve downstream the accumulator is connected to the environment. Therewith, air is released and the pressure inside the accumulator decreases.

The test rig includes two mass flow sensors measuring the supply mass flow and the high pressure output mass flow, respectively. Additionally, the pressures upstream and downstream the booster are measured.

The working model developed in the study is shown in figure 7. It consists of a 5 cylinder radial piston motor (left) and a 3 piston radial piston compressor (right). To apply the standard radial piston motors, the cylinders were modified. New cylinders were designed to ensure sealing of the chambers and check as well as switching valves were implemented. The torque sensor between motor and compressor is used to define the rotational position of the shaft and, therewith, the radial positions of the pistons.

In addition to the working model shown in figure 7, a second version comprising of a three piston compressor with a smaller piston diameter and a larger three piston motor was constructed and examined in the study. This smaller configuration is suitable for applications with lower demand for high pressure air but is still able to produce air at the same pressure ratio as the initial configuration.

4.2 Experimental results
4.2.1 Standard piston booster
Figure 8 shows sample experimental results for the exergy efficiency of a standard pneumatic booster in double piston configuration. The set up used for the investigation equals the set up for the novel booster concept. The absolute supply pressure was set to 5 bar. When the pressure inside the accumulator reaches 7.5 bar, the booster is switched off and the volume is exhausted to the environment. At a pressure lower than 7 bar, the booster restarts.

The input and output exergy shown in figure 8 leads to an exergy efficiency of 47%. An overview over the exergy efficiency obtained with the standard booster for different input pressure values as well as pressure ratios is given in table 2.

<table>
<thead>
<tr>
<th>Supply pressure (bar)</th>
<th>Output pressure (bar)</th>
<th>Pressure ratio</th>
<th>Exergy efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.5</td>
<td>5.5-5.5</td>
<td>1.43-1.57</td>
<td>49.0 %</td>
</tr>
<tr>
<td>4</td>
<td>6-6.5</td>
<td>1.5-1.625</td>
<td>47.6 %</td>
</tr>
<tr>
<td>5</td>
<td>7-7.5</td>
<td>1.4-1.5</td>
<td>46.6 %</td>
</tr>
<tr>
<td>5-8</td>
<td>5.5-8</td>
<td>1.5-1.6</td>
<td>47.1 %</td>
</tr>
</tbody>
</table>

All values for the efficiency lie within the range of 46 and 49 %.

4.2.2 Novel booster concept
First experimental results for a compression from a supply pressure of 5 bar to an output pressure of 7 to 7.5 bar are shown in figure 9. Besides the pressure in the supply line and inside the high pressure accumulator, the exergy fed to the booster and the output exergy are shown.

Figure 7: Pneumatic transformer of novel design concept (5 cylinder motor, three cylinder compressor)
4.2.3 Optimisation of switching positions

The variation of the switching points is depicted in figure 10. Without any optimisation, the air supply is switched off at a radial position of the piston which is near to the IDC. This leads to a very high mean torque of the motor but no expansion energy comprised in the air is used. Now, the switching point is moved closer to the outer dead center (ODC). This leads to a much smaller amount of air fed to the cylinder chamber and, therewith, a higher efficiency of the booster. On the downside, the maximum torque of the motor is reduced and the maximum pressure ratio decreases as well.

The resulting values for the exergy supplied to the transformer and the exergy output to the accumulator are shown in figure 11. It is obvious, that the exergy supplied to the booster is much lower than before optimisation due to the lower mass flow into the booster. This leads to an exergy efficiency of 49.6 % which is increased significantly with respect to the initial approach shown in figure 9.
An additional approach to increasing the efficiency is the reduction of friction losses. The pressure ratio is determined by the ratio between the driving torque and the load torque which comprises of the compressor torque and the friction torque. The pistons of the functional model presented in the paper are sealed using o-rings due to the available installation space. In the future, a more sophisticated sealing concept shall be designed and implemented to reduce the necessary driving torque and therewith the input air mass.

### 6 Acknowledgements

The authors thank the Research Association for Fluid Power of the German Engineering Federation VDMA for its financial support. Special gratitude is expressed to the participating companies and their representatives in the accompanying industrial committee for their advisory and technical support.

### Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$c_p$</td>
<td>Specific, isobaric heat capacity of air</td>
<td>[J/kgK]</td>
</tr>
<tr>
<td>$E_{air}$</td>
<td>Exergy contained in the air mass</td>
<td>[J]</td>
</tr>
<tr>
<td>$F_G, F_TG$</td>
<td>Pressure force / Pressure force in tangential direction</td>
<td>[N]</td>
</tr>
<tr>
<td>$h_{amb} - h_{air}$</td>
<td>Specific enthalpy of air at ambient/current conditions</td>
<td>[J/kg]</td>
</tr>
<tr>
<td>IDC, ODC</td>
<td>Inner/outer dead center</td>
<td>[°]</td>
</tr>
<tr>
<td>$l_{crank} / l_{rod}$</td>
<td>Crank/rod length</td>
<td>[m]</td>
</tr>
<tr>
<td>$M$</td>
<td>Torque</td>
<td>[Nm]</td>
</tr>
<tr>
<td>$m_{air}$</td>
<td>Air mass</td>
<td>[kg]</td>
</tr>
<tr>
<td>$n$</td>
<td>Polytropic coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>$n$</td>
<td>Rotational speed</td>
<td>[1/s]</td>
</tr>
<tr>
<td>$p_{amb} - p_{sup} - p_{out}$</td>
<td>Ambient/supply/output pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>$R$</td>
<td>Specific gas constant of air</td>
<td>[J/kgK]</td>
</tr>
<tr>
<td>$s$</td>
<td>Piston stroke</td>
<td>[m]</td>
</tr>
<tr>
<td>$s_{amb} - s_{air}$</td>
<td>Specific entropy of air at ambient/current conditions</td>
<td>[J/kgK]</td>
</tr>
<tr>
<td>$T_{amb}, T_{sup}, T_{out}$</td>
<td>Ambient/supply/output temperature</td>
<td>[K]</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume</td>
<td>[m³]</td>
</tr>
<tr>
<td>$W_{el, in}$</td>
<td>Electrical energy supplied to the system</td>
<td>[J]</td>
</tr>
<tr>
<td>$z_{cyl}$</td>
<td>Number of cylinders</td>
<td>[-]</td>
</tr>
<tr>
<td>$\zeta$</td>
<td>Exergy efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>$\lambda_s$</td>
<td>Push rod ratio</td>
<td>[-]</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
<td>[kg/m³]</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>Rotational position</td>
<td>[°]</td>
</tr>
</tbody>
</table>

### Table 4: Resulting exergy efficiency for different supply pressure and pressure ratio values

<table>
<thead>
<tr>
<th>Supply pressure</th>
<th>Output pressure</th>
<th>Pressure ratio</th>
<th>Min. angle at switching off</th>
<th>Exergy efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>3 bar$_{abs}$</td>
<td>5-5.5 bar$_{abs}$</td>
<td>1.43-1.57</td>
<td>270°</td>
<td>23.0 %</td>
</tr>
<tr>
<td>4 bar$_{abs}$</td>
<td>6-6.5 bar$_{abs}$</td>
<td>1.5-1.625</td>
<td>240°</td>
<td>33.6 %</td>
</tr>
<tr>
<td>5 bar$_{abs}$</td>
<td>7-7.5 bar$_{abs}$</td>
<td>1.4-1.5</td>
<td>240°</td>
<td>36.9 %</td>
</tr>
<tr>
<td>5 bar$_{abs}$</td>
<td>7.5-8 bar$_{abs}$</td>
<td>1.5-1.6</td>
<td>240°</td>
<td>37.1 %</td>
</tr>
</tbody>
</table>

The overall efficiency of the smaller configuration is about 12 to 22 percentage points lower than the efficiency obtained with the larger configuration. This is mainly caused by the larger percentage of dead volume in comparison to the chamber volume. As the driving power of the smaller motor is lower, friction losses have a larger influence on the overall efficiency which is revealed in the efficiency decrease with lower supply pressure.

### 5 Summary and Conclusion

The paper shows a novel concept of pneumatic pressure boosters based on radial piston units. The motor is actuated by two fast switching valves per cylinder whereas the compressor is passively controlled by two check valves. To identify the efficiency potential of the concept, a simulation study was conducted. Afterwards, a working model was built and the exergy efficiency was measured.

The simulation results show exergy efficiency values up to 71 %. This large potential was verified in the experimental study where exergy efficiency values up to 59 % were identified. The measurements show an increase of efficiency with driving pressure as well as pressure ratio. It is shown that the larger configuration, comprising of a five cylinder motor and a three cylinder compressor, has significantly better efficiency than the smaller configuration comprising of two three cylinder units with different piston diameters.

A reduction of dead volumes leads to an increase in efficiency as the output mass flow is increased and the input mass flow is reduced. In the functional model, a design using off-the-shelf components was implemented. This approach leads to large dead volumes of up to 22 % of the smaller compressor’s cylinder chamber volume. Due to this fact the efficiency of the small booster configuration is lower.
References


Experimental and Theoretical Investigation of Lightweight Pumps and Fluid Reservoirs for Electrically Driven Vacuum Systems in Automated Handling Processes

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It is known that the performance of a hydraulic system can be increased significantly by a combination of a pump and a reservoir. As the electrical vacuum generation’s ability to compete compared to classical ejectors is limited in this article the combination of pumps and reservoirs is applied to the vacuum technology used in automated handling processes. Evacuation times and energy consumption of the electrical vacuum pumps are measured. Two possible use case scenarios are the basis for investigations how a fluid reservoir influences evacuation time and energy consumption. The results are then compared to a pneumatic ejector.

Keywords: automation, vacuum handling, vacuum pump, fluid reservoir
Target audience: automation, vacuum handling

1 Vacuum generation and vacuum grippers in automated handling processes

In the context of the megatrend sustainability and the associated ecological efficiency companies are obliged to significantly reduce energy consumption. Considering increasing costs of energy make a low energy use worthwhile. At this point the optimization of existing systems offers great potential /1/–/3/. Electric vacuum generation becomes more and more attractive as the overall efficiency of an electric vacuum pump is much higher than of a pneumatic ejector combined with a compressor to supply the needed compressed air /4/. Furthermore future factories might relinquish compressed air due to said low level of efficiency and high losses. In this case electric vacuum generation is the way to go in handling processes /5/.

Table 1 shows a comparison of the three setups that can be used for vacuum generation. The ejector setup provides the highest volume flow rate as it is connected to a compressed-air lines which is applied with several bars of pressure. This causes a high input flowrate resulting in high flow rates at the vacuum side of the ejector. Compared to this the electric vacuum pump offers the lowest volume flow rate when using similar outer dimensions. A combination of a pump and reservoir has a higher volume flow rate compared to the sole use of an electric vacuum pump as the vacuum generation in the suction cup is decoupled from the pump’s direct flow rate. A comparison of more than 500 lightweight electric vacuum pumps shows that the vacuum level of several pumps (up to ~900 mbarg) is beyond the typical vacuum level of ejectors (around ~050 mbarg). In order to generate vacuum in the suction cup with a reservoir said reservoir has to have a higher vacuum level than the one that is desired in the suction cup resulting in a reduced vacuum level in the suction cup compared to a electric vacuum pump setup. Regarding acquisition cost ejectors have the lowest followed by a

Table 1: Qualitative comparison of different forms of vacuum generation

<table>
<thead>
<tr>
<th>Form of vacuum generation</th>
<th>Ejector</th>
<th>Lightweight electric vacuum pump</th>
<th>Electric vacuum pump and reservoir</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volume flow rate</td>
<td>+</td>
<td>-</td>
<td>0</td>
</tr>
<tr>
<td>Vacuum level</td>
<td>0</td>
<td>+</td>
<td>-</td>
</tr>
<tr>
<td>Acquisition costs</td>
<td>+</td>
<td>0</td>
<td>-</td>
</tr>
<tr>
<td>Operating costs</td>
<td>-</td>
<td>+</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure 1 depicts the classification of a vacuum gripper in a production system. The vacuum gripper is the connection between the handling device (e.g. a robot) and the workpiece. It consists out of one or more vacuum systems that are made up of a vacuum generator that is attached to the suction cup by a fluidic connection /7/. In general there are two possible ways of generating a vacuum. Both of them are shown in Figure 2. On the left the typical setup for pneumatic vacuum generation is shown. This usually consists of a compressor supplying the ejector with compressed air via a compressed-air line. The ejector generates a vacuum with the venturi principle. In the middle the electric vacuum generation is shown where the electric vacuum pump is directly connected to the suction cup. The right side depicts electric vacuum generation in combination with a fluidic reservoir. This setup is used in the following chapters.

Figure 2: vacuum generation: pneumatic (left), electric (middle) and electric with reservoir (right)
2 Handling process and use case scenarios

Figure 4 depicts a typical sequence of a handling process performed by a vacuum gripping system. It can be described with the steps gripping, clamping and releasing /9/. This basic approach contains the main steps of a handling process and can be applied not only to vacuum gripping systems. Said steps can be divided in sub-steps for a more detailed description /10/. This allows diverting the different means of gripping. During the sub-step “applying gripping force” the vacuum is generated and has to be sustained during “lifting” and “transporting”. After “backing away” the vacuum gripper returns to its initial position to start over the handling process. The evacuation time \( t_{\text{evac}} \) is how long the duration of “applying gripping force” takes. This step as well as “dropping off” are crucial for the gripping system as they are the only ones to be influencing the time of the handling process directly. The speed of all other steps is set by the handling device (cf. Figure 1).

Table 2: Use case scenarios

<table>
<thead>
<tr>
<th>No.</th>
<th>1</th>
<th>2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Name</td>
<td>Vacuum end effector with pouch</td>
<td>Sheet handling in pressing plant</td>
</tr>
<tr>
<td>Picture</td>
<td><img src="image1" alt="Vacuum end effector with pouch" /></td>
<td><img src="image2" alt="Sheet handling in pressing plant" /></td>
</tr>
<tr>
<td>Suction cups</td>
<td>SPH1 20</td>
<td>SAB 50</td>
</tr>
<tr>
<td>Quantity of suction cups</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>Length of tubes/hoses</td>
<td>0,3 m</td>
<td>2 m</td>
</tr>
<tr>
<td>Inner diameter of hoses</td>
<td>15 mm</td>
<td>4 mm</td>
</tr>
<tr>
<td>( V_{\text{evac}} )</td>
<td>400 ml</td>
<td>150 ml</td>
</tr>
<tr>
<td>( P_{\text{required}} )</td>
<td>-300 mbar</td>
<td>-600 mbar</td>
</tr>
<tr>
<td>( t_{\text{evac, max}} )</td>
<td>0,5 s</td>
<td>0,2 s</td>
</tr>
<tr>
<td>transport time</td>
<td>1-2 s</td>
<td>3 s</td>
</tr>
<tr>
<td>return time</td>
<td>1-2 s</td>
<td>4 s</td>
</tr>
</tbody>
</table>

These two use case scenarios are now used to investigate a substitution of the pneumatic vacuum generation with an electric vacuum pump and a fluidic reservoir in order to still reach the maximum evacuation time but significantly lower the energy consumption.

3 Electric vacuum pumps in automated handling systems and modelling approach

In order to be able to make a statement considering the basic factors evacuation time and energy consumption first a theoretical approach is carried out. With slight modifications of the ideal gas law the relation between evacuation time and evacuation volume can be written as

\[
t_{\text{evac}} = \frac{V_{\text{evac}}}{\dot{m}}
\]  

(1)

which is a linear correlation. This is applicable for a target vacuum value \( \beta \) and depends on the pumps flow rate \( m \) as well as the ambient temperature \( T \). A connection between energy consumption and evacuation volume can be established when taking into account the conservation of energy in the electric vacuum pump i.e.

\[
E_{\text{evac}} = \frac{P_{\text{pump}}}{\eta}
\]  

(2)
Transformed to
\[ E_{\text{vac}} = \frac{V \cdot \hat{\beta}}{\eta} \]  
(3)

which is again a linear correlation applicable for a target vacuum value \( \hat{\beta} \) and the pump's efficiency \( \eta \). In order to confirm above statements considering evacuation time and energy consumption measurements with different pumps, evacuation volumes and vacuum levels are conducted. The electric vacuum pumps (EVPs) are labelled according to the scheme

“EVP [max. flow rate in 1/min] [max. vacuum level in mbarg]”.

Figure 5 shows the setup for measuring evacuation times as well as energy consumption. The pump (1.1: EVP 34|-780, 1.2: EVP 12|-800, 1.3: EVP 2,1|-670) is turned on via control device (7) that starts a timer and evacuates the reservoir (2.1: 0.4 l, 2.2: 0.75 l, 2.3: 2 l) until the vacuum switch (3) detects the desired pre-set pressure and turns off the pump via relays (4). The same signal that switches the relays stops the control device’s timer and thereby determines the evacuation time. During the whole time voltage and current are measured by the R&S HMC8015 Power Analyzer (6) and logged via computer (5). The design of experiments is a full factorial experiment with the following factors:

- pump = {EVP 34|-780; EVP 12|-800; EVP 2,1|-670}
- reservoir size = {0.4 l; 0.75 l; 2 l}
- vacuum levels = {-300 mbarg; -600 mbarg; -750 mbarg}

The results are shown in Figure 6 and Figure 7. In Figure 6 each diagram shows the evacuation time of one pump evacuating different reservoir sizes to different vacuum levels. Said vacuum levels are typical for various applications. Figure 7 depicts the energy consumption. Each diagram shows the energy consumption of one pump. In both figures the last diagram does not contain values for -750 mbarg as the respective pump cannot reach this vacuum level.
Using the calculated values for evacuation rate and energy consumption rate, each pump can be applied to both use case scenarios using the following equations with the $V_{res}$ and $p_{new}$ values from Table 2:

$$t_{evac} = \frac{V_{res}}{n_{vacuum}} \cdot l_{vacuum} \quad (4)$$

$$E_{evac} = \frac{E_{res}}{l_{reservoir}} \cdot l_{reservoir} \quad (5)$$

The calculated results for both use case scenarios are shown in Table 5.

<table>
<thead>
<tr>
<th>Pump</th>
<th>Use case scenario</th>
<th>Evacuation time $t_{evac}/s$</th>
<th>Energy consumption $E_{evac}/J$</th>
</tr>
</thead>
<tbody>
<tr>
<td>EVP 34:780</td>
<td>1</td>
<td>0,28</td>
<td>1,20</td>
</tr>
<tr>
<td>EVP 12:800</td>
<td>1</td>
<td>0,48</td>
<td>0,72</td>
</tr>
<tr>
<td>EVP 2,1</td>
<td>-670</td>
<td>1</td>
<td>3,12</td>
</tr>
<tr>
<td>EVP 34:780</td>
<td>2</td>
<td>0,42</td>
<td>1,65</td>
</tr>
<tr>
<td>EVP 12:800</td>
<td>2</td>
<td>0,75</td>
<td>1,08</td>
</tr>
<tr>
<td>EVP 2,1</td>
<td>-670</td>
<td>2</td>
<td>4,86</td>
</tr>
</tbody>
</table>

Table 5: Comparison of pump models with different use cases

The results show that a vast decrease in evacuation time comes with a relatively small increase in energy consumption. It also shows that the requirement of evacuation time in use case scenario 1 can be met by an electric vacuum pump whereas the evacuation times for use case scenario 2 are too high. Therefore a decrease here is required. Use case scenario 1 is also investigated in concerns of adding a fluidic reservoir to see at what cost the evacuation time can be reduced.

4 Influence on the performance by a fluidic reservoir

Regarding the calculations from above the EVP 12:800 seems to be the perfect fit for the use case scenario 1. The evacuation time is below the limit and energy consumption is the lowest among all measured pumps. Whereas the required evacuation time in use case scenario 2 cannot be reached with any of the available pumps. Therefore an improvement of said parameter has to be achieved. This can be done by using a pump reservoir system. As the results above (cf. Table 5) depict the pump itself can hardly or not at all fulfill the requirements considering evacuation times (cf. Table 2). Therefore, possible improvements using a reservoir are investigated.

The setup for these measurements is depicted in Figure 8. The Pump (1) evacuates the reservoir (2) until the vacuum switch (3) detects the desired pressure and turns off the pump via a relays (4). Then the valve (8) connecting the reservoir (2) with the second reservoir (5) which represents the vacuum gripping systems volume is opened via the control device (7). Said control device starts a timer when opening the valve that is stopped as soon as the second vacuum switch (6) has reached the desired pressure and sends a signal to the control device.

4.1 Use case scenario 1: Vacuum end effector with pouch

The system setup of use case 1 with a pump reservoir combination is shown in Figure 2 (right). With the ideal gas law /8/:

$$\Delta m = \frac{V_{air}}{\rho_{air}} \quad (6)$$

the amount of air that has to be taken out of the system to reach the desired vacuum value is 0,128g. When using a 0,75l reservoir with an vacuum value of 250mbar the pressure increase within said reservoir when being connected to the vacuum gripper is 144mbar. With conservation of mass

$$p_{new} = p_{initial} \cdot \frac{(V_{reservoir} + V_{reservoir}) - p_{vacuum} V_{reservoir}}{V_{reservoir}} \quad (7)$$

the theoretical maximal absolute pressure in the reservoir to still be able to evacuate the vacuum system is 556mbar. So in theory a reservoir with a volume of 0,75l and a starting pressure of 250mbar can be used twice to evacuate the vacuum gripper. Measurements show that with the initial pressure of 250mbar the evacuation time is between 0,099s and 0,113s. For the second evacuation process with a pressure in the reservoir of 400mbar the evacuation time is between 0,108s and 0,121s. This shows a vast improvement in evacuation times compared to the use of only an electric pump. Although this was not required in the first place there is an additional advantage of using a fluidic reservoir. In this setup, the pump has to evacuate the reservoir only every two cycles instead of every cycle when not using a reservoir. This leads to an increased lifetime of the pump. But also comes with increased energy consumptions. The conventional setup with no reservoir and the EVP 12:800 takes up 0,72J each cycle, resulting in 1,44J for two cycles. Whereas the combination of pump and reservoir consumes 1,12J (cf. Figure 7 middle) every two cycles leading to an average consumption of 5,6J per cycle. So improvement in evacuation time of up to 75% and pump life time comes with a trade off in energy consumption up to 300%.

4.2 Use case scenario 2: Sheet handling in pressing plant

The results of use case 1 show that a reduction of evacuation time can be achieved with an additional reservoir. The reservoir volume and pump have to be chosen in a way that the reservoir can be evacuated to the desired
vacuum level during the transport time of 5 seconds (see Table 2). The minimal volume of the reservoir with a vacuum level of 250 mbar is 0.6 l. In order to have some reserve a reservoir with a volume of 0.75 l is used. When using the EVP 12|800 because of the lowest energy consumption rate (cf. Table 4) this results in a measured evacuation time of 0.139 s. Again the evacuation time is significantly decreased by more than 70% and thus below the demanded value. Evacuation of the reservoir from 400 mbar to 250 mbar takes about 7 seconds (cf. Figure 6 (middle)) which is low enough to be done during transport and return time. Energy consumption in this use case scenario is 6.5 J with the additional reservoir and 2.5 J without reservoir which is an increase of 160%.

5 Comparison with ejector

In order to be able to make thorough statements about energy consumption and efficiency it is inevitable to compare the results of the electric vacuum pump without and with fluidic reservoir to a conventional ejector setup including the compressor. The air consumption of an ejector is twice as high as or even higher than the suction rate \( \frac{V}{8} \). A typical operating pressure for ejectors is 5 bar. Compressing 1 m\(^3\) of air to said operating pressure takes up at least 0.054 kWh \( \frac{1}{14} \). With these values the energy consumption of an evacuation process can be calculated for both use case scenarios using the ratio of air consumption to suction rate \( R \), the evacuation mass \( m_{\text{evac}} \) and the specific energy consumption of the compressor \( E_{\text{comp}} \)

\[
E_{\text{evac}} = \frac{m_{\text{evac}}}{\rho} \cdot R \cdot E_{\text{comp}}
\]

(8)

This leads to an energy consumption of 61.2 J for use case scenario 1 and an energy consumption of 36 J for use case scenario 2. These results compared to both the other setups are depicted in Figure 9.

6 Summary and Conclusion

Compared to a setup with only a pump the combination of pump and reservoir shows a reduction of evacuation times between 70% and 75% while increasing the amount of consumed energy between 160% and 300%. This is due to the lower pressure needed when using a reservoir in order to evacuate the vacuum gripping system compared to directly evacuating the vacuum gripping system without a reservoir. But when set side by side with an ejector the difference in energy consumption between the setups with electrical vacuum generation becomes extremely slight as the ejector setup takes up more than ten times the amount of energy in use case scenario 1 and more than five times the energy in use case scenario 2. A general assumption is that in use cases with a low vacuum level and a comparatively high evacuation volume pumps alone fulfil the required evacuation time. An additional fluidic reservoir in this case can function as a buffer in order to increase the lifetime of the electric vacuum pump. In both cases the energy savings compared to a conventional setup with an ejector are significant. Whereas in use case scenarios with a high vacuum level and a comparatively low evacuation volume the pump alone cannot fulfil the required evacuation time. But in combination with a fluidic reservoir it does. In general it can be said that adding a fluidic reservoir into a system with an electric vacuum pump decreases evacuation time, increases energy consumption but is still way more energy efficient than an ejector setup. In order to provide an approach to determine the size of the reservoir it is necessary to collect more data. Furthermore the linear correlations of evacuation time as well as energy consumption over volume at a constant pressure can be used as a basis for simulation models.
The 11th International Fluid Power Conference, IFK, March 19–21, 2018, Aachen, Germany

280, 2016


References

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_{evac}$</td>
<td>Evacuation energy</td>
<td>[J]</td>
</tr>
<tr>
<td>$E'_{evac}$</td>
<td>Energy consumption</td>
<td>[J/l]</td>
</tr>
<tr>
<td>$m$</td>
<td>mass</td>
<td>[kg]</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>$p_0$</td>
<td>ambient pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>$p_{needed}$</td>
<td>Needed vacuum (relative)</td>
<td>[bar]</td>
</tr>
<tr>
<td>$p_{reservoir}$</td>
<td>pressure in the reservoir</td>
<td>[bar]</td>
</tr>
<tr>
<td>$R$</td>
<td>gas constant</td>
<td>[J/(kg*K)]</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
<td>[K]</td>
</tr>
<tr>
<td>$t_{evac}$</td>
<td>Evacuation time</td>
<td>[s]</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume</td>
<td>[l]</td>
</tr>
<tr>
<td>$V_{evac}$</td>
<td>Evacuation volume</td>
<td>[l]</td>
</tr>
<tr>
<td>$V_{res}$</td>
<td>Evacuation rate</td>
<td>[l/s]</td>
</tr>
<tr>
<td>$V_{reservoir}$</td>
<td>reservoir volume</td>
<td>[l]</td>
</tr>
</tbody>
</table>


/12/ https://cdn.schmalz.com/media/01_gripping-systems/end-effector/en/end-effector-vee-03.jpg

/13/ https://cdn.schmalz.com/media/19_media-center/Anwendungen/73/Komponenten-fuer-Karosseriebauteile-073-DE.jpg

/14/ “EnEffAH: Energieeffizienz in der Produktion im Bereich Antriebs- und Handhabungstechnik”, 2012
Fast Switching Pneumatic Valves Driven by Magnetic Shape Memory Materials


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The increasing requirements on fast switching pneumatic valves, especially regarding the installation size, durability and high dynamics, demand for innovative systems. Magnetic shape memory (MSM) alloys are smart materials that can be activated by magnetic field to produce force and motion. Due to their high work-output and dynamics they are a promising alternative technology for a new generation of fast valves. This paper presents an investigation on the design process of a fast switching pneumatic valve based on MSM alloys. In particular, two valve concepts are described: a lever valve concept based on the magnetic elongation and mechanical resetting of the MSM element by a spring, and a seat valve consisting of an air-cored coil with a MSM element which opens the valve during its compression. This concept is later referenced by push design and the design steps are described in details in section 3 with focus on an optimization of the actuator. The second valve concept, developed by ETO MAGNETIC GmbH (Stockach, Germany), consists of an air-core coil with a MSM element in its centre which acts as the valve body to open the valve during its compression. This concept is later referenced by pull design. This concept is briefly described in section 4.

1 Introduction

Due to increasing performance requirements on fast switching pneumatic valves with respect to durability, functionality, installation size and response time unconventional actuators based on smart materials are in focus as an interesting alternative device technology for a new generation of valves. Especially magnetic shape memory (MSM) alloys seem to be promising because of improved characteristics in the last decade of research /1/. These are new smart materials, typically monocristalline Ni-Mn-Ga Heusler alloys /2/, which deform up to 6 - 12 % when subjected to a magnetic field or to an external stress, so they can be used as a basis for a new type of electromagnetic actuators. The MSM effect takes place in the martensitic phase of the alloy and is a result of a realignment of magnetically anisotropic alloy regions. These regions, called twinning phases, are defined by the orientation of the axes of the crystal structure regarding to the magnetic conductivity. The axes of high magnetic conductivity are called easy axes. Therefore its crystal structure is of prime importance. The reason for the shape memory effect are interior material stresses between the twinning phases, called twinning stress $\sigma_{tw}$. The presented work aims at the development of an innovative and competitive fast switching pneumatic valve for challenging industrial stationary pneumatic applications which should emphasize the potential advantages of the MSM technology with respect to the state-of-the-art pneumatic valves. The following applications are addressed:
- Emergency shutdown of production machines by quick exhausting of air
- Pneumatic drives in automation technology (e.g. placement of pins, sorting, cutting processes)
- Sorting of fraction with little dimensions (e.g. food and recyclable fractions)
- Sorting of fraction with little dimensions (e.g. food and recyclable fractions)

One feature that can be an advantage for all these applications is the durability of the MSM alloy. Many million times of activation and resetting an MSM element are experimentally proven in /3/. This premises an increased durability of the valve compared to the commonly applied seat valves. The durability of these valves are limited by wear of the elastomeric seal body caused by the transmission of kinetic energy when the highly accelerated anchor impacts to the seat during closing /4/. Depending on the design of the valve system the closing process can be realised without accelerating a whole mass but by elongation of the MSM. With respect to the other performance properties, the working principle of the MSM is promising to satisfy the requirements. Because MSM elements can be activated and reset both magnetically and mechanically, different actuator configurations are allowed corresponding to different operating modes /5/, giving the flexibility in design process regarding to installation size and functionality, e.g. self-sensing and multistability with near-zero current consumption in any position. The three addressed applications have different requirements concerning the response time for switching on $\tau_{rise}$ and $\tau_{fall}$ and the holding time of the open state. In case of an emergency shutdown the motion of pneumatic drives needs to be stopped in a duration of a split second, realised by fast venting. The holding time equally to the time of venting depends on the gas volume and maximum volume flow determined by the valve geometry. In pneumatic applications for automation technology, the productivity is limited by the response time of the pneumatic drives while the holding time depends on the task. This is different to the sorting application, which demands for a pulsed cycling. The minimum holding time is therefore limited by the time needed for formation of the air flow to blow away the fraction, depending on the size and mass of the sorting material and on the geometry of the valve. In addition the installation size, concrete the modular size effects directly the productivity.

For the definition of constructional aims of a fast switching pneumatic valve based on MSM actuator a requirement analysis of these application scenario was done, yielding to different demands on the functionality and performance of the valve. As a result, compromised main targets are defined in a requirement list, which are shortly commented in section 2. As a result of a morphological analysis based on this list two valve concepts with different MSM operating modes and valve principles turned out as suitable for the chosen application requirements. The first valve concept, developed by IFD (TU Dresden, Germany), combines the most commonly investigated operating mode with magnetic actuation and mechanical resetting by a spring with a lever valve. This concept is later referenced by push design and the design steps are described in details in section 3 with focus on an optimization of the actuator. The second valve concept, developed by ETO MAGNETIC GmbH (Stockach, Germany), consists of an air-core coil with a MSM element in its centre which acts as the valve body to open the valve during its compression. This concept is later referenced by pull design. This concept is briefly described in section 4.

2 Requirement Analysis and Conceptional Design

A prior investigation on the development of a competitive pneumatic fast switching valve based on MSM exists /6/. Therefore, a brief overview of valves primary applied for sorting issues, which are of the state-of-the-art is given and their features are discussed. Two designs with two MSM elements working antagonistically for opening and closing are presented and compared regarding the pressure drop and the needed displacement for opening. Both designs are validated by numerical approaches and by experimentation. For the preferred design according to this analysis a prototype is build and experimental tested. While the functionality of the concept could be proven at different upstream pressures $p_{u,abs}$ the aims on the dynamic and installation size could not be met yet for a voltage input of $U = 40 V$. In contrast to this prior investigation, the focus of this paper is the development of a fast switching valve not only for a pulsed air jet generation but also for control purposes. Based on the motivation to find a valve concept that can meet both purposes, the following requirement list Table 1, derived during a research of currently applied valves, is chosen to set the main development targets of this investigation. Most of these requirements can be considered in a morphological analysis, a very common method for designing by expanding a solution matrix of concept ideas. This concerns the requirements 1 to 5 from Table 1. The requirement on the valve function influences the configuration of the system components that geometrically influences the opening and closing process: the actuator consisting of a magnetic circuit and MSM element and the fluid stage. For the realisation of 2/2 valve with normally closed function, it is reasonable that opening process corresponds to
the magnetically activation of a mechanically pre-loaded MSM element. That means, the mechanic load (e.g. by a spring) holds the valve closed and only for opening an external energy effort is necessary for generating the magnetic field to activate the MSM element. The other requirements are quantitative considered to close the morphological analysis by an evaluation of all created concept ideas to choose a preference for further analysis and construction.

<table>
<thead>
<tr>
<th>No</th>
<th>Feature</th>
<th>Requirement</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Valve function</td>
<td>2/2 NC</td>
</tr>
<tr>
<td>2</td>
<td>Width (grid dimension)</td>
<td>Goal: w &lt; 18 mm</td>
</tr>
<tr>
<td>3</td>
<td>Nominal volume flow</td>
<td>$V_{\text{c}} = 100 \text{ NI/min}$</td>
</tr>
<tr>
<td>4</td>
<td>Operation pressure range (absolute)</td>
<td>$p_{\text{l,abs}} = 0 \ldots 8 \text{ bar}$</td>
</tr>
<tr>
<td>5</td>
<td>Switching time on/off</td>
<td>$\tau_{\text{sw,eff}} &lt; 1 \text{ ms}$</td>
</tr>
<tr>
<td>6</td>
<td>Maximum available supply voltage</td>
<td>$U_{\text{max}} = 48 \text{ V}$</td>
</tr>
<tr>
<td>7</td>
<td>Durability</td>
<td>500 Mio. cycles of operation</td>
</tr>
</tbody>
</table>

Table 1: Requirement list

For the evaluation based on a MSM element of dimension $[l \times w \times h]_{\text{MSM}} = 2 \times 3 \times 12 \text{ mm}$, the following criteria are of interest. The maximum feasible nominal diameter $d_{\text{nominal}}$ depends on the geometry of the opening area and the maximal stroke (assumed as $s_{\text{MSM,max}} = 0.5 \text{ mm}$) and force (assumed as $F_{\text{MSM,max}} = 9 \text{ N}$) of the MSM element and gives an orientation for scalability of the concept. The number and kinds of sealing are influencing the leak-tightness properties and the appearance of friction forces. The minimal feasible time for fluidic transient effects has a direct impact on the dynamic behaviour and can be estimated according to /?/. At least the minimum installation size is evaluated, which determines the minimum grid dimension of a valve terminal. Two valve concepts with different operation modes and valve principles turn out as promising to comply the requirements. Further analysis is described in section 3 and 4.

3 Valve Concept based on Push Actuator (Push Design)

The first valve concept is based on the operation mode of the MSM element with magnetically actuation and mechanical resetting by a spring with a lever valve. An actuator primary consisting of an iron core, at least one coil and one MSM placed in the air gap of the iron core, elongating in the magnetic field, is referenced as push actuator (Figure 1a). The concept of the fluid stage, shown in Figure 1b, allows the realization of a pressure balance resulting in a decreased mechanical load on the MSM element in both directions of motion and furthermore the flexibility to adjust the relation between stroke and actuation force according to the lever rule. The opening process corresponds to the expansion of the MSM element. The challenge in designing this concept is to satisfy the dynamic requirements despite the high complexity.

For the design process of the valve a numerical model of the whole system is necessary, because the working principle of the MSM element makes it difficult to separate the design of the fluid stage and the actuator. The working area depends on the force balance determined by the magnetic field strength of magnetic circuit, the mechanical load $P_{\text{mech}}$ influenced by the fluid stage and the twinning stress $\sigma_{\text{tw}}$ of the MSM element. The twinning stress of the MSM element leads to a hysteric behaviour, means the force needed for actuation is higher than the force needed for resetting, when the magnetic field is shut off. There are different approaches considering this magneto-mechanical hysteresis, e.g. based on physical approach in combination with experimental data /8/ or based on correlations of experimental data /9/. In this work we use a modified version of the hysteresis model proposed in /10/, which is implemented in the simulation software SimulationX. This model is able to describe the dependency of the strain $\varepsilon$ of a MSM element from the input magnetic field and mechanical stress $\sigma_{\text{load}}$.

The development of the fluid stage will lead to iterations in designing and testing, because it is the more critical sub-system with respect to the behaviour at high frequencies. There are different influences, e.g. friction, that cannot be modelled without parametrization by experimental data. In contrast, good developed methods are available to analyse the magnetostatic and transient behaviour of a magnetic circuit. In this paper the focus is on the optimization of the magnetic circuit with a MSM permeability of air as the worst case scenario referring to the stationary behaviour and the best case scenario referring to the dynamic behaviour of the magnetic flux.

3.1 Network model of the magnetic circuit

A commonly used simulation tool in designing an electromagnetic circuit is the finite element method, which enables the magnetostatic and transient analysis of distributed parameters of 2D or 3D models. But particularly transient simulations based on this method are very time expensive. For optimization issues regarding to satisfy the requirement list it is more efficient to find a description of the magnetic circuit model on 1D level which can be built in SimulationX as a reluctance model with concentrated parameters. In the following sections a counting index $i$ is generally used as a reference to the branches of the reluctance model.

3.1.1 Magnetostatic behaviour

Building a network model of the magnetic circuit as a 1D model with concentrated parameters requires a geometrical reference model. Therefore, a first functional prototype is designed by ETO based on the knowledge that the reluctance of the MSM element is at the maximum value when its elongation is zero (compressed state).

In this state, the relative permeability is $\mu_{\text{rel,MSM}} = 2$ and by experience, a magnetic field strength of $H_{\text{MSM}} = 500 \text{ 000 A/m}$ is necessary for actuation against mechanical load of $\sigma_{\text{load}} = 1.5 \text{ MPa}$. This is the worst-case scenario, because during the elongation the magnetic conductivity rises with the number of oriented easy axis of the twinning phases in direction of the magnetic flux. As a validation reference for the reluctance network a geometrical 3D model of the functional prototype (Figure 2a) is created in ANSYS Maxwell, because in contrast to the functional prototype it enables the visualisation of the magnetic flux distribution in all locations. First a magnetostatic simulation of the ANSYS 3D model with a parameter analysis of the magnetic flux density distribution against the magnetomotive force $\Theta = (0.2: 0.2: 10) \text{ kA}$ was performed. Exemplary the results of the parameter analysis for the specification of $\Theta = 500 \text{ A}$ and $\Theta = 4000 \text{ A}$ are shown in Figure 2b. Two findings could be achieved:
1. The magnetic circuit already passes the magnetic saturation state (maximum magnetization of the iron core) for a magnetomotive force $\Theta$ much smaller than the expected maximum value of the actuator, estimated by formula (1).

2. During the magnetic saturation state, the geometrically expansion of stray fluxes $\Phi_{s2}$ increases, and are not negligible anymore. Important for a reluctance model suitable for optimization issues, all stray fluxes need to be considered.

$$\theta = \frac{U_{\text{max}}}{R_s,\mu_{\text{base}}} = \frac{48 \text{ V}}{1.75 \Omega} = 28 \text{ A}$$

(1)

To create a reluctance network model, a representative structure needs to be found, which is detailed enough to catch the distribution of the magnetic fluxes in the iron core $\Phi_{a1}$ and the stray fluxes through the air $\Phi_{s2}$. Regarding to the results of the magnetostatic simulation the iron core is divided into segments (Figure 3a) that will be described as reluctances $R_{m,a1}$. Then, surfaces normal to the magnetic fluxes which are representative for the iron core segments and the stray fluxes and which also need to be modelled as reluctances $R_{m,s2}$, are defined as illustrated in Figure 3d. These fluxes can be calculated by integration of the normal magnetic flux density $B$ over these surfaces. Based on these definitions the parameter analysis of the 3D model for $\theta = (0.2; 0.2; 10)$ kA has to be repeated to create the reference data for the validation of the reluctance network. According to the first of Kirchhoff’s circuit laws, junctions can be identified, resulting in the network structure shown in Figure 3e.

For optimisation issue it is essential that all magnetic resistances are described as a function of geometric parameters. While the magnetic resistances of the iron core segments can be clearly delimited, the three-dimensional expansion of the stray fluxes rises with increasing saturation of the iron core as mentioned and shown in Figure 2b. The magnetic resistance for easy geometries, e.g. like stray flux $S1$ in Figure 3b, can be described as a function of a characteristic mean length $l_{\text{m}}$, longitudinal and area $A_{\text{m}}$ normal to the magnetic flux:

$$R_{m} = \frac{l_{\text{m}}}{\mu_0 \mu \mu_{\text{m}}}$$

(2)

In [11] a summary of formulas for different geometric shapes is given, e.g. for stray fluxes over poles of a magnet (suitable for stray flux S4 in Figure 3c). The relative permeability $\mu_r$, characteristic for the dependency of the magnetic flux density $B$ on the field strength $H$ (formula (3)) is constant $\mu_{r,a1} = 1$ for air, meaning to be equal to the permeability in vacuum $\mu_0$. In contrast the relative permeability of magnetic materials is non-linear dependent on the field strength. For this case the BH-curve of the iron material was measured and the experimental data were used within the simulations.

$$B = \mu_0 \mu H$$

(3)

For the validation of the network model, implemented in Simulation X, the magnetic fluxes calculated by it are compared to the solution of the 3D model implemented in ANSYS. Essential for the design procedure of the actuator is the magnetic flux density in the air gap $\Phi_{\text{air}}$, where the MSM element will be placed as sketched in Figure 1. Figure 4a shows a good agreement between the 1D reluctance network and the 3D ANSYS model for the full observed range of magnetomotive force $\Theta$. It depicts the calculated magnetic fluxes on different locations of the ANSYS model and the appropriate magnetic fluxes of the reluctance model. The maximum deviation of the magnetic flux in the air gap, shown in Figure 4b, is around 7 %. The value of the deviation depends on the magnetomotive force and can be explained by the assumption of the stray flux resistances as constant values. Below the saturation state of the iron core the stray fluxes are small and in the saturation state their ratio to the total flux $\Phi_{\text{total}}$ rises, leading to an increasing influence on the approximation error.
The dynamic behaviour of the actuator depends on the geometry and the material properties of the iron core and the resistance of the coils. The electrical conductivity of the iron core leads to induction of eddy current. These in turn induce secondary magnetic fluxes in opposite direction to the main flux resulting in a delay of the magnetic flux build-up. Analog magnetic resistances eddy currents can be modelled as electrical resistances $R_{el}$ as a function of a characteristic length $r_{eddy}$ and area $A_{eddy}$ on formula (4), operating as magnetic inductances $L_m$. Further details in modelling eddy currents can be looked up in [11].

$$R_{el} = \frac{1}{L_m} = \frac{r_{eddy}}{\bar{v}_p A_{eddy}} \quad (4)$$

For the validation of simulation models, both the 1D reluctance network and the 3D model, the magnetic flux density in the air gap of the functional prototype of the actuator, controlled by a peak and hold voltage signal, is measured. To ensure numerical stability of the simulation models, the signal was smoothed to a mean voltage value during the hold phase. The results are plotted in Figure 5 and show a good accordance of the electrical current and the magnetic flux density about the stationary value during the hold phase. While the dynamic behaviour of the electrical current is well caught by the simulation models (Figure 5b), a delayed behaviour of the magnetic flux density of the 1D reluctance model compared to the 3D and the measurement can be observed. Perhaps, this can be explained by simplifying the geometry of the magnetic resistances.

### 3.2 Optimization of the magnetic circuit

The response time of the valve for switching on $\tau_{on}$ and off $\tau_{off}$ is influenced by the dynamic behaviour of the magnetic circuit. Thus, the aim of the optimization is to improve the dynamic behaviour of the actuator, with respect to the build-up of the magnetic flux in the air gap $\Phi_{Air}$ while limiting the installation size of the actuator, especially the width, which determines the minimum possible grid dimension of a valve terminal. As a characteristic value the rise time $\tau_{ref}$ is introduced defined by the time interval bounded by time $t_{90\%}$, where the magnetic circuit is provided with a voltage step signal and the time $t_{ref}$, where the magnetic flux density reaches 90% of the stationary value. Beside the target of minimization of the rise time, two further criteria need to be considered: the stationary end value of the magnetic flux density and the required energy input. A minimum of the magnetic flux density needs to be considered during the optimization to ensure the actuation of the MSM element. Parallel the energy input should be limited to avoid overheating. These targets can be considered by minimizing: the time integral of the absolute deviation between the solution of the magnetic flux density $B_{ref}$ to a reference value $B_{ref}$, the time integral of the power input $P\_in$, defined as the product of the input voltage $U$ and the current $i\_in$, equation (6). The reference value $B_{ref}$ equals to the magnetic flux density in the air gap of the functional prototype. The installation size can be considered either as boundary condition or as a third objective (equation (7)).

$$y_1 = \int_0^{r_{end}} |B_{ref} - B(t)\rangle \ dt$$
$$y_2 = \int_0^{r_{end}} P_{in}(t) \ dt = \int_0^{r_{end}} U(t) \cdot i(t) \ dt = \int_0^{r_{end}} \frac{U(t)^3}{R_{coil}} \ dt$$
$$y_3 = l \cdot w \cdot h$$

This leads to an optimization task with a two-way negative effect of the two objectives $y_1$ and $y_2$. A high input power $P_{in}$ increases the dynamic behaviour of the magnetic circuit and leads to higher stationary end value of the magnetic flux density $B_{ref}$. A compromise of the two objectives needs to be found. The definition of the target function and the boundary conditions depend on the type of optimisation algorithm, whether it’s a local or a global algorithm. For the use of a local algorithm two objectives need to be concentrated to one value and the initial values have a great impact on the solution. Global algorithms are less dependent of the initial guess of the parameters and can be multibjective, but also their computing time is higher than that of the local optimization algorithms. It is reasonable to combine the advantages of a local and a global algorithms by following a two-step optimization, as listed in Figure 6, which gives an overview of the approached optimization.

The optimisation is performed by the software MATLAB, which offers lots of optimisation tools and enables the editing, controlling and evaluation of the reluctance model in SimulationX via COM-interface (component object model). Therefore, the reluctance network, derived in section 3.1 and created in SimulationX is used to generate the objectives values $y_1$ and $y_2$ in every iteration of the optimization algorithm. The network model is connected to a coil, provided by a voltage signal as a step function from $U = 0$ to the maximum available voltage $U_{max} = 48 V$ according to the requirement list (Table 1). The objective $y_3$ is calculated as a function of the
parameter setting $x_{opt+1}$ for each iteration. For the first step of optimization the MATLAB algorithm `gamultiobj` is suitable to find an initial guess of the parameters inserted for the second optimization step. This algorithm is based on a so called

![Diagram](image)

**Figure 6: Optimization proceeding**

*pareto optimization*, characterized by the phenomenon that one objective cannot be minimized without increasing the other objective. This phenomenon can be seen in Figure 7, which shows the solutions for the electrical input energy $E_u$ and the rise time $\tau_{90\%}$ for a set of parameter settings $x_{opt}$. For the second step of optimization, the algorithm `patternsearch` is used, which is a simple local optimization tool and allows the specification of boundaries and relations for the parameter settings $x_{opt}$ to avoid geometric invalid solutions. Nevertheless, at the end of the optimization a verification and analysis of the solution are important.

![Diagram](image)

**Figure 7: Solution set of Pareto optimization sorted by time constant $\tau_{90\%}$**

### 3.3 Discussion of the optimization results

For the verification and analysis of the solution parameter settings $x_{opt+2}$, generated by the optimization, a new 3D model in ANSYS is created by duplicating the 3D model of the functional prototype and adapting the geometric dimensions. A transient simulation is made for both, the functional prototype and the optimized actuator, with the same input voltage as applied to the 1D SimulationX model during the optimisation. The results for the magnetic flux density distribution are shown in Figure 8, in which the two models are depicted with realistic relation to each other. In Table 2 the interesting characteristics regarding to the requirement list from Table 1 are shown in order to compare the optimized actuator to the functional prototype and evaluation of the optimization. It can be seen, that the deviations between the 1D SimulationX and 3D ANSYS simulations are of same scale, meaning the derivation of a 1D reluctance model based on concentrated geometrically dependent parameters is a good approach to describe the transient analysis of a reference 3D model for modified parameters. Furthermore the success of the optimization can be observed. The rise time $\tau_{90\%}$ could be decreased about 1 ms, when also minimizing the installation size and restricting of the electrical energy input.

![Diagram](image)

**Figure 8: Comparison of the distribution of the magnetic flux a) functional prototype, b) optimization result based on $B_{ref} = 0.7$ T**

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Functional Prototype</th>
<th>Optimized Actuator</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stationary end value of magnetic flux density in air gap $B_{air_end}$</td>
<td>0.7 T</td>
<td>0.68 T</td>
</tr>
<tr>
<td>Rise time $\tau_{90%}$</td>
<td>2.1 ms</td>
<td>2.5 ms</td>
</tr>
<tr>
<td>Electrical energy input</td>
<td>10.8 J</td>
<td>10.8 J</td>
</tr>
<tr>
<td>Width $x$ Length $x$ Height</td>
<td>(15 $x$ 35 $x$ 37) mm</td>
<td>(11 $x$ 38.3 $x$ 27.8) mm</td>
</tr>
</tbody>
</table>

*Table 2: Comparison of the characteristics between the functional prototype and the optimized actuator*

As explained in the beginning of this section, the design of the part systems of the valve (fluid stage and the actuator) cannot be separated because of the hysteric behaviour of the MSM element. Therefore, a numerical model of the whole system is build in SimulationX including the 1D reluctance model and the material model of the MSM /10/ working against a spring. This model allows the comparison of the functional prototype and the optimized actuator to an appropriate experiment with the functional prototype. The solutions of the stroke are shown in Figure 9. In the SimulationX models the measured voltage of the prototype is provided as an input voltage, but this signal equals to the induced voltage for the functional design dependent on the characteristic of the coil and the electric circuit. The negative voltage peak between $t = 10$ ms and $t = 12$ ms arises during the
discharge of the current in the coil. In reality no negative magnetic flux would occur like in the simulation of the optimized actuator. The discharge of the coils would rather lead to a less negative potential.

That is why the MSM element in the simulation model is less reset than in the experiment. In the end, the results reflect the expected decrease of the rise time between the functional prototype and optimized actuator of about 1 ms.

### 4 Valve Concept based on Pull Actuator

In the second valve concept, the so-called pull design, the MSM actuator is integrated into the fluid stage and basically consists of a cylindrical air-cooled coil wrapped around an MSM element. A sketch of the valve is shown by Figure 10. The inlet and the outlet are placed near the tips of the MSM element, which is the actuating unit of the valve. The MSM element has been welded on both sides to two metallic supports. The coil is dimensioned to provide enough magnetic excitation to the element along the axis of motion. In this configuration, the magnetic field forces the element to go into the compressed state, generating a “pull” force on the supports, which will move accordingly compressing the spring. The preload spring, this time placed around the MSM element, is dimensioned to restore the element to full elongation once the magnetic excitation is removed.

The corresponding FEM simulation (with the software FEMM) is shown in Figure 11, the flux density measurement is taken in the middle of the MSM element. However, it must be remarked that this “pull” design has been developed for pulsed actuation, meaning that it needs big current pulses but for very short times (realistically under 2 ms). Therefore, the pull design addresses other applications than the push design.

![Figure 9: Validation of the simulation of a valve operation cycle](image)

Despite this assumption, the comparison shows a good accordance in rise time and maximum stroke between the simulation and experimental model of the functional prototype. But as already mentioned in section 3.1.2 the dynamic behaviour in the reluctance model is not described well enough regarding to a whole operation cycle.

### 5 Summary and Conclusion

In this paper an investigation on the design process of a fast switching pneumatic valve based on a MSM actuator is presented. This includes a requirement analysis of commonly used valves for selected applications, a morphological analysis with a short comment on the evaluation progress to choose a preference design concept.

As a result two innovative competitive valve concepts that are promising to satisfy the requirements are introduced: a lever valve concept with operating mode of magnetic actuation and mechanical resetting of the MSM element by a spring (push design) and a seat valve consisting of an air core coil with a MSM element in its centre which acts as the valve body to open the valve during its compression (pull design). The first valve concept is characterised by a lower dynamic behaviour compared to the second valve concept, but in return by less power input need for activation. Hence, control purposes are addressed with the first concept and sorting applications with the second concept, which leads to different approaches in designing.

For the valve with push design, an optimization method is presented, which proves as a suitable tool for the design of the actuator. The requirements according to Table 1 can be nearly met. The optimized actuator is a compromise of the rise time, magnetic flux density in the air gap and the electrical input and does not represent the final design.

In this paper, a reference value for the electrical energy input is considered as an orientation, which poses a limiting parameter for the optimization task. In reality this limitation is given by overheating of the system. With a thermal analysis a more realistic reference value of the electrical energy input could be established in order to shift the limitations set in this paper, maybe enabling a further decrease of the rise time. Also the reference value of the magnetic flux density can be influenced, because the magnetic flux needed for activation of the MSM element depends on the mechanic stress. This means, the optimization needs to be reviewed and maybe repeated during the design process of the fluid stage using a numeric model of the whole system.

For the concept with pull actuator a design of the whole valve is presented, created based on a 2D FEM model with rotationally symmetry conditions. The simulation shows that a very high current pulse between 40 and 80 A is necessary to excite the MSM element properly with a magnetic flux density between $|B| = 1 \text{T}$ and $|B| = 1.4 \text{T}$ (see Figure 11, the flux density measurement is taken in the middle of the MSM element). However, it must be remarked that this “pull” design has been developed for pulsed actuation, meaning that it needs big current pulses but for very short times (realistically under 2 ms). Therefore, the pull design addresses other applications than the push design.
# Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_{\text{open}}$</td>
<td>Opening area of valve</td>
<td>$[\text{m}^2]$</td>
</tr>
<tr>
<td>$A_w$</td>
<td>Characteristic Cross Area of Eddy Current Flux</td>
<td>$[\text{m}^2]$</td>
</tr>
<tr>
<td>$\bar{B}$</td>
<td>Magnetic Flux Density</td>
<td>$[\text{T}]$</td>
</tr>
<tr>
<td>$d_n$</td>
<td>Nominal diameter</td>
<td>$[\text{m}]$</td>
</tr>
<tr>
<td>$F$</td>
<td>Force</td>
<td>$[\text{N}]$</td>
</tr>
<tr>
<td>$h$</td>
<td>Installation Height</td>
<td>$[\text{m}]$</td>
</tr>
<tr>
<td>$i$</td>
<td>Electrical Current</td>
<td>$[\text{A}]$</td>
</tr>
<tr>
<td>$L_m$</td>
<td>Magnetic Inductivity</td>
<td>$[\text{H}]$</td>
</tr>
<tr>
<td>$l_w$</td>
<td>Characteristic Length of Eddy Current Flux</td>
<td>$[\text{m}]$</td>
</tr>
<tr>
<td>$p_{\text{1,abs}}$</td>
<td>Operation Inlet Pressure (absolute)</td>
<td>$[\text{bar}]$</td>
</tr>
<tr>
<td>$R_{\text{el}}$</td>
<td>Electrical Resistance</td>
<td>$[\Omega]$</td>
</tr>
<tr>
<td>$R_m$</td>
<td>Magnetic Resistance</td>
<td>$[\text{H}^{-1}]$</td>
</tr>
<tr>
<td>$s$</td>
<td>Stroke</td>
<td>$[\text{m}]$</td>
</tr>
<tr>
<td>$t$</td>
<td>Time</td>
<td>$[\text{s}]$</td>
</tr>
<tr>
<td>$U$</td>
<td>Electrical Voltage</td>
<td>$[\text{V}]$</td>
</tr>
<tr>
<td>$V_{\text{n}}$</td>
<td>Nominal volume flow</td>
<td>$[\text{NL/min}]$</td>
</tr>
<tr>
<td>$w$</td>
<td>Installation width</td>
<td>$[\text{m}]$</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>Function value</td>
<td>[-]</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>Strain of the MSM element</td>
<td>[-]</td>
</tr>
<tr>
<td>$\Theta$</td>
<td>Magnetomotive Force</td>
<td>$[\text{A}]$</td>
</tr>
<tr>
<td>$\kappa_{\text{fe}}$</td>
<td>Electrical Conductivity of Iron</td>
<td>$[(\Omega \text{m})^{-1}]$</td>
</tr>
<tr>
<td>$\mu_R$</td>
<td>Relative Permeability</td>
<td>[-]</td>
</tr>
<tr>
<td>$\mu_0$</td>
<td>Permeability of vacuum</td>
<td>[-]</td>
</tr>
<tr>
<td>$\sigma_{\text{cond}}$</td>
<td>Mechanical stress</td>
<td>$[\text{MPa}]$</td>
</tr>
<tr>
<td>$\sigma_{\text{tw}}$</td>
<td>Twining stress</td>
<td>$[\text{MPa}]$</td>
</tr>
<tr>
<td>$\tau_{90%}$</td>
<td>Time constant</td>
<td>$[\text{s}]$</td>
</tr>
<tr>
<td>$\tau_{\text{on/off}}$</td>
<td>Switching time on/off</td>
<td>$[\text{s}]$</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>Magnetic flux</td>
<td>$[\text{Wh}]$</td>
</tr>
</tbody>
</table>

## References


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Closed-loop control algorithm for fast switching pneumatic valves

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In this paper, a control algorithm for PWM based control of fast switching pneumatic solenoid valves is studied on the basis of the measured fluid flow characteristics. The dynamic nonlinear behaviour of fast switching valves is analysed using state-of-the-art mass flow sensors. The minimum PWM pulse width and nonlinear flow characteristics depending on PWM pulse width and pressure difference are observed. On the basis of the experiment data a new intelligent control algorithm based on the customized bilinear interpolation method is developed and tested on pneumatic muscle.

Keywords: Fast pneumatic switching valves, PWM modulation, flow characteristics, algorithm

Target audience: Pneumatics

1 Introduction

Pneumatic actuators are widely used in industry and are generally used for two position controls. Most of the time when the continuous position control is needed, pneumatic servo or proportional valves are used. Pneumatic servo valves are expensive and proportional valves do not have the fastest response time due to the spool dead band. The alternative is to use fast switching valves with digital control techniques trying to achieve linear flow control characteristics with the fastest possible response. The implementation of fast switching valves for position control using digital control techniques has been in development for the last 10 years. The main reasons for the use of the PWM control method for fast switching valves are the shortening of the response time, miniaturization of the valve control pistons and advanced electronics.

Many researchers used PWM control techniques for driving pneumatic switching valves with good results. The used PWM signal frequencies depend on the valve response time and are generally between 20–100 Hz [1]-[6]. Some efforts were made to develop electro-pneumatic valve models based on the electrical and pneumatic parts modeling and to use these models in a PWM driven pneumatic system [2], [4], [7]. The relationship between the PWM pulse width and the fluid flow was always defined to be linear, and only in one paper, the minimum PWM pulse width. The analysis of fast switching valves for position control using digital control techniques has been in development for the last 10 years. The main reasons for the use of the PWM control method for fast switching valves are the shortening of the response time, miniaturization of the valve control pistons and advanced electronics.

The pressure sensor, the fast switching valve and the air mass flow sensor are connected with piping that has a 4 mm internal diameter. To avoid air turbulence the components must be mounted at least 100 mm apart, as shown in Fig 2.

2 Experimental setup

The dependence of the fluid flow characteristics on the PWM pulse width and frequency needs to be accurately measured and modeled. Therefore, the experimental analysis is made at different input pressure values (ranging from \( P = 0.1 \) bar to \( P = 6 \) bar) and also different PWM frequencies and pulse widths. The response time of \( 1 \) ms enables us to use faster PWM frequencies. The analyzation frequencies are \( f_{\text{PWM}} = 200, 250 \) and \( 300 \) Hz. Fig 1. shows the experimental setup scheme and all used electrical and pneumatic components where: A is air preparation, B the pressure regulator, C the pressure sensor, D the fast switching valve, E the air mass flow sensor, F the PC with Matlab and G the CX controller with modules for pressure sensor and PWM modulation.

\[ V_s = \begin{cases} P_i C \left( \frac{T_0}{T_1} \right) \sqrt{1 - \left( \frac{P_2}{P_1} \right)^2} & \text{for } \frac{P_2}{P_1} > b \\ P_i C \left( \frac{T_0}{T_1} \right) & \text{for } \frac{P_2}{P_1} \leq b \end{cases} \]  

(1)

Where \( V_s \) represents the volume flow (m³/s), \( P_i \) the absolute inlet pressure (Pa), \( P_2 \) the absolute outlet pressure (Pa), \( C \) the acoustic conductivity (m³/(s·Pa)), \( T_0 \) the ambient temperature, \( T_1 \) the temperature of inlet air, \( b \) the critical ratio. The data for the tested valve MHJ10-LF is: \( C = 2.6167 \times 10^{-6} \) m³/(s·Pa) and \( b = 0.433 \). But this model does not describe what happens when the valve is controlled with the PWM signal and the valve is in constant transit states between being opened and closed. Therefore, this model can be used only when the valve is fully opened.

In this paper, we propose a new control algorithm for fast switching pneumatic valves which is based on the fluid flow characteristics measurements of the fast pneumatic switching valve MHJ10-LF. This valve has a response time less than \( 1 \) ms. The control algorithm includes the data about the response time which depends on the pressure difference, the optimal PWM control signal frequencies and the dependence of the flow characteristics on the PWM pulse width. The algorithm enables us to get the fastest response possible and also allows the fluid flow control with linear dependence on the control signal. It will be used in the future for the contraction control of pneumatic artificial muscles which have variable dynamic characteristics and need a very fast control loop.

Fig. 1: Schematic diagram of the experimental setup

The pressure sensor, the fast switching valve and the air mass flow sensor are connected with piping that has a 4 mm internal diameter. To avoid air turbulence the components must be mounted at least 100 mm apart, as shown in Fig 2.

Fig. 2: Main pneumatic components connected: A – air mass flow sensor, B – fast switching valve, C – pressure sensor.
The most important part of the experimental setup is the air mass flow sensor F-112AC by Bronkhorst. Its measuring range is $0 \leq \dot{m} \leq 199.99 \text{nl/min}$ with an accuracy of ±0.1 % Full Scale (FS). The tested valve, MH10-MF, is the fastest switching valve from Festo with the following data: nominal flow rate of $\dot{m}_{N} = 100 \text{nl/min}$, response time for opening $t_{on} = 0.7 \text{ms}$, closing time $t_{off} = 0.5 \text{ms}$ at the pressure difference $\Delta P = 0.1 \text{ bar}$, and $t_{on} = 0.9 \text{ ms}$, closing time $t_{off} = 0.4 \text{ ms}$ at $\Delta P = 6 \text{ bar}$. The opening time, $t_{on}$, increases and the closing time, $t_{off}$, decreases with the increase of the pressure difference due to the internal structure of the valve. The valve is normally closed due to pressure forces acting on the spool. The increased pressure difference increases the necessary electromagnetic force to open the valve and also closes the valve more rapidly. The pressure difference is set manually with the help of the pressure regulator and the pressure sensor positioned before the valve. In this way, the accuracy of the set differential pressure is $\Delta P = 0.05 \text{ bar}$.

3 Measurement of the fluid flow characteristics

The experimental analysis is performed under three different frequencies and pulse widths. The PWM pulse width is defined with the percentage of the PWM period. The PWM pulse width is changed in steps of 1 % for the range 0 to 10 % and with steps of 10 % from 10 to 100 % of pulse width. The minimum pulse width when the air mass flow sensor detects the flow is set with the accuracy of 0.1 % of the pulse width. The executed measurement is shown in Table 1.

<table>
<thead>
<tr>
<th>PWM frequency [Hz]</th>
<th>$\Delta P$ [bar]</th>
<th>Pulse 0 – 10 %</th>
<th>Pulse 10 – 90 %</th>
<th>Min. pulse [%]</th>
<th>Max. pulse [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>200</td>
<td>0.10</td>
<td>Manual with 1 % step</td>
<td>Manual with 10 % step</td>
<td>0.1 % step</td>
<td>10 % step</td>
</tr>
<tr>
<td>250</td>
<td>0.30</td>
<td>Manual with 1 % step</td>
<td>Manual with 10 % step</td>
<td>0.3 % step</td>
<td>3 % step</td>
</tr>
<tr>
<td>300</td>
<td>0.50</td>
<td>Manual with 1 % step</td>
<td>Manual with 10 % step</td>
<td>0.5 % step</td>
<td>5 % step</td>
</tr>
<tr>
<td>350</td>
<td>0.60</td>
<td>Manual with 1 % step</td>
<td>Manual with 10 % step</td>
<td>0.6 % step</td>
<td>6 % step</td>
</tr>
</tbody>
</table>

Table 1 – Measurement plan for PWM control signal

The measurement is made for every frequency in the following order:

1. Set the PWM frequency, $f_{PWM}$.
2. Set the desired pressure difference, $\Delta P$, with the pressure regulator and fine tune using the pressure sensor for reference.
3. Increase the PWM pulse, $t_{PWM\%}$, until the flow is detected.
4. Increase the $t_{PWM\%}$ from 0 to 10 % with 1 % steps and save the measured flow.
5. Increase the $t_{PWM\%}$ from 10 to 90 % with 10 % steps and save the measured flow.
6. Find the maximum pulse width, $t_{PWM\%}$, that still has an effect on the flow.
7. Set the new pressure difference, $\Delta P$, and repeat steps 3. to 7.
8. Repeat steps 1. to 7. for all frequencies, $f_{PWM} = 200, 250$ and 300 Hz.

3.1 Experimental results

All gathered data are written in Matlab data matrices for further analysis. The pulse width, $t_{PWM\%}$, is converted to time, $t_{PWM\%} \Delta P$, for better compatibility between frequencies. Fig. 3(a) presents the results for the measured flow at the PWM frequency $f_{PWM} = 200 \text{ Hz}$ and in Fig. 3(b), detail A of the pulse width from $t_{PWM\%} \Delta P = 0.1$ to 0.5 ms is shown.

The data for $f_{PWM} = 250 \text{ Hz}$ and $f_{PWM} = 300 \text{ Hz}$ are similar and therefore the comparison is needed to choose the best frequency. The results of the measured flow characteristics show that the minimum pulse width increases with the increase of the pressure difference. The minimum pulse width increases from 0.14 ms to 0.4 ms. In Fig. 4(a), the comparison of all three frequencies and different pressure differences is given.

The minimum pulse width is used to define the minimum pulse width as part of the new control algorithm. The widest pulse that has an effect on the flow characteristics is the function of frequency because the frequency determines the largest possible width of the pulse. The experimental results of the maximum pulse width are presented in Fig 4(b).

![Fig. 3: (a) Test results for the PWM frequency $f_{PWM} = 200 \text{ Hz}$, (b) Detail A (of Fig. 3) for the PWM freq. $f_{PWM} = 200 \text{ Hz}$](image)

![Fig. 4: (a) Minimum PWM pulse, $t_{PWM\%} \Delta P$, as a function of $\Delta P$. (b) Maximum PWM pulse, $t_{PWM\%} \Delta P$, as a function of $\Delta P$.](image)
The presented data show that there are differences between frequencies but it is hard to see them at lower pressure differences. To choose the optimum PWM frequency all the data is normalized. This is done by dividing the mass air flow data with the maximum air flow of all pressure differences and by converting the PWM pulse width from milliseconds to a percentage value. In Fig. 6, the comparison of all three different frequencies at three differential pressures is shown.

It is possible to see that at lower pressure differences the flow through the valve has a very steep curve which normalizes at about 10% of the pulse width and then progresses relatively linearly. At higher pressure differences, the characteristics are almost linear. To achieve the best control using fast pneumatic valves and PWM modulation it is crucial to have linear characteristics also at the lower pressure differences. The smoothest transition from the closed valve to the first 10% of the PWM duty cycle is found for the frequency $f_{PWM} = 250$ Hz. The fact that the transition is the smoothest and also that the higher frequencies decrease the life span and the control resolution of the valve defines this as the optimal tested frequency.

The function does not accurately define the minimum pulse width. The steep first 10% of the pulse width data for the lower pressure differences and transformation to smoother curve is not fitted satisfactorily. The transformation between different pressure differences should be smooth but has local extremes. We are not able to mathematically calculate the inverse of some of the tested functions.

The next step is to use alternative numerical approaches that are based on experimental data and calculate the pulse width from interpolation. The most often used interpolation techniques include linear, spline and polynomial interpolation [11]. The testing of different interpolation techniques shows that bilinear interpolation in 3D data matrix is the most usable. The reason for this is because there are a lot of data points and therefore the spline or polynomial interpolation does not increase the accuracy. Also, the calculation is done on the controller and higher grade interpolation techniques can slow the control loop algorithm and therefore increase the possibility for a larger control error or instability of the system.

The fastest and the most accurate solution is the modified bilinear interpolation [12]. Due to the nature of the interpolation, we implement the detection of boundary conditions for the pressure differences. If the pressure difference is lower or higher than measured, then the algorithm uses the lowest or highest pressure used in the experiment. The control algorithm value restriction and the PWM pulse width calculation are shown in Fig. 7. The program, Matlab, already has interpolation modules, but since the algorithm will be used on CX Controllers, we need to design it ourselves on the basis of the bilinear interpolation. This way we will be able to transfer it from the Matlab to the ST programming language of the PLC controller.

The input for the algorithm with bilinear interpolation is the desired proportional opening (flow) in the given time. The algorithm works in the following steps:

1. Gets pressure difference data, $\Delta P_o$.
2. Checks if the pressure difference, $\Delta P_o$, is in the correct range. If not, it changes the value to the nearest one.
3. Finds two closest pressure difference neighbors from the experimental data, $\Delta P_1$ and $\Delta P_2$.
4. Finds two closest neighbors of the valve flow for both pressure differences ($\Phi_{o1}$, $\Phi_{o2}$, $\Phi_{o3}$ and $\Phi_{o4}$).
5. Gets the PWM pulse width data for these four experimental points ($O_1$, $O_2$, $O_3$ and $O_4$).
6. Calculates the desired pulse width, $O_i$, with the use of the modified bilinear interpolation, as shown in Eq. 8.
If the pressure is exactly the same as the pressure of the experimental data, then the PWM pulse width is calculated using linear interpolation only. With the use of the modified bilinear interpolation and added boundary limits, the pulse width is defined on a large working area of pressures and flows. The algorithm is tested on an entire working spectrum to show if the interpolated points are positioned correctly and if the interpolated points between the measured pressure differences are calculated correctly and logically. Fig. 9 shows the experimental data and algorithm outputs with smaller dots, where the algorithm is tested not only for known pressure differences but also for pressures between the measured ones (ΔP = 0.3 bar, ΔP = 0.75 bar, …). The results show that the algorithm works very accurately without mistakes and interpolates correctly on the entire working area.

Fig. 9: Final bilinear interpolation used on the intelligent control algorithm.

5 Testing of control algorithm

The control algorithm for fast switching valves was tested on a linear actuator with a pneumatic muscle. The control signals for valves were:

- PID algorithm with output directly connected to PWM pulse width.
- PID algorithm with output being optimized with algorithm for fast switching valves that calculate appropriate PWM pulse width depending on PID signal and pressure difference.

Fig. 10 contains a scheme of all electric and pneumatic components of algorithm testing experimental setup.

Fig. 10: Scheme of fast switching valve controlling pneumatic muscle contraction for valve algorithm testing.
The static position error is also the result of input and output valve leakage because valves use hard seals on control pistons. The leakage increases with pressure difference and both valves must compensate for this. The PID+VA algorithm compensates for the leakage much better with oscillation around a reference point while plain PID algorithm settles at quasi static error.

The main difference between PID and PID with valve algorithm is in accuracy of static position. This is logical since the valve response time is shorter. The response time and static error for plain PID algorithm and PID with valve algorithm are presented in Table 1.

<table>
<thead>
<tr>
<th></th>
<th>PID</th>
<th>PID+VA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Response time, [s]</td>
<td>0.184</td>
<td>0.162</td>
</tr>
<tr>
<td>Overshoot, [mm]</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Position error, [mm]</td>
<td>0.38 ±0.160</td>
<td>0.07 ±0.15</td>
</tr>
</tbody>
</table>

Table 1: Analysis of PID and PID+VA pneumatic actuator response.

The PID+VA has a shorter response time and smaller position error compared to the plain PID algorithm. The reason is the shorter response time that is manifested in higher PWM duty cycles at the same position errors.

6 Summary and Conclusion

This paper presents the development of the closed-loop algorithm for the PWM flow control of fast switching pneumatic valves. The developed algorithm enables use of fast switching valves and as accurate control of flow characteristics as with servo valves. Experimental data of volume flow dependent on pulse width and pressure difference was used to develop control algorithm. Different mathematical approaches were tested to mathematically model experimental data. The most accurate was the bilinear interpolation that was used for algorithm development. The algorithm was tested on pneumatic muscle position control.

Using acquired experimental data, an algorithm was developed that takes into account the measured or calculated pressure difference of the inlet and outlet pressures to calculate the correct pulse width for a desired flow rate. The calculation of an appropriate PWM pulse width is done using a modified bilinear interpolation that finds the neighboring experimental values and then calculates the pulse width in relation to them. The Control algorithm for fast switching pneumatic valves was implemented on a controller of linear actuator with a pneumatic muscle and two fast switching valves. Actuator's system response actuator controlled with PID and PID with valve algorithm was tested on step reference signal. The system response analysis showed the accuracy increase by 81.5 %.

In this paper, we proved that the fast switching valves with algorithm that takes into account the minimal PWM pulse needed to open the valve and nonlinear flow dependent on the pulse width can increase the accuracy of the pneumatic actuators position control.
References


Transient simulation of a pneumatic sharp edged L-shaped pipe

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The increase of system dynamic within the area of pneumatics requires sophisticated numerical methods to determine the systems' performance. Cycle durations in the range of just a few milliseconds and below require the implementation of transient gas dynamic solvers to predict the systems behavior accurately and to save computational time. Yet, such solvers lack of accuracy for sharp edged elbows. This paper presents a hybrid approach using a one dimensional and a two dimensional finite volume Riemann-Solver. The results are compared to analytical acoustics theory and to a CFD approach using a turbulence model.

Keywords: System Simulation, Pneumatics, Numerical Solver, Gasdynamics
Target audience: Automation, Pneumatic Systems, Simulation

1 Introduction

In general, it is possible to calculate several flow parameters for transient pneumatic flows using computational fluid dynamics (CFD) software including turbulence modelling for instance for a highly dynamic automation application or a gaseous fuel injection system. Despite increasing processing power of modern computers solving particular problems is yet time-consuming. A simulation of a few milliseconds results in a computational time of several hours which makes the design of a highly dynamic pneumatic system resource consuming. Therefore, one dimensional numerical solvers based on Euler Equations are commonly used to achieve reasonably short simulation duration and yet retaining sufficient accuracy. However, such solvers are not capable of simulating the entire spectrum of flow regimes and geometries which occur in pneumatic components like sharp edged elbow fittings. The flow regime of a miter joint lies within a full reflection and transmission of the incoming mass flow and pressure waves. A one dimensional solver can only offer one of the two regimes, either full reflection or full transmission. Accordingly, a solution of a partially transmitted and partially reflected wave is not possible to compute. This paper presents an approach to simulate a sharp edged elbow using a two dimensional solver within the inner part of the joint which is coupled to one dimensional solver at the outer parts. Both solvers are based on the Flux Vector Splitting (FVS) approach proposed by Steger and Warming /1/.

This approach allows the calculation of a partial transmission and partial reflection of a miter joint time efficiently. The results are compared to acoustic theory and CFD solution in order to validate the solver.

2 Governing equations and numerical approach

To describe one dimensional gaseous flows Euler Equations are commonly used since for large Reynolds numbers viscosity influence can be neglected. These include the continuity equation, momentum and energy equation and are given by Equation (4). Herein, e is the specific energy described by Equation (2).

\[
\frac{\partial}{\partial t} \left( \frac{\rho}{\gamma} \right) + \frac{\partial}{\partial x} \left( \rho u + \rho u^2 + p \right) = 0
\]  
(1)

\[
e = \rho c_{p} T + \rho u^2
\]  
(2)

For a perfect gas the isochoric heat capacity \(c_v\) can be expressed via Equation (3) using the specific gas constant \(R\) and the heat capacity ratio \(\gamma\) of the given gas. Subsequently, the specific energy of the system can be described using pressure, density and velocity only, leading to Equation (4).

\[
c_v = \frac{R}{\gamma - 1}
\]  
(3)

\[
e = \frac{p}{\gamma - 1} + \rho u^2
\]  
(4)

This results in a decoupled system given by Equation (5) which consists of conservative variables \(U\). Herein, the function \(F(U)\) is the flux function of the mass, energy and momentum flux. The conservative variables \(U_i\) can be expressed via primitive variables \(\rho\) and \(u\) applying Equation (6) /1/.

\[
U_i + F(U)_i x = 0
\]  
(5)

\[
U_1 = \rho, \quad U_2 = \rho u, \quad U_3 = \frac{p}{\gamma - 1} + \rho u^2
\]  
(6)

To allow the occurrence of shocks and other discontinuities which, for instance, can be caused during the opening process of a valve it is necessary to use the conservative formulation of Eulerian Equations. If the time integration is done numerically and the computational domain is discretised (figure 1 a) the numerical scheme is given by Equation (7). Herein, \(i\) is the local index of a cell and \(n\) is the index of the current time step.

\[
\frac{B^i_{n+1} - B^i_n}{\Delta t} = \nabla \cdot \left( F(B^i_n) \right)
\]  
(7)

![Figure 1: Finite volume and explicit solver.](image)

The flux values of the inter cell fluxes \(F(B^i_{n+1/2})\) and \(F(B^i_{n-1/2})\) are calculated using the current averaged cell value \(i\) and its neighbours \(i-1\) and \(i+1\), which is illustrated in figure 1b. The flux formulation is chosen the way it is proposed by Steger and Warming /1/. This scheme utilises the hyperbolic character of Equation (1), which means that it possesses defined propagation speeds of information, namely the eigenvalues \(\lambda_1 = u - a\), \(\lambda_2 = u\) and \(\lambda_3 = u + a\). Herein, \(a\) is the speed of sound and can be calculated for a perfect gas applying Equation (8). For a more detailed description of hyperbolic systems and its properties the reader is referred to /2/.

\[a = \sqrt{\frac{
u}{\rho}} \sqrt{\frac{p}{\rho}}\]  
(8)

Steger and Warming construct a numerical inter cell flux which depends on the eigenvalues \(\lambda_i\) of the system and can therefore be given a defined propagation achieving a separation of positive and negative flux components described by Equation (9) wherein \(H\) is described by Equation (10).

![Figure 1: Finite volume and explicit solver.](image)
The incoming flux $F(\tilde{U})_i^{-1}$ at the left cell interface is composed of the positive part of the left cell’s flux and the negative component of the current cell, see figure 2 a,b. The outgoing flux $F(\tilde{U})_i^1$ is built with the positive flux component of the current cell and the negative component of the right cell.

Finally the entire numerical scheme can be summarised in Equation (11). To verify its accuracy for small disturbances the acoustic theory for a closed and opened resonator is applied. The modulation of boundary conditions for a closed and an open end for a finite volume solver is taken from /2/ and /3/.

\[
\tilde{\phi}^{n+1}_i = \frac{\Delta t}{\Delta x} \left( P^{n+1}_i + P^{n+1}_{i+1} - P^n_{i-1} - P^n_i \right)
\]

(11)

2.1 1-D Solver validity for small disturbances

To demonstrate the validity of the solver a pipe with the length $L$ and an open end is excited by a sinusoidal pressure signal at the very first cell. The frequency is chosen to be the second order harmonic oscillation $f_{open,2}$ Equation (12) /4/. If the solver does represent the physics correctly, three nodes and two anti-nodes should occur along the tube’s length. The first node has to be at the very beginning of the pipe, the second one has to be exactly in the middle of the pipe and the third one is to be expected at the end of the pipe. Whereas two anti-nodes have to be exactly in between the nodes.

\[
f_2 = \frac{a}{\Delta x}
\]

(12)

Figure 3 shows the pressure distribution over time and space in a pipe of the length $L = 0.2 \text{ m}$. To assure the validity of the acoustic theory the pressure disturbance is chosen to be very small (0.1 bar) compared to the mean pressure which is 10 bar. It can clearly be seen, that as predicted by the theory there are exactly three nodes at the beginning, in the middle and at the end of the pipe. In between the two anti-nodes are located.

Furthermore a second order oscillation for a closed end is examined (figure 4), whose frequency is given by Equation (13) /5/.

\[
f_2 = \frac{3a}{4\pi}
\]

As expected, the solver calculates the position of the nodes and anti-nodes as predicted by the acoustic theory.

To demonstrate the impact of a sharp edged elbow on the harmonic oscillation a 2-D solver is introduced in the following section.

3 Implementation of the 2-D solver

This chapter deals with the implementation of the two dimensional scheme and its coupling to the one dimensional solver in for an elbow. The solver is tested in conjunction with the acoustic theory for a closed end and an open end resonator.

3.1 Governing numerical scheme

To produce a partial reflection numerically within an elbow it is necessary to implement a two dimensional solver within the inner part of it. As presented before, the finite volume solver by Steger and Warming is taken into account for the two dimensional case as well. The increase of one dimension implies an additional momentum equation and flux $G$, yet the principle of the solver remains the same and is depicted in figure 5.
The numerical scheme for the solver is the same as for the one dimensional solver and if the local discretisation in x-direction is of the same size as for the y-direction the numerical scheme is presented in Equation (14).

\[
\Delta t^{n+1} = \Delta t - \frac{\Delta x}{\lambda_1} \left( F^{+}_{n+1,i,j} - F^{-}_{n+1,i,j} \right) - \frac{\Delta y}{\lambda_2} \left( G^{+}_{n+1,i,j} - G^{-}_{n+1,i,j} \right)
\]  \hspace{1cm} (14)

The numerical flux calculation requires the knowledge of the eigenvalues, i.e. the characteristic speeds. In the case of a two dimensional problem there are eight of them in total (four for each direction) instead of three for the one dimensional case. For the x-direction the first one is \( \lambda_1 = u - a \), the fourth one is \( \lambda_4 = u + a \) and the second and third are identical \( \lambda_2 = \lambda_3 = u \). Hence, to calculate the fluxes in x-direction for each cell Equation (15) is applied.

\[
F_{ij} = \frac{\rho_{ij}}{2\Delta x} \left( u_{i}^{2} + 2(u_{i} - a_{i})u_{i} + \left( \frac{a_{i}^{2}}{\gamma - 1} + \frac{a_{i}^{2}}{2} \right) \frac{\Delta x}{\Delta y} \right)
\]  \hspace{1cm} (15)

The eigenvalues \( \xi_{i} \) for the y-direction are calculated the same way as for the x-direction but instead of using the horizontal velocity \( u \) the vertical velocity \( v \) is applied leading to \( \xi_{1} = v - a \), \( \xi_{2} = v + a \) and \( \xi_{3} = \xi_{4} = v \). Finally the numerical flux in y-direction is constructed in Equation (16).

\[
G_{ij} = \frac{\rho_{ij}}{2\Delta y} \left( \xi_{i}^{2} + 2(u_{i} - a_{i})\xi_{i} + \frac{a_{i}^{2}}{\gamma - 1} + \frac{a_{i}^{2}}{2} + \frac{\gamma a_{i}^{2}}{2} \frac{\Delta y}{\Delta x} \right)
\]  \hspace{1cm} (16)

Since the additional velocity component \( v \) does not only contribute to an additional momentum but also to an additional kinetic energy, the energy flux has to consider that component which results in Equation (17) which is different to that of Equation (10) for the one dimensional case.

\[
K = \frac{1}{2} \left( \frac{\gamma a_{i}^{2}}{2} \frac{\Delta y}{\Delta x} \right)
\]  \hspace{1cm} (17)

Having defined the numerical fluxes for the two dimensional system, it is now possible to compute an elbow using the two dimensional solver only but obviously the computational time would increase by the power of two. To overcome that problem only the inner part of the elbow is calculated using the two dimensional scheme, which is depicted in figure 6. Here, the upper part is divided into a region with length \( L_{1D} \) where the one dimensional solver is applied and a two dimensional region \( L_{2D} \) connecting the adjacent ones. The computational effort is reduced thereby but the border between the regions needs special mathematical treatment.

Regarding Equation (11) the one dimensional cell at the border requires a neighbour cell of equivalent size to calculate the inter cell fluxes. Since the next neighbour cell belongs to the two dimensional solver the necessity of an averaging method arises. One could simply use an averaging of the primitive variables \( p, \rho, u, \) and \( v \) for the two dimensional cells but that would result in momentum and energy annihilation. This is caused by the fact that for the vertical one dimensional solver the horizontal velocity components of the two dimensional solver would be neglected. For the horizontal one dimensional border cell the same problem occurs for the vertical velocity components of the two dimensional solver, meaning that all vertical velocity components would disappear since they cannot be included into the one dimensional scheme.

Therefore, another approach is presented which prevents the annihilation. Instead of averaging the primitive variables an averaging of the conservative variables is presented and retransformed to the primitive variables using Equation (5) which is discussed more detailed in the following.

\[
\Delta \rho (\Delta x) = \frac{1}{2} \left( \rho_{1} + \rho_{2} \right) \Delta x
\]  \hspace{1cm} (18)

The total averaged mass \( M_{2D,vert} \) of the two dimensional cells at the border is, therefore, the sum of all cell masses in y-direction given by Equation (19).

\[
M_{2D,vert} = \sum_{j=1}^{j_{end}} \rho_{ij} \Delta \Delta y
\]  \hspace{1cm} (19)

To guarantee mass conservation this mass has to be exactly the same as the mass of a one dimensional cell with the width \( W \), the length \( \Delta x \) and the averaged density \( \bar{\rho} \) which is given by Equation (20). Equalising (19) with (20) leads to Equation (21) to calculate the averaged density for the two dimensional cells.

\[
\bar{\rho} = \frac{M_{2D,vert}}{W} \Delta \Delta y
\]  \hspace{1cm} (20)

\[
\bar{\rho}_{vert} = \frac{M_{2D,vert}}{W} \Delta \Delta y
\]  \hspace{1cm} (21)

For the momentum conversion two contributions have to be considered, the horizontal one (22) and the vertical one (23).

\[
I_{2D,vert} = \sum_{j=1}^{j_{end}} \rho_{ij} \Delta \Delta y
\]  \hspace{1cm} (22)
\[ I_{2D,\text{vert}} = \sum_{j=1}^{N} \rho_{j}v_{j} \Delta x \Delta y \] (23)

The sum of the momentum contributions \( I_{2D,\text{vert}} \) and \( I_{2D,\text{vert}} \) have to be exactly the momentum of an averaged cell with the length \( \Delta x \), the width \( W \) and averaged momentum \( \bar{p} \) given by Equation (24). Equalizing the sum of (22) and (23) with (24) leads to Equation (25), which is the averaged velocity in x-direction of the two dimensional solver.

\[ \bar{p} = \bar{u} \Delta x W \] (24)

\[ \bar{u}_{\text{vert}} = \frac{1}{\bar{p}_{\text{vert}}} \left( \sum_{j=1}^{N} \rho_{j}v_{j} + \sum_{j=1}^{N} \rho_{j}u_{j} \right) \Delta y W \] (25)

In order to obtain an averaged pressure for the two dimensional border region firstly the total energy given by Equation (26) is calculated and the averaged pressure \( \bar{p}_{\text{vert}} \) is extracted from it subsequently since the averaged velocity and density are known by now.

\[ \bar{E}_{\text{vert}} = \left( \frac{\bar{p}}{\rho_{1}} + \frac{1}{2} \bar{u}^{2} \right) \Delta x W \] (26)

The total amount of energy of the two dimensional cells along in the y-direction has to be calculated invoking Equation (27) which contains the vertical and the horizontal velocity components. The conservation of energy requires the equality of Equation (26) and (27) which results in Equation (28), giving an expression for an averaged pressure \( \bar{p}_{\text{vert}} \).

\[ \bar{E}_{2D,\text{vert}} = \sum_{j=1}^{N} \left( \frac{\rho_{j}u_{j}}{\rho_{j}} + \frac{1}{2} \rho_{j} \left( u_{j}^{2} + v_{j}^{2} \right) \right) \Delta x \Delta y \] (27)

\[ \bar{p}_{\text{vert}} = \sum_{j=1}^{N} \left( \frac{\rho_{j}u_{j}}{\rho_{j}} + \frac{1}{2} \rho_{j} \left( u_{j}^{2} + v_{j}^{2} \right) \frac{\Delta y}{W} + \frac{1}{2} \bar{u}^{2} \right) \left( 1 - \frac{1}{\kappa} \right) \] (28)

All primitive variables for the border region of the horizontal part of the elbow have been obtained avoiding momentum or energy annihilation. For the lower part i.e. for the averaging in x-direction the procedure is straightforward and will not be explained in detail here.

3.3 Numerical results using 2-D solver

A sharp elbow is examined in terms of the acoustic theory in this section. The horizontal pipe has the same length \( L = 0.1 \text{ m} \) as the vertical pipe. To validate the behaviour of the 2-D solver two cases, an open elbow and a semi closed elbow are presented.

The coupled 1-D solver is used for the front and back end of the elbow with the length of \( L_{1D} = 0.075 \text{ m} \). To reveal streamline curvature behind the elbow the 2-D solver is not only used for the actual corner but also for a region before and behind the corner with a length 0.05m.

Figures 7 and 8 show that no discontinuity at \( x = 0.075 \text{ m} \) and \( x = 0.125 \text{ m} \) is caused by the averaging method. Therefore, the mathematical coupling as described in section 3.2 results in an accurate numerical solution without causing numerical oscillation which would lead to artificial numerical pressure waves disturbing the pattern predicted by the acoustic theory.

In comparison to the acoustic behaviour of the sharp elbow to the straight pipe, the first cell is excited by the same sinusoidal pressure input as in section 2.1. With the second order harmonic oscillation \( f_{\text{open,2}} \) the solution for an open elbow is calculated. Figure 7 shows the pressure distribution over time and space.

The same pattern as in figure 3 is given. There are three nodes, one at the beginning, one in the middle and one at the end of the elbow. Additionally, there is an anti-node in between two nodes respectively. A difference to the 1-D solver is the pressure distribution in the horizontal pipe, from \( x = 0 \text{ m} \) to \( x = 0.1 \text{ m} \), which is slightly warped in comparison to figure 3.

The second case is a semi closed sharp elbow. The excitation frequency is the second harmonic frequency \( f_{L,\text{closed}} \) for a semi closed pipe. The expected pressure distribution is calculated by the 2-D solver (figure 8). The characteristic number of nodes and antinodes for a semi closed pipes can be seen. Two nodes, one at the beginning and one at two thirds of the elbow’s total length and two antinodes, one at one third and one at the end can be observed. Obviously the difference between the 2-D solver of a sharp edged elbow and 1-D solver is negligible which would not justify the computational effort for the acoustic case. In the following chapter another numerical test is presented.

![Figure 7: Open end elbow resonator test](image)

![Figure 8: Closed end elbow resonator test](image)

4 Closed shock tube test case

The shock tube is a classical test case for numerical Riemann solvers to examine its performance for discontinuities /6/, /7/. A pipe with an isothermal gas has an initial pressure and density discontinuity distribution in the middle. This initial set up allows to produce shock waves of arbitrary strength by increasing of the initial pressure difference resulting a higher amplitude of the shock wave and in a faster traveling speed which is always supersonic. It can be expected that in contrast to the acoustic waves which are of negligible altitude and
which travel with the speed of sound the reflexion of shock waves will be stronger within the sharp edge region of an elbow. Hence, the results for the two dimensional solver are expected to be different compared to the 1-D solution and reproduce the physics more exactly. To validate the results of the 2-D solver, a 3-D CFD simulation of the elbow is used.

4.1 Closed shock tube

To understand the effects of shockwave reflection in a sharp elbow, a numerical shock tube experiment in a closed straight pipe using the 1-D solver is presented. A pipe with a length of $L = 0.2\,\text{m}$, an initial temperature $T_0 = 300\,\text{K}$, the left region with high pressure $p_{\text{high}} = 16\,\text{bar}$ and the right region with low pressure $p_{\text{low}} = 1\,\text{bar}$ is considered. Figure 9 shows the pressure distribution at four different time steps. The initial distribution (blue line) causes a rarefaction wave travelling to the left and a shockwave travelling to the right (red line). As soon as the shockwave hits the wall it is reflected and moving to the left (black line). The same phenomenon occurs when the shockwave hits the left wall. At $t = 0.058061\,\text{ms}$ (green line) the rarefaction wave has been reflected, travelling to the right and interacts with the shockwave.

![Figure 9: Pressure distribution of a closed Shock Tube.](image)

4.2 Closed shock elbow

Now, a numerical shock tube experiment for an elbow with a total length of $0.1\,\text{m}$ is presented, figure 10 depicts its initial conditions. The virtual membrane separating high and low pressure regions is located at $x = 0.03\,\text{m}$. The high pressure region is on the left side and the low pressure region on the right side of a membrane. As depicted in figure 10 the pressure at the three following locations is recorded: $x = 0.0099\,\text{m}$, $x = 0.04\,\text{m}$, $y = 0.02\,\text{m}$. The 3-D CFD simulation has been calculated with the solver FLUENT 17 using an explicit solver and applying the k-epsilon model.

![Figure 10: Shock tube experiment in an elbow.](image)
especially near the sharp edge at $x = 0.0399\, \text{m}$. Table 1 summarises the mean deviation over time for each virtual pressure sensor. The two dimensional solver assures a deviation lower than 7\%, whereas the deviation of the 1-D model is between 12\% and 29\% and is therefore unacceptable compared to the 2-D solver solution.

<table>
<thead>
<tr>
<th>Pressure Sensor Position</th>
<th>1D</th>
<th>2D</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x = 0.009, \text{m}$</td>
<td>12.4%</td>
<td>6.93%</td>
</tr>
<tr>
<td>$x = 0.0399, \text{m}$</td>
<td>28.8%</td>
<td>3.11%</td>
</tr>
<tr>
<td>$y = 0.02, \text{m}$</td>
<td>12.88%</td>
<td>7.16%</td>
</tr>
</tbody>
</table>

Table 1: Mean pressure deviation in reference to CFD

5 Summary and Conclusion

This paper presents an approach to modulate a sharp edged elbow using a coupling of a two dimensional and a one dimensional finite volume solver. The approach is validated in two ways: For small pressure disturbances analytical solution from the acoustic theory is compared to the solver’s solution and for large pressure disturbances Sod’s Shock Tube experiment for a closed pipe is examined. It has been revealed that for the acoustic case, i.e. for a small disturbance there is barely a difference the approaches and therefore the computational effort is not justified. But for large disturbances the one dimensional solver reveals an unacceptable deviation and it is necessary to apply the two dimensional approach if shock phenomena are expected. Since the computational time of the two dimensional solver is in the order of several minutes, whereas the CFD calculation presented above took 35 hours the two dimensional approach is to be preferred. Although for the shown case the deviation of the two dimensional solver is rather acceptable compared to the one dimensional approach, there is still improvement potential for future works which can be summarised in two main tasks. The given solver does not include friction and the flow is assumed to be planar which implies that the solver does not take dissipation into account and the curvature of the streamline caused by the sharp edge at the area where the upper and lower pipe hit each other is neglected. Both phenomena can be treated introducing a source term and will be subject to future works.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a$</td>
<td>Speed of sound</td>
<td>$\frac{\text{m}}{\text{s}}$</td>
</tr>
<tr>
<td>$c_v$</td>
<td>Isochoric heat capacity</td>
<td>$\frac{\text{J}}{\text{kg} \cdot \text{K}}$</td>
</tr>
<tr>
<td>$e$</td>
<td>Specific Energy</td>
<td>$\frac{\text{J}}{\text{m}^3}$</td>
</tr>
<tr>
<td>$f$</td>
<td>Frequency</td>
<td>$\frac{1}{\text{s}}$</td>
</tr>
<tr>
<td>$F$</td>
<td>Horizontal Flux</td>
<td>$[-]$</td>
</tr>
<tr>
<td>$G$</td>
<td>Vertical Flux</td>
<td>$[-]$</td>
</tr>
<tr>
<td>$I$</td>
<td>Momentum</td>
<td>$\frac{\text{kg} \cdot \text{m}}{\text{s}}$</td>
</tr>
<tr>
<td>$L$</td>
<td>Length</td>
<td>$\text{m}$</td>
</tr>
</tbody>
</table>
\( M \)  Mass \[ [\text{kg}] \]

\( p \)  Pressure \[ [\text{N/m}^2] \]

\( R \)  Gas constant \[ [\text{J/kg} \cdot \text{K}] \]

\( T \)  Temperature \[ [\text{K}] \]

\( t \)  Time \[ [\text{s}] \]

\( U \)  Conservative variables \[ [-] \]

\( u \)  Horizontal velocity \[ [\text{m/s}] \]

\( v \)  Vertical velocity \[ [\text{m/s}] \]

\( W \)  Width \[ [\text{m}] \]

\( x \)  Horizontal coordinate \[ [-] \]

\( y \)  Vertical coordinate \[ [-] \]

\( \kappa \)  Heat capacity ratio \[ [-] \]

\( \lambda \)  Eigenvalue in x-direction \[ [\text{m/s}] \]

\( \rho \)  Density \[ [\text{kg/m}^3] \]

\( \xi \)  Eigenvalue in y-direction \[ [\text{m/s}] \]

References


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Low compressibility of ionic liquids
and its effects on pulsation within hydraulic system

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The paper presents possible use of Ionic Liquids as a lubricant suitable for use as a hydraulic fluid. After a short presentation of ionic liquids and their interesting properties, the paper focuses on very low compressibility (resp. very high Bulk modulus) of ILs compared to the common hydraulic mineral oils and investigates the effects of their high bulk modulus on pressure pulsation and flow ripple of hydraulic pump.

Two most adequate ionic liquids for hydraulic application with highest bulk modulus where chosen and a special test rig was built using bent axis 7-piston pump powered by a servo motor. Results show change of resonance frequencies of entire hydraulic system due to higher bulk modulus and higher density of the ionic liquids. On the other hand, there is no significant change in pump pressure pulsation in non-resonance frequency range below 2500 rpm.

Keywords: hydraulic fluid, Ionic liquid, pump pulsation, flow ripple,

Target audience: Tribology and Fluids

1 Introduction to IL – Ionic Liquids

In recent two decades Ionic Liquids have gained in importance, causing a growing number of scientists and engineers to investigate possible applications of these liquids with unique physical and chemical properties. Their outstanding advantages such as high flash point, high thermal, mechanical and chemical stability, low gas solubility, very low compressibility, attractive tribological properties... make them very interesting for applications in mechanical engineering, offering a great potential for new innovative processes. Due to their physico-chemical properties they are also very interesting for the use within fluid power systems, as a promising hydraulic fluid for the future.

Ionic liquids are defined as molten salts with a melting temperature below 100 °C or lower – they are in the liquid state. A large number of them even being liquid at room temperature and then called “room temperature molten salts” with some characteristic properties e.g. melting point ≤ 100 °C down to approx. -60 °C; mild chemical reactivity, low corrosion; virtually no vapour pressure; organic cations; weakly coordinating ions [1].

That ILs are in essential salts (a combination of cations and anions), they typically consist of an organic cation and an inorganic or organic anion. Ionic liquids with melting point at ambient temperature consist of extensive and asymmetrical organic cations, such as 1-alkyl-3-methylimidazolium, 1-alkyl pyridine, 1-methyl-1-alkyl pyrrolidine or ammonium salts. The anions used range from simple halides, reducing the high temperatures of the melting point, to inorganic anions such as tetrafluoroborates and hexafluorophosphates and to extensive organic anions such as bis(trifluorosulphony)amides, triflates or tosylates – Figure 1.

When combined with certain specified cation anions you will obtain another salt – which, if it does not work by trial and error but intentionally, can be synthesised into a completely new material with entirely new properties.

However the cations and anions present in ionic liquids are so formulated that the resulting salts hardly crystallise. Therefore the ionic liquid is liquid within a wide temperature range. An important feature of ionic liquids is the possibility of adapting these physical-chemical properties through changing the natures of the anions and cations. The number of possible combinations is extremely high, that is why the best ionic liquid is supposed to be adapted for different usage: 1-10^6.

The possibilities for their usages within different areas of technology have been researched because of the numerous good properties of ionic liquids and their advantages over conventional liquids. Within different spheres of industry the ILs they are already used in practice. They are used e.g. during chemical synthesis and separation, cellulose production, crude oil processing, paint production, nuclear and solar energy, hydrogen storage, waste processing and battery production. Also their applicability’s regarding heat transfer and storage is at a stage of research and development. The scope of applications is constantly expanding. In addition the arena of functional fluids including lubricants and therefore hydraulic fluids as well are within a the phase of research and development. E.g. [2], [3], [4], [5].

2 Physico chemical Properties of ILs

Due to their excellent properties ILs catches the attention of a broad professional public. The most outstanding properties of ionic liquids are outlined below.

ILs have virtually no vapour pressure; pure and degassed ionic liquids show no cavitations. They also do not have boiling point and are non-flammable below the high thermal decomposition point, while their melting points can go down to -60 °C. Due to their excellent thermo-oxidative long term stability up to 300 °C and more they are not prone to ageing. ILs usually have small friction coefficients and good lubrication properties: quite often pure ionic liquids without any EP and AW additives can be found, which show friction coefficients better (and sometimes even much better) than a conventional fully synthetic high performance fluid under identical conditions.
Viscosities of ILs range from 15 mPas to several 10,000 mPas at 20 °C, whereas many of them are within the range between 35 to several 100 mPas. They also have good viscosity index and high tribological shear stability. Many ionic liquids show VI numbers between 150 and 270, in extreme cases even 500. In contrast to conventional fluids these high values are not a result of added VI-improvers, but an intrinsic property. Densities of ILs are typically between 0,85 and 1,50 g cm⁻³ at 20 °C, but frequently between 0,90 and 1,25 g cm⁻³.

ILs are typically fully water miscible and more or less hygroscopic, but with hydrophobic ionic liquids available as well. The uncontrolled uptake of water is critical and may have negative influences on viscosity, lubricity, corrosion and long-term stability.

Although ILs are salts, they are not automatically corrosive. Some ionic liquids are even used as corrosion protecting agents. Their common contact angles on metal surfaces typically range between 10° and 35°, meaning that ILs have moderate to good wettability of metal surfaces, while surface tensions typically range between 25 and 60 nN m⁻¹ at 20 °C.

Another very interesting property of ILs are their very low compressibilities that can go down to κ = 0,25 GPa⁻¹, which is even 50 % lower than the value of water. Their κ increases with temperature by 10 % every 100 °C and decreases with pressure by some % every MPa.

The specific heat capacities of ILs are usually between 1,5 and 2,6 kJ/kg·K at 20 °C, increasing in temperature by 10 to 20 % every 100 °C, whereas their thermal conductivities typically range between 0,1 to 0,2 W·m⁻¹·K⁻¹ and are almost constant with increasing temperature. The electrical conductivities of ILs range from 50 mS/cm to 1 µS/cm at 20 °C, although many of them are in the mS/cm range. Last but not least, ILs have bacteriostatic properties and no biofouling.

Despite excellent individual properties, it is very difficult to find a liquid that would combine the majority of good characteristics [6], [7].

## 3 Pump pulsation

The research objective was to evaluate pressure pulsation and flow ripple of hydraulic pump using two different ionic liquids with high bulk modulus in comparison to standard mineral hydraulic oil HLP. In this pilot stage of research – using ionic liquids as hydraulic fluids – we first have to evaluate the adequacy of pump design for use with ionic liquids that have higher density and bulk modulus which affect the operation of the hydraulic pump.

Hydraulic pumps, as all positive displacement pumps, have typical pulsation. The hydraulic pump cannot provide absolutely constant flow at constant speed. These small pulsations in the flow speed are called flow ripple. The flow ripple and consequently also, pressure pulsation, generated by an axial piston pump are relatively large compared to the rest of positive-displacement pumps [8].

In case of axial piston pump (Figure 2) the piston movement is sinusoidal where the flow rate by each piston is product of piston area and speed. The pump delivery is the summation of the flow rate delivered by all of the pistons in connection with the delivery port [10].

However, the pressure pulsation is not only related to the pump but also to the system which the pump is feeding [8]. Since this can make simulations and calculations complicated, we have tried to build as simple test rig as possible.

### 4 Test setup

#### 4.1 Hydraulic fluids

In order to evaluate effects of high bulk modulus and higher density of ionic liquids on pressure pulsation and flow ripple of hydraulic pumps, three different hydraulic fluids were studied and compared. Two ionic liquids with most adequate properties to be used as a hydraulic fluid with high bulk modulus were chosen (IL1 and IL2) and tested against standard mineral hydraulic oil HLP ISO VG 32. The most important properties of tested fluids are presented in Table 1, while detailed information on bulk modulus of fluids is given in Figure 3.

<table>
<thead>
<tr>
<th>Property</th>
<th>HLP VG 32</th>
<th>IL1</th>
<th>IL2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density @ 15°C [g/cm³]</td>
<td>0,869</td>
<td>1,266</td>
<td>1,241</td>
</tr>
<tr>
<td>Viscosity @ 40°C [mm²/s]</td>
<td>31,56</td>
<td>15,42</td>
<td>39,44</td>
</tr>
<tr>
<td>Viscosity @ 100°C [mm²/s]</td>
<td>5,41</td>
<td>4,313</td>
<td>7,66</td>
</tr>
<tr>
<td>Viscosity index</td>
<td>106</td>
<td>208</td>
<td>168</td>
</tr>
<tr>
<td>Bulk modulus [bar · 10⁹] @ 0 bar</td>
<td>1,64</td>
<td>2,85</td>
<td>3,09</td>
</tr>
<tr>
<td>Bulk modulus [bar · 10⁹] @ 100 bar</td>
<td>1,74</td>
<td>2,91</td>
<td>3,20</td>
</tr>
<tr>
<td>Bulk modulus [bar · 10⁹] @ 200 bar</td>
<td>1,86</td>
<td>2,97</td>
<td>3,32</td>
</tr>
<tr>
<td>Bulk modulus [bar · 10⁹] @ 400 bar</td>
<td>2,15</td>
<td>3,08</td>
<td>3,60</td>
</tr>
</tbody>
</table>

**Table 1: Properties of tested hydraulic fluids.**

All liquids were tested at two different viscosities, approx. 15,6 and 31,6 cSt, which were achieved by performing the test at different temperatures to match the viscosities of the liquids. These two viscosities were chosen because...
they represent lower and upper temperature limits of usual hydraulic oil applications (40 and 60 °C). Table 2 presents viscosities and temperatures of the fluids at which the tests were performed.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Temperature [°C]</th>
<th>Viscosity [cSt]</th>
</tr>
</thead>
<tbody>
<tr>
<td>HLP VG 32</td>
<td>59,0</td>
<td>15,6</td>
</tr>
<tr>
<td>IL1</td>
<td>39,5</td>
<td>15,6</td>
</tr>
<tr>
<td>IL2</td>
<td>69,0</td>
<td>15,6</td>
</tr>
<tr>
<td>HLP VG 32</td>
<td>40,0</td>
<td>31,6</td>
</tr>
<tr>
<td>IL1</td>
<td>18,3</td>
<td>31,6</td>
</tr>
<tr>
<td>IL2</td>
<td>46,0</td>
<td>31,7</td>
</tr>
</tbody>
</table>

Table 2: Viscosities and temperatures of fluids at which the tests were performed.

4.2 Hydraulic test rig

The test rig was designed and constructed as simple as possible to allow comparison between the test and the simulation model which will be developed in the future. Each additional component in the system would only introduce more and more unknown variables into the simulation model.

The pressure pulsation and flow ripple was studied on a Rexroth A2F5/60W-C3 pump, which is a 4.93 cm³ bent axis 7-piston pump that can also be used as a motor. The pump was driven by a servomotor with closed-loop speed control at different rotational speeds.

The test rig setup is presented in Figure 4. The pump was placed into a simple hydraulic circuit with 4 pressure sensors and a variable orifice at the end. A pressure relief valve, set to 300 bar, was placed at the end of the line for safety reasons. The lengths of pipelines (with outside diameter of 12 mm and wall thickness of 2 mm) between each component are also presented in Figure 3. A 20 L hydraulic tank was used, filled with approx. 15 L of tested liquid and equipped with temperature probe to measure the temperature of the liquid near the suction line.

4.3 Data acquisition setup

Four precision pressure sensors GE PDCR 4060 were installed into the pipeline, as presented in Figure 4, and were coupled to NI 9237 data acquisition unit (±25 mV/V, Bridge Analog Input, 50 kS/s/ch, 4 Ch Module) driven by National Instruments cRIO-9024 Real-Time FPGA Controller. The measurement system allowed us to take pressure readings at 4 different locations in the pipeline at 50 kHz sample rate. To compensate the temperature drift of the pressure sensors, the sensors were calibrated after each measurement series at given temperature.

5 Results

Measurements of pump pulsation were made at two different pressures of 100 and 200 bar, two different viscosities of 15.6 and 31.6 cSt and several different rotational speeds ranging from 1000 to 4000 rpms in 100 rpm steps.

There were some additional limitations while performing measurements:

- if resonance of the system was to excessive, the measurements were not taken in high-resonance rotational speeds,
- measurements with IL1 at high viscosity of 31.6 cSt at $T=18,3\, ^{\circ}\mathrm{C}$ were only taken at 500 rpm steps due to ineffective cooling.

Figure 5 presents hydraulic pump pressure pulsation using different tested fluids at $n=1500$ rpm.

As described above there were approx. 360 measurements made (3 fluids at 2 viscosities and 2 pressures in 30 steps by 100 rpm steps; multiplied), thus it is impossible to present all measurements in this paper.

Figure 5 presents hydraulic pump pressure pulsation (measuring point p1, just after the pump) at nominal speed of $n=1500$ rpm, at 200 bar and viscosity of 15.6 cSt using different tested fluids. Pressure pulsation is shown only for 1 revolution, although 6 revolutions were recorded for each measurement and used for all further calculations.

Further on, the pressure pulsation was evaluated in two ways:

- $p_{\text{max}} - p_{\text{min}}$: the absolute difference between maximal and minimal pressure reading
- RMS: RMS value of pulsation was calculated by subtracting the average pressure from measured pressure signal.

Figure 6 presents $p_{\text{max}} - p_{\text{min}}$ and RMS value of pressure pulsation from 1000 to 4000 rpm for each tested fluid at (low) viscosity that corresponds to HLP VG 32 viscosity at 60 °C and at pressure 200 bar. Note that the $p_{\text{max}} - p_{\text{min}}$ values and lines are divided by 5 in order to fit the same y-axis scale as RMS values.
On the other side, there is no significant change in pump pressure pulsation in non-resonance frequency range below 2500 rpm. Thus, data for non-resonance frequency was further investigated.

Table 3 shows the hydraulic pump pressure pulsation in non-resonance frequency range below 2500 rpm in comparison to standard HLP VG 32 oil:

- For IL1 was approx. 6 % higher at low viscosity (15.6 cSt).
- For IL2 was approx. 11 % higher at low viscosity (15.6 cSt).
- For IL1 was approx. 19 % higher at high viscosity (31.6 cSt).
- For IL2 was approx. 16 % higher at low viscosity (15.6 cSt).

6 Summary and Conclusion

The aim of the research was to evaluate the adequacy of hydraulic pump design for use with ionic liquids that have higher density and bulk modulus than common hydraulic fluids. A special test rig was built to investigate the pump flow ripple and pressure pulsation.

Based on the numerous measurements made we can conclude that both ionic liquids performed very well as a hydraulic fluid in a hydraulic system with no extra modifications. The results revealed that there is little to no increase in pump pulsation while using ionic liquids at non-resonance frequencies below 2500 rpm. However, since the pressure pulsation is not only related to the pump but also to the system which the pump is feeding, we can notice significant change of resonance frequency of the hydraulic system and pipeline.

Based on the results, research work on ionic liquids used as hydraulic fluids can be continued. The next planned step is to evaluate performance of ILs on long term endurance tests.

References

High Pressure Falling Cylinder Viscometer - Error Analysis and Improvement Proposal

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With pressure levels rising for applications such as compression-ignition engines and numerical design approaches are used to optimise fluid power components, rheological properties of the fluid in the according operation points gain interest. The measurement of viscosity under high-pressure has been subject to research for many years. However, to this day, it still bears uncertainty. This paper presents typical errors for high-pressure measurements and strategies to minimise uncertainty. With a focus on material combinations, geometric parameters and the measurement principle, the errors are explained, and an improvement proposal is given based on the findings.

**Keywords:** Viscosity, Viscometer, High-pressure, Rheology, Measurement

**Target audience:** Tribology & Fluids, Design Process

**1 Introduction**

Within the cluster of excellence “Tailor-Made Fuel from Biomass” at RWTH Aachen University, alternative fuel candidates are investigated. With the increase in the people’s mobility as well as global emissions, the need for new and sustainable fuel sources is rising. To meet the increasing demand, many technological pathways are possible. With hybrid drives, electrical drives, hydrogen fuel cells and CNG-engines being researched and refined, the possibilities of alternative propulsion are already on the market. A different approach is to use the existing combustion engine infrastructure with all its components and replace the fuel with sustainable alternatives. The advantages are a clean combustion and only minor adaptions of the combustion system to these new fuels, ensuring a minor entry-barrier for this technology.

One promising candidate for usage in compression-ignition (CI) engines is di-n-butylether (DnBE). Looking at engine performance, DnBE outperforms EN 590 (Diesel) in many aspects. Especially soot emissions can be reduced drastically compared to EN 590. With injection pressures steadily rising to almost 300 MPa, fuel pumps need to be adapted to higher pressures and different fluids with hydrodynamic properties of the fluid playing a crucial role for pump performance. Figure 1 shows the dynamic viscosities of EN 590 and DnBE over pressure and temperature /1/. With a viscosity ratio of approximately six between EN 590 and DnBE, the injection pump performance using DnBE is significantly worse than for EN 590. Volumetric efficiencies reach values as low as 40 % instead of 90 % for EN 590 /2/.

The knowledge of hydrodynamic properties enables the analysis of potential improvements under consideration of the fluid. In case of the injection pump, measures can be taken to reduce solid friction by influencing the piston geometry in a way that the piston runs more concentrically. In case of volumetric losses, hollow pistons increase the volumetric efficiency up to 40% in comparison to the standard pump /3/.

In hydraulics, even simple systems are heavily influenced by rheological properties. When using a standard hydraulic oil such as HLP46, a 10 K decrease in fluid temperature can change the dynamic viscosity by up to 40%, see Figure 2 left. In a generic hydraulic resistance like a throttle, depicted in Figure 2 right, such a change in viscosity results in a differing flow rate.

**Keywords:** Viscosity, Viscometer, High-pressure, Rheology, Measurement

**Target audience:** Tribology & Fluids, Design Process

**1.1 Falling cylinder viscometers**

Falling cylinder viscometers base on the principle of stokes flow. A cylinder is falling inside a fluid filled tube at low speed enabling quasistationary stokes flow with Reynolds numbers below one. The terminal velocity or the time it takes between two distinct positions is measured and can be related to the fluid viscosity if the densities of fluid and cylinder as well as the geometrical relations between tube and falling cylinder are known. Figure 3 shows the relevant parts of a falling body viscometer cell according to /5/. From the upper contact the cylinder starts falling towards the lower contact. Once it reaches the bottom position, the time is measured. In order to bring the cylinder back into initial position, a coil is positioned around the tube. By setting the voltage to a certain value, the ferromagnetic cylinder is moved up through induced magnetic forces on the cylinder. Once the cylinder reaches top position, the voltage is shut off, causing the cylinder to plummet again. One advantage of
this principle is that the measurement cell can be integrated into high pressure chambers. Only small amounts of wiring are necessary. The measurement itself works without visual sensors.

After each falling process, the viscosity can be calculated analytically with the densities of the fluid $\rho_F$ and the falling cylinder $\rho_{FC}$ as well as the falling time $t_{fall}$, falling distance $x_{fall}$ and a geometry constant $C$, see Equation (1), which will be explained in more detail later. Especially the so-called calibration factor $C$ and the density of the falling body can play a major role for the improvement of this measurement principle.

$$\eta = C \cdot (\rho_{FC} - \rho_F) \cdot \frac{x_{fall}}{t_{fall}} = C \cdot (\rho_{FC} - \rho_F) \cdot \frac{1}{t_{fall}} \quad \text{(1)}$$

Essential for robust viscosity measurements are reproducible falling times. Figure 4 shows the falling times over pressure for vegetable oil /6/. With increasing pressure, the falling times of the falling cylinder increase, due to higher viscosities. Ideally, these falling times move along a steady curve increasing with rising pressure and decreasing with rising temperature. Figure 4 suggests that this is hardly achieved. With deviations of 16% and more in some cases, the resulting viscosities are highly influenced by these discrepancies. Reasons for varying falling times can be found in the positioning of the falling cylinder relative to the tube. Different to the assumption of Equation (1) defining an analytical correlation of viscosity to the falling time for a cylinder positioned concentrically inside the tube, the cylinder position might be eccentric. Therefore, the resulting gap is not always uniform, and the flow cannot be calculated as easily.

![Figure 3: Principle design of a falling cylinder viscometer with essential components and flow profile in the gap between falling cylinder and tube](image)

Figure 3: Principle design of a falling cylinder viscometer with essential components and flow profile in the gap between falling cylinder and tube.

For an eccentric positioning, the resulting gap height varies over the circumference of the cylinder. Using the analytical relationship for a flow through a circumferential gap with and without eccentricity, see Equation (2), the difference in falling time can be calculated using a normalised eccentricity with a maximum value of 1 /7/, see Equation (3). Figure 5 shows the falling time for normalised eccentricities of the falling cylinder inside the tube. Friction effects were neglected for this theoretical study. For increasing eccentricity values, falling times are reduced significantly and can reach theoretical values of only 40% of concentric falling times, resulting in deviating viscosities. Experiments validate these findings /8/. Using a mean for falling time scatter at constant conditions therefore does not ensure the correct time since higher values indicate a more concentric position.

$$Q = \frac{D n \Delta \rho^2}{12 \pi l} \left[ 1 + 1.5 \left( \varepsilon \frac{x}{r} \right)^2 \right] \Delta p \quad \text{(2)}$$

$$t_{\text{eccentric}} = \frac{1}{1 + 1.5 \left( \varepsilon \frac{x}{r} \right)^2} \quad \text{(3)}$$

![Figure 5: Falling times dependent on the eccentricity of the cylinder](image)

Figure 5: Falling times dependent on the eccentricity of the cylinder.

To ensure concentric alignment of falling body and tube, design changes can be made to the falling body. Here, a spherical tip is beneficial as well as a hollow end, see /7/. These changes have proven to improve the operation significantly. Nonetheless, Figure 4 shows that even with these types of sinkers, continuous falling times can hardly be achieved. Observations do show that eccentricities and tilting is still possible with adapted sinker designs. Ultimately, improvements still have to be made for an optimal design.

Ideally, the terminal velocity of the falling cylinder should be used for determination of the dynamic viscosity, see Equation (1). This would enable the detection of changes in falling behaviour due to eccentricities, tilting and other flow effects. Using a set of coils, real-time position detection based on the LVDT-principle can be realised. With a time-stamp on each position signal, a direct velocity determination is possible, enabling the evaluation of dynamic viscosities using short falling distances with higher accuracy. In order to exploit this enhanced accuracy, geometrical dependencies between tube and sinker need to be known. In the following, the highly geometry dependent calibration factor $C$ is introduced.

### 1.2 Calibration factor $C$

Next to densities of the fluid and the falling cylinder, geometrical parameters of the tube and cylinder influence the measurement via the so-called calibration factor $C$. It can be derived by solving the navier-stokes equation for a cylindrical flow /9/. The analytical correlation of $C$ with the radii of the falling cylinder (index $FC$) and the tube (index $T$) is shown in Equation (4).

$$C = \frac{1}{2} \pi r_T^2 C - \left( \ln \frac{r_{FC}}{r_T} \right) \frac{r_T - r_{FC}}{r_T} \quad \text{(4)}$$

In general, this constant is determined at atmospheric conditions. For a small interval of pressure and temperature, $C$ can be viewed as constant. At high pressure or temperature conditions however, deformations significantly influence $C$. Deformations can be reduced by designing the viscometer in a way of limiting differential pressures of components. This has been done for the measurement cell at IFAS by placing it inside a
high-pressure chamber. Therefore, pressure forces act on tube and cylinder both from the in- and outside of the tube.

2 Application oriented material selection

Increasing pressure levels of diesel injection systems up to 300 MPa required an adaption of the viscometer towards higher pressures. The test-rig was upgraded for pressures up to 750 MPa and temperatures between 253.15 K and 353.15 K. These conditions can cause significant deformations of cylinder and tube. Therefore, suitable material selections based on the operating conditions of the test-rig are necessary. Due to the usage of the LVDT-principle, the falling body has to be made of ferromagnetic material whereas the tube has to be non-magnetic. In the following, three different material-pairings are investigated, see Table 1.

<table>
<thead>
<tr>
<th>Pairing No.</th>
<th>Falling cylinder</th>
<th>Tube</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>S355-steel</td>
<td>CuZn37 - brass</td>
</tr>
<tr>
<td>2</td>
<td>CuZn37 – brass (ferromagnetic)</td>
<td>CuZn37 - brass</td>
</tr>
<tr>
<td>3</td>
<td>S355-steel</td>
<td>X5CrNi18-10 – 1.4301</td>
</tr>
</tbody>
</table>

Table 1: Investigated material pairings for the falling body viscometer.

In general, materials with good thermal and mechanical stress-resistance are suited for usage within the pressure cell. Limiting factor is the ability of the materials for mechanical treatment and achievable tolerances. The pairings chosen display good properties for all aforementioned aspects or have been used in other applications, see /10/.

With FEM analysis pressure- and temperature-dependent deformations can be calculated. Figure 6 shows the resulting deformations of tube and cylinder as well as the effective gap change over pressure and temperature for the first material pairing. Negative values indicate a shrinking while positive values signal expansion. For the first material pairing, the tube changes its geometrical shape with higher gradients than the cylinder resulting in an overall change in gap-height with values of up to -5.793 µm for 253.15 K and 800 MPa and 5.479 µm for 353.15 K and 0 MPa.

The resulting gap height for pairings no. 2 and 3 are shown in Figure 7. Both graphs indicate a significant improvement compared to the S355-brass-pairing. Both pairings show less change over temperature and pressure. Overall results with the maximum deformations can be found in Table 2. Here, the error displays the relative change in gap height compared to ambient conditions. Pairing 1 shows deviations in temperature (T) and pressure (p) domain of up to 15.5 %. For pairing 2 and 3 those values are around 10 % with pairing 2 performing better over temperature and pairing 3 showing advantages during pressure dominant operation.

Using the information of deformation over pressure and temperature, the C-factor can be determined for every pairing in every possible measurement condition, see Figure 8. Thus, the pressure and temperature dependency can be visualised. A lower gradient indicates a more constant value over the domain. For a constant C-factor over temperature and pressure, a pairing has to be found, that has optimal thermal and mechanical behaviour. If the S355-stell falling body is selected, the tube has to have a bulk modulus of around 70 % of S355-steel to provide constant values over pressure. As for temperature, the thermal expansion coefficient needs to be around 50 % of the value of S355-steel. In addition, the potential material needs to be non-magnetic in order to not influence the position measurement of the falling body. A material pairing enabling a temperature and pressure independent calibration factor has not yet been found.
3 Improvement towards more robust measurements

According to Equation (1) viscosity measurements based on the falling body principle always need information of the fluids density at the investigated conditions. To minimize measurement errors, the density must be determined simultaneously with the viscosity measurement in one pressure cell. This is not possible, when the cell requires disassembling for integrating different measurement devices. As a solution a novel approach was proposed /11/. Here, two different falling bodies are used within the same tube, causing two different falling speeds. If both bodies do fall within the stokes flow regime, the viscosity and the density can be determined simultaneously. With the bodies 1 and 2 falling with two different speeds, two equations with two unknowns result. Therefore, density can be substituted by the geometry and velocity of the other falling body, see Equation (5) to (7).

\[ \eta = C_1 \cdot (\rho_{C2} - \rho) \cdot \frac{1}{u_1} = C_2 \cdot (\rho_{C2} - \rho) \cdot \frac{1}{u_2} \]  
(5)

\[ \rho_T = \frac{C_2 \rho_{C2} u_2 - C_2 \rho_{C2} u_1}{C_1 u_2 - C_2 u_1} \]  
(6)

\[ \eta = \frac{C_1 \rho_{C2} u_2 - C_2 \rho_{C2} u_1}{C_1 u_2 - C_2 u_1} \]  
(7)

With current viscometer designs, it is only possible to use one falling body inside a tube. Therefore, a disassembly of the pressure cell is necessary in order to exchange falling bodies. This results in different conditions as well. To reduce the measurement error, a new viscometer design is proposed. Figure 9 shows this design. A cascade of coils that can be actuated individually, initially keeps both bodies at the top of the tube. With individual control of each coil, it is possible to cause body 2 to fall while body 1 is still at the top position. Once body 2 arrives at the bottom of the tube, the falling process for body 1 is initiated. This way, both bodies are not influencing each other during the fall. For this principle to work, the tube length has to be sufficient for two falling bodies within the measurement device. If the aforementioned criteria are met, an in-situ measurement of density and viscosity is possible, reducing measurement error and enhancing reliability of the data.

4 Summary and Conclusion

In this paper, an error analysis of falling body viscometers was carried out considering material pairings and resulting pressure- and temperature-dependent deformations as well as the measurement principle. Falling body viscometers enable viscosity measurement at high pressures and temperatures since they do not rely on optical sensors. Based on the LVDT-principle, determination of the terminal velocity is possible, enhancing the measurement accuracy. To further increase measurement precision, material pairings of tube and body can be found that cause a favourable performance regarding the deformation under pressure and temperature. In addition, regarding errors caused by material deformations, a sequential falling process with two geometrically similar falling bodies of different was proposed, enabling in-situ viscosity and density measurements.

With the knowledge of material behaviour, measurement devices can be designed with regard to planned operation conditions. Combining the knowledge of material pairings with an improved viscometer design, the measurement accuracy can be increased significantly, providing more reliable rheological data for simulations and the design of fluid power applications.

To realise this potential, a new measurement device has to be developed. Additionally, suited material pairings have to be investigated in regards of achievable manufacturing tolerances.

5 Acknowledgements

This work was performed as part of the Cluster of Excellence "Tailor-Made Fuels from Biomass", which is funded by the Excellence Initiative by the German federal and state governments to promote science and research at German universities.

Nomenclature

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>CI</td>
<td>Compression-ignition</td>
</tr>
<tr>
<td>CNG</td>
<td>Compressed natural gas</td>
</tr>
<tr>
<td>DnBE</td>
<td>Di-n-Butylether</td>
</tr>
<tr>
<td>EN 590</td>
<td>Diesel fuel</td>
</tr>
<tr>
<td>HLP</td>
<td>Mineral based hydraulic oil with corrosion and high-pressure additives</td>
</tr>
<tr>
<td>IFAS</td>
<td>Institute for fluid power drives and controls</td>
</tr>
<tr>
<td>LVDT</td>
<td>Linear variable differential transformer</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>Calibration factor</td>
<td>(m^3/s^2)</td>
</tr>
<tr>
<td>D</td>
<td>Diameter</td>
<td>m</td>
</tr>
<tr>
<td>e</td>
<td>Eccentricity</td>
<td>m</td>
</tr>
<tr>
<td>g</td>
<td>Gravitational acceleration</td>
<td>m/s²</td>
</tr>
<tr>
<td>h</td>
<td>Height</td>
<td>m</td>
</tr>
<tr>
<td>l</td>
<td>Length</td>
<td>m</td>
</tr>
<tr>
<td>p</td>
<td>Pressure</td>
<td>Pa</td>
</tr>
</tbody>
</table>
Flow rate \[ \text{m}^3/\text{s} \]
Radius \[ \text{m} \]
Temperature \[ \text{K} \]
Time \[ \text{s} \]
Velocity \[ \text{m/s} \]
x - x-direction \[ \text{m} \]
z - z-direction \[ \text{m} \]
Dynamic viscosity \[ \text{Pas} \]
Density \[ \text{kg/m}^3 \]

Index Description
1 Body/position no. 1
2 Body/position no. 2
Concentric Concentric alignment
Eccentric Eccentric alignment
\( F \) Fluid
\( FC \) Falling cylinder
\( \text{fall} \) During the falling process
\( T \) Tube

References

Accumulators with sorbent material – an innovative approach towards size and weight reduction

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Utilizing accumulators in hydraulic systems with the purpose of energy storage, temporal changes in state of the storage medium must be considered during design and prospectively also monitored during operation. High efficiency aside, the reduction of weight and size is of high interest, especially in mobile applications. Regarding these objectives, accumulators with sorbent material are an innovative and promising development. The herein introduced generic physical model enables the consideration of sorption processes in the description of such accumulators. The results are discussed by means of time response analysis and compared to the behaviour of conventional accumulators. Potential use cases are investigated and the model application to a practical duty cycle is shown.

Keywords: accumulator, size reduction, sorbent material
Target audience: mobile hydraulics, design process, component manufacturer

1 Introduction

Accumulators are utilized in hydraulic systems for various functionalities. They serve as energy storage, supply of demanded volume flow, compensation of leakage or as capacitive element of a Helmholtz-Resonator for absorption of pressure pulsation. In this paper we focus on the accumulator as energy storage. The stored energy serves as coverage for the peak load of a rotating or linear hydro motor (boosting). When reversing the power flow the motor serves as a pump, so braking energy can be recuperated. Boosting and recuperation enables downsizing of the drive unit and, as a result, the reduction of energy consumption.

Aside from energy consumption, the reduction of weight and size is a primary objective in mobile applications. Given this background, recent development of accumulators involves employing sorbent material, utilized e.g. as vehicle air springs /1/ and gas pressure tank /2/. The physical phenomenon of adsorption and desorption is exploited for an only ostensible volume increase. An agglomeration of gas molecules takes place on the inner surface of a highly porous solid material, called sorbent, due to Van-der-Waals-forces. The transition from gas to adsorbed phase is an exothermic process, which means energy is released as adsorption heat. Desorption denotes the inverse process to adsorption. These sorption processes are described by means of sorption isotherms, the functional correlation between capacity of the sorbent and system state (pressure, temperature) /3/ /4/.

The theoretical part of this paper is close to the recent work of the third author published in german /5/, which presents an axiomatic model for hydraulic accumulators with sorbent material. The conservation equations for mass and energy are extended by appropriate terms representing the sorption processes and discussed in detail. This is followed by an in-depth analysis of the dynamic storage behaviour. New is the concluding part of this article in section 5, which discusses potential applications of accumulators with sorbent material and shows the model application on a practical duty cycle. These studies are carried out in context of the CRC 805 ‘Control of uncertainty in load carrying structures in mechanical engineering’ (SFB 805) at the Technische Universität Darmstadt. The research project is funded by the German Research Foundation (DFG).

2 Conventional hydraulic accumulators

The following section describes the state transitions of a conventional hydraulic accumulator (c.f. Figure 1 a). During expansion the gas pressure drops and at the same time, if the expansion happens fast enough, the gas temperature. This process is reversed during compression. Transitions are called fast if they occur in time frames much shorter than the thermal relaxation time of the system. The thermal relaxation time $\tau$ is the ratio of gas heat capacity at constant volume $c_p/Q_C$ with respect to the thermal conductivity $k$: $\tau = c_p/Q_C/kA$. Assuming that the thermal resistance is dominated by the heat transfer on the inner surface of the accumulator, the heat transmission coefficient is $k = \text{Nu} \cdot \lambda$, where $\text{Nu} = 3$ is the Nusselt number, /6/, /7/. Here, $\lambda$ is the coefficient of thermal conductivity and $S = A/V$ the volume specific surface of the accumulator. With the results of Pelz et al. /6/, /7/ the thermal relaxation time can be obtained by the equation

$$\tau = \frac{c_p c_v}{k A} = \frac{\alpha}{\text{Nu} \cdot \lambda S^2} = \frac{1}{\text{Nu} \cdot \lambda S^2} \cdot \frac{\alpha}{\gamma \cdot c_p / c_v}$$

(1)

where $\alpha = 1/(c_p \cdot \rho_0)$ is the thermal diffusivity of the gas and $\gamma = c_p / c_v$ is the isentropic exponent obtained by the ratio of the specific heat capacities. The isentropic exponent has a value of $\gamma = 7/5 = 1.4$ assuming the gas is diatomic.

Figure 1: Hydraulic accumulators based on two different physical mechanisms: a) mere gas expansion, b) parallel setup of porous sorbent as accumulator in addition to gas expansion.

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**Figure 1**: Hydraulic accumulators based on two different physical mechanisms: a) mere gas expansion, b) parallel setup of porous sorbent as accumulator in addition to gas expansion.
The earliest studies concerned with thermal equilibrium processes in hydraulic accumulators were conducted by Otis /8/, who introduced the thermal relaxation time as a significant quantity. A spherical accumulator with a diameter \( D \) has the specific surface \( S = 6/D \), yielding the thermal relaxation time \( \tau = D^2/(151.2 \ a) \). Assuming \( D = 0.2 \ m \) and \( a = 34 \times 10^{-4} \ m^2/s \) (applicable for air at 7 bar and 20 °C) results in the thermal relaxation time \( \tau = 78 \ s \). This result is a typical value, confirmed by measurements of air springs /6/. With increasing precharge pressure \( p \) and change of the volume specific surface, oil hydraulics applications typically achieve values up to \( \tau = 100 \ s \). With \( D = 1 \ mm \) the relaxation time decreases by four orders of magnitude down to \( \tau = 0.3 \ s \). The changes in state inside small oscillating gas bubbles in liquids are for example usually isothermal /7/, /9/, /10/. To summarize: All processes taking place in timeframes significantly shorter than \( \tau \) are adiabatic and thus isentropic. All changes in state occurring in significantly longer timeframes than \( \Delta t \gg \tau \) are isothermal.

In thermodynamics, processes are usually categorized into isentropic, isothermal, isochoric and isobaric processes. As is apparent in the discussion above, this classification is associated with the consideration of characteristic timescales. This distinction into isentropic and isobaric processes is not possible for changes in state occurring in timeframes of \( \Delta t \gg \tau \). Thus, the goal is to predict the temporal behaviour of the gas state for timescales of \( \Delta t = \tau \). This is achieved by means of evolution equations based on axioms, namely the continuity equation and the energy equation.

3 Dynamic model of an hydraulic accumulator with sorbent material

The following assumptions are made for the modelling:

(i) zero-dimensional model:

The thermodynamic states are only considered to be functions of the time \( t \), but not of the location. Especially the gas pressure \( p(t) \) as well as the gas temperature \( T(t) \) inside the cylinder volume are considered spatially averaged. The spatial resolution of impulse- and temperature boundary layer is possible /7/, /9/, but is not considered in this paper.

(ii) equations of state

The gas is assumed to be caloric and thermally ideal. Therefore the caloric equations of state for the internal energy \( e = c_v T + \text{const} \) and enthalpy \( h = c_v T + \text{const} \) are applicable, as well as thermal equation of state \( p = \rho R T \), with the density \( \rho \) and the specific constant \( R = c_p - c_v \). The adaptation for real gas is possible by means of applicable equations of state.

(iii) sorption

The sorbent is geometrically characterized by the porosity \( \varepsilon = \lim_{\Delta V_{\text{surf}} \to 0} \Delta V_{\text{surf}}/\Delta V_{\text{surf}} \) and the specific surface \( s = \lim_{\Delta V_{\text{surf}} \to 0} \Delta A_{\text{surf}}/((1 - \varepsilon)\Delta V_{\text{surf}}) \), as figure 1 b) illustrates. Assuming spherical bulk material or nonwoven material with an average sphere- or thread-diameter this results in \( s = s_0 \) for dimensional reasons. During adsorption of gas molecules bond energy is released. The needed bond energy for desorption is conducted to the surface from the environment. At the equilibrium, while assuming the sorption isotherm to be linear, the surface specific mass of agglomerated molecules \( q \) is proportional to the \( \rho \), with the pressure specific sorbent load \( q^* := \rho q / p ) \). The model design could also account for separated gas components by means of partial pressures. Likewise it is possible to replace the linear sorption isotherm with a non-linear isotherm (eg. Langmuir-isotherm), /4/, /11/, /12/.

The continuity-, energy- and state equations in conjunction with the above assumptions (i) through (iii) yield the evolution model for the thermodynamic state of the gas inside the accumulator, and thus the work capacity. The model is an extension of the simulation model for air springs and dampers ADASS (Air Damping Air Spring Simulation), developed by the third author. This simulation model is employed by the companies Vibraacoustic and Daimler to aid the design of pneumatic suspension systems /13/.

The following section shows continuity equation, energy equation and equation of state for the discussed processes. The continuity equation yields the evolution equation

\[
\frac{d \rho}{dt} + \rho q + \dot{m}_{\text{sorb}} = -A_{\text{surf}} \frac{d q}{dt}(p, T)
\]

(2a)

where \( \dot{m}_{\text{sorb}} \) considers a mass flow due to permeation through membrane or seal. The impact of an associated loss of mass is only noticeable in significantly longer timeframes compared to the relaxation time \( \tau \). The right-hand term of the continuity equation describes the behaviour of the sorbent. The structure of the equation implies the analogy with a capacity: The flow (here the mass flow) is obtained by multiplication of capacity and time derivative of the potential (here the surface specific mass of agglomerated molecules). In this context the sorbent takes the role of an additive capacity.

The energy equation includes a term on the right-hand side, describing the heat source that results from the sorption processes, in addition to the heat flow. During adsorption the bond energy \( E_b \) is released and conducted to the gas, during desorption the bond energy is drawn from the gas of molar mass \( M \):

\[
V \frac{d q}{dt} T + \frac{\dot{m}_{\text{sorb}}}{M} E_b \frac{d p}{dt} (p, T)
\]

(2b)

Usually, the sorbent’s change of inner energy is negligible due to the low mass of the sorbent. To complete the system, the thermal equation of state (alternatively a compressibility factor can be employed)

\[
p = \rho R T
\]

(2c)

is needed, as well as the sorption equilibrium

\[
q = q(p, T)
\]

(2d)

The system can be solved for a known initial state

\[
p(0) = p_0, T(0) = T_0 = T_u
\]

(2e)

given a diffusion model for the permeation flow is established. The unknown quantities are gas pressure \( p \), gas temperature \( T \), gas density \( \rho \) and gas volume \( V \). The equations (2a) through (2d) are sufficient to solve this system. For example given a kinematic stimulation of the volume flow \( Q(t) \), the first term of the continuity equation and the energy equation with the time integral \( V(t) = V_0 + \int_0^t Q(t) \) dt, yield the accumulator volume \( V \).

4 Analysis of the dynamic behaviour of a hydraulic accumulator with sorbent material

This section shows a more detailed consideration of the developed model in the specific case of negligible permeation \( \dot{m}_{\text{sorb}} = 0 \) and a linear sorption isotherm \( q = q^* p \):

\[
V \frac{d q}{dt} + q q^* = -A_{\text{surf}} \frac{d q^*}{dt}\]

(3a)

\[
p V + \frac{Q}{T} p \rho + (y - 1) k A(T - T_u) = (y - 1) A_{\text{surf}} E_b \frac{d q^*}{dt}\]

(3b)

\[
p = \rho R T
\]

(3c)

\[
p(0) = p_0, T(0) = T_0 = T_u
\]

(3d)

The dimensional analysis /14/ for a harmonic stimulation with the cycle time \( 2\pi / \Omega \) motivates the introduction of the dimensionless quantities

\[
p_v := p/p_0, T_v := T/T_0, q_v := q/q_0, V_v := V/V_0, Q_v := Q/(V_0 \Omega), \Omega_v := \Omega, \Omega_v := \Omega
\]

(4)
As well as the dimensionless form of the equation system (3):

\[ \begin{align*}
V_\epsilon \dot{q}_\epsilon + \dot{q}_\epsilon & + q_\epsilon \dot{p}_\epsilon + q_\epsilon \dot{\rho}_\epsilon = 0, \\
p_i V_i + \left( \frac{T_i}{\rho_i} \right) - \frac{1}{\rho_i} & = E_{d, i} q_i^\prime \dot{p}_i, \\
p_x = q_x T_x, \\
p_x(0) = 1, T_x(0) = 1.
\end{align*} \tag{5a,b,c,d} \]

The four following dimensionless parameters resulting from the dimensional analysis characterize the accumulator explicitly:

\[ \begin{align*}
\Omega_x & = \frac{\Omega_x}{\tau}, \quad q_x^\prime = \frac{m_{\text{adsorption}}}{m_{\text{ref}}}, \\
q_x & = \frac{-k T_x^\prime}{\mu V_x}, \quad E_{d, x} = \frac{E_d}{M R T_0^2} (y - 1) = \frac{E_d}{M c v_s}, \quad \gamma.
\end{align*} \tag{6} \]

The dimensionless frequency \( \Omega_x \) is the product of relaxation time \( \tau \) and stimulation frequency \( \Omega \) and quantifies the temporal behaviour (\( \Omega_x \to 0 \) implies isentropic behaviour, \( \Omega_x \to \infty \) an isothermal one). The dimensionless sorbent charge \( q_x^\prime \) describes the sorbent material and represents the ratio of adsorbed gas mass \( m_{\text{adsorption}} \) and free gas mass \( m_{\text{ref}} \) at initial state. Thus, it quantifies the sorption capability of the accumulator and can be easily obtained by measurement. The dimensionless bond energy \( E_{d, x} \) represents the ratio of bond energy and inner energy at initial state and therefore quantifies the inner heat source of the accumulator. \( \gamma \) denotes the isentropic exponent of the gas.

The dynamic behaviour of the accumulator can be derived from the model (5) using ‘pen and paper’, by linearizing the non-linear initial value problem by means of perturbation theory and transforming it into the frequency domain, with the ansatz \( p_x = 1 + \beta_x = \Re(1 + \rho_x \exp \left( i \Omega_x \right)), \quad \dot{q}_x = 1 + \dot{\rho}_x = \Re(1 + \dot{\rho}_x \exp \left( i \Omega_x \right)), \quad \dot{T}_x = 1 + \dot{T}_x = \Re(1 + \dot{T}_x \exp \left( i \Omega_x \right)), \quad V_x = 1 + \dot{V}_x = \Re(1 + \dot{V}_x \exp \left( i \Omega_x \right)), \quad \text{where } i = \sqrt{-1}. \) Origination from the initial value problem (5), results the algebraic formulation for the system’s steady state

\[ \begin{align*}
\beta_x + \dot{\beta}_x + q_x^\prime \dot{\rho}_x & = 0, \\
\beta_x + \gamma \dot{\beta}_x + \frac{1}{\Omega_x} \dot{T}_x & = E_{d, x} q_x^\prime \dot{\rho}_x, \\
\beta_x & = \dot{\beta}_x + \dot{T}_x.
\end{align*} \tag{7a,b,c} \]

The model described by the equations (7) through (7c) can be represented in form of a linear system of equations

\[ \begin{pmatrix}
1 - E_{d, x} q_x^\prime & 0 & 1 \\
1 & 1/\Omega_x & 0 \\
1/\Omega_x & 0 & -1
\end{pmatrix} \begin{pmatrix}
\dot{\beta}_x \\
\dot{T}_x \\
\dot{\rho}_x
\end{pmatrix} = \begin{pmatrix}
-\gamma \\
0 \\
0
\end{pmatrix} \begin{pmatrix}
\beta_x \\
T_x \\
\rho_x
\end{pmatrix}. \tag{8} \]

The complex and dimensionless stiffness \( \beta_x / \rho_x \) is obtained by means of Cramer’s rule

\[ \frac{\dot{\beta}_x}{\dot{\rho}_x} = \frac{\begin{vmatrix} -\gamma & 0 & 1 \\ 0 & 1/\Omega_x & 0 \\ 0 & 1 & -1 \end{vmatrix}}{\begin{vmatrix} 1 & 0 & 0 \\ 1 - E_{d, x} q_x^\prime & 1/\Omega_x & 0 \\ 1 & 1/\Omega_x & 0 \end{vmatrix}} = \frac{1 + \gamma i \Omega_x}{1 + q_x^\prime + \Omega_x (1 - E_{d, x} q_x^\prime)}. \tag{9} \]

derived from the system of equations (8) and characterizes the behaviour of the accumulator. We conduct a parametric study of the sorbent load \( q_x^\prime \) to investigate the accumulators characteristics. Figure 2 shows the respective resulting dynamic stiffness as a function of the frequency. Initially, the following two edge cases are considered. For \( \Omega_x \to 0 \) (first edge case \( \Delta t \to \tau \), i.e. isothermal behaviour) the dimensionless stiffness has the value \( 1/(1 + q_x^\prime) \). A conventional accumulator (fig. 1a) on the contrary has the asymptote one. The stiffness of an accumulator with sorbent material decreases due to the adsorption of gas molecules. Such behaviour verifies the above presumed rule as additional parallel capacity. An equally stimulated accumulator with sorbent material delivers lower pressure levels. For \( \Omega_x \to \infty \) (second edge case \( \Delta t \ll \tau \), i.e. isentropic behaviour) the dimensionless stiffness is \( \gamma/(1 - E_{d, x} q_x^\prime) \). Therefore an accumulator with sorbent material has a higher dimensionless stiffness compared to a conventional accumulator, which has a dimensionless stiffness of asymptotically \( \gamma \).

The transition range between both edge cases is found for the dimensionless frequencies \( 0.1 < \Omega_x < 100 \). The above results show, that this transition range of an accumulator with sorbent material has a higher gradient as well as an increased phase shift.

![Figure 2: Dynamic behaviour (transmission behaviour) of an accumulator partially filled with sorbent material.](image)

Aside from the dynamic stiffness, figure 2 additionally shows the temperature behaviour for various dimensionless sorbent charges \( q_x^\prime \) as a function of the dimensionless frequency \( \Omega_x \). The temperature behaviour is analogously computed by means of Cramer’s rule:

\[ \begin{align*}
\frac{\dot{V}_x}{\dot{T}_x} & = \frac{-\Omega_x (q_x^\prime + \gamma - (1 - E_{d, x} q_x^\prime))}{1 + q_x^\prime + \Omega_x (1 - E_{d, x} q_x^\prime)}.
\end{align*} \tag{10} \]

For \( \Omega_x \to 0 \) the accumulator shows isothermal behaviour, as above mentioned. For \( \Omega_x \to \infty \) the gas temperature of an accumulator with sorbent material increases compared to a conventional accumulator. This temperature increase is largely dependent on the sorption characteristics \( q_x^\prime \) and \( E_{d, x} \), as equation (8) implies. These observations emphasize the sorbent’s role as a heat source.

As an alternative to solving the non-linear differential equation system (5) by means of the ‘pen and paper’ method, numeric methods can be employed. The pressure- and temperature-response of the accumulator is numerically computed as a result of a given stimulation, for example a volume flow.
Figure 3 shows the behaviour of the stiffness as known from figure 2. The scatter plots represent the corresponding numeric solution of the non-linear differential equation system (5). It is clearly recognizable that the results obtained by means of linearization match the numeric results very well, especially for lower dimensionless stimulation frequencies. The transition range between purely isothermal and purely isentropic behaviour is represented appropriately as well. For higher dimensionless stimulation frequencies however, the results diverge significantly. These discrepancies can be attributed to the non-linear terms of the model (4), which become increasingly important at higher stimulation frequencies.

Figure 3: dynamic behaviour (transmission) behaviour of an accumulator partially filled with sorbent material.

Comparison of linearization and numeric computation of the model.

In the following section the behaviour of the accumulator is discussed by means of a p-V-diagram, for the four sorbent charges ($q'_s = 0, 0.841, 4.205$ and $8.41$) shown above. Figure 4 shows the typical hysteresis curves of the accumulators for a dimensionless frequency of $\Omega_1 = 1$. The dotted curves depict isotherms, the dashed curves depict isentropic behaviour. The known behaviour of a conventional accumulator is represented by $q'_s = 0$.

Examining accumulators with sorbent material ($q'_s > 0$), the isentropic pressure amplitude (dashed curve) increases slightly with increasing sorbent portion. In contrast, the isothermal pressure amplitude decreases with increasing portion of sorbent. Such behaviour was already evident in figure 2. At $\Omega_1 = 1$ two aspects of the accumulator’s behaviour are significant. First, the slope of the hysteresis decreases with increasing sorbent portion and converges towards isothermal behaviour, as was already evident in figure 2 as well. Secondly, the area of the hysteresis decreases with increasing sorbent portion. Since the area of the hysteresis quantifies the dissipation due to heat transfer into the environment, this result motivates the consideration of the efficiency $\eta$ as a function of the dimensionless frequency and dimensionless sorbent charge. Otis /15/ was the first to make the connection between efficiency and dimensionless frequency in 1975. He defined the efficiency as the ratio of emitted work $W_{21}$ and absorbed work $W_{12}$ (cf. figure 5) of the accumulator

$$\eta = \frac{W_{21}}{W_{12}}$$

(11)

Figure 6 accordingly shows the behaviour of the efficiency as a function of the dimensionless frequency for accumulators with and without sorbent material. The conventional accumulator reaches an efficiency minimum at $\Omega_1 = 1$, as already shown by Otis /15/. Examining accumulators with sorbent material, the following three important insights can be obtained: (i) The efficiency minimum decreases with increasing sorbent portion. (ii) The efficiency minimum occurs at higher dimensionless frequencies with increasing sorbent portion. (iii) The efficiency characteristics of accumulators with sorbent material and the efficiency characteristic of conventional accumulators intersect due to the displacement towards higher dimensionless frequencies (cf. figure 6). The accumulator with sorbent material therefore achieves higher efficiency compared to the conventional accumulator towards the left of this intersection. In the examined steady state of $\Omega_1 = 1$ this is the case for a dimensionless sorbent charge of 8.41 (cf. figure 6).
The design of a hydraulic accumulator is highly dependent on the system’s objectives and the corresponding operating parameters. The generally most important design criterion is the required displacement volume or the hydrostatic discharge energy, while the operating pressure range is mostly determined by the hydraulic system. Additional restrictions are imposed by safety requirements or other basic system conditions /16/.

Applications are usually categorized by the time scale of their duty cycle and the corresponding state transition. In practice, a process is considered slow (and thus isothermal) for charge and discharge cycles taking place in time frames $\Delta t > 3$ min, for example in leakage compensation applications. Duty cycles recognized as fast (and thus isentropic) processes have typical timeframes of $\Delta t < 1$ min. Application examples are shock absorbers or suspension systems /16/.

5 Model application on a practical duty cycle

The following section discusses the application of accumulators with sorbent material more detailed. First, it is essential to summarize the assumptions for the model design and the above findings. Secondly; it is important to consider the design restrictions imposed by the objective and the state transition characteristics of possible applications.

Assumptions

1. Zero-dimensional model design, i.e. all state variables are only functions of time.
2. The heat capacity of the sorbent is assumed to be negligibly small compared to the heat capacity of the gas. An extension of the equation (2b) is readily possible.
3. The permeation is neglected (not a general restriction imposed by the model).
4. Ideal gas characteristics are assumed (no general adjustment of the model needed).
5. A linear sorption isotherm is assumed, which does not include saturation of the sorbent material (no general adjustment of the model needed).

Findings

1. At low dimensional frequencies the sorbent acts as an additive capacity, which results in a reduction of the stiffness of the accumulator. This behaviour motivates the application of hydraulic accumulators with sorbent material in the context of an assembly size reduction.
2. The heat build-up of the accumulator as well as the increase of the stiffness at higher dimensionless frequencies is to be considered by the end user.

As can be conducted from Figure 2, the deployment of accumulators with sorbent material is only favorable for applications of isothermal behaviour, else the accumulator characteristic converges towards that of a conventional accumulator. Therefore, applications with generally faster duty cycles like pulsation dampeners for displacement pumps, blow molding or pressure injection machines and suspension systems cannot benefit from the utilization of accumulators with sorbent material.

Reasonable application objectives with generally longer duty cycles are leakage compensation as mentioned above, but also lubricant supply and maintaining system pressure over long time periods. Examples for the latter are clamp systems, throttle valves, hydraulic bearings or rolling mills /16/ /17/ /18/.

5.2 Applications

Given this background, we will exemplary examine the potential of an accumulator with sorbent material on the practical duty cycle of a hydraulic molding press. This is an application where the accumulator is used as a leakage compensator under isothermal conditions, hence the dimensionless frequency can be assumed to be $\Omega < 1$. The maximum permitted leakage is $2 \text{ cm}^3/\text{minute}$ over a time period of 60 minutes. The target pressure level is 200 bar with a minimum of 198 bar. This specification results in an initial volume of $13.3 \text{ l}$ and a displacement volume of $0.12 \text{ l}$. These are typical conditions and operating parameters for this type of application /19/ /20/.

We again study the system characteristics as a function of the remaining influencing parameter for the system $q'_c$. To solve the dimensionless equation system (5) of our model numerically, we use the volume as a function of time

$$V = V_0 - \Delta V \frac{1}{2} \left[1 + \sin \left(\Omega + \frac{3}{2} \pi \right)\right],$$

in the dimensionless form

$$V_c' = \frac{V'}{V_0} = 1 - \Delta V \frac{1}{2} \left[1 + \sin \left(\Omega' + \frac{3}{2} \pi \right)\right].$$

As figure 7 shows, the operating pressure levels of the original conventional accumulator can be retained by increasing the dimensionless displacement volume $\Delta V_c = \Delta V/V_0$. Since the displacement volume in this application is significantly smaller than the initial volume ($\Delta V / V_0$), we can on the one hand interpret the increase of the dimensionless displacement volume as a reduction of the initial volume and therefore the assembly size, while retaining the discharged volume and amount of energy (left-hand side of figure 7). On the other hand this can represent an increase of the achievable amount of discharge energy in an existing system with unchanged size by means of utilizing a sorbent material (right-hand side of figure 7).

Further applications with the purpose of covering a higher volume demand ($\Delta V / V_0$ over a long period of time are plausible. However, the ratio of $\Delta V / V_0$ must be considered properly. It restricts the achievable size reduction or gain of discharge energy and more importantly the ability to retain the original operating pressure level. The latter might not be an obstacle if a reduction of the operating pressure levels is acceptable or even desired.

Figure 6: Efficiency characteristic as a function of dimensionless frequency for different configurations.

Figure 7: Duty cycle of a conventional accumulator and an accumulator with sorbent material.
In the context of energy storage for emergency supply or kinetic energy recovery, the duty cycle time with respect to the relaxation time of typical accumulators cannot be considered ‘slow’ as clear-cut as the above mentioned examples. Both the expected charge and discharge cycle time, as well as the relaxation time may vary greatly depending on each particular use case. Many applications utilizing hydraulic accumulators for this purpose today are represented by dimensionless frequencies of $f_\text{rel} = 10$ and above. Typical duty cycle times range from 1 second to 1 minute, with accumulator volumes from 20 l to several 100 l and operating pressure levels of 100 bar to 400 bar (21/22/23). Nonetheless, accumulators of smaller size and lower pressure levels have decently low relaxation times (24), which in conjunction with peak load coverage occurring over longer time periods results in the desired slow state transition process.

6 Conclusion and future research

In regard to the achievable reduction of mass and size of hydraulic systems, hydraulic accumulators with sorbent material are an innovative and promising development. This paper introduces a generic model description for these novel accumulators. The results show, that an accumulator with sorbent material is softer at low dimensionless frequencies (i.e. in the isothermal edge case) and stiffer at high dimensionless frequencies (i.e. in the isentropic edge case) compared to conventional accumulators. Furthermore, the heat build-up of an accumulator with sorbent material at high dimensionless frequencies has to be considered.

A relevant application of accumulators with sorbent material is the deployment at low dimensionless frequencies, allowing a considerable reduction in size compared to a conventional accumulator. The operation of an accumulator with sorbent material in the transition range between isothermal and isentropic behaviour generally is to be avoided. In this range, small changes of the simulation frequency can lead to increasing pressure- and temperature amplitudes, due to the high gradients in the transition behaviour. Thus, reasonable application examples are leakage compensation, lubricant supply or maintaining system pressure. The sorption model assumes quasi-stationary behaviour, diffusion processes as a time determining aspect are not considered here. Achievable improvements over conventional accumulators depend on the sorbent characteristics which are unknown thus far. The applicability of accumulators with sorbent material for the purpose of energy storage or recuperation remains an open question for future research.

The next research step is the development of a prototype accumulator with sorbent material – corresponding to the outlined idea - for the validation of the model design. Characterizing applicable sorbent materials imposes an additional challenge and will require experimental investigations. These investigations are to be conducted in the scope of the Collaborative Research Center 805 ‘Control of uncertainty in load carrying structures in mechanical engineering’ (SFB 805) at the Technical University Darmstadt.

7 Acknowledgements

The authors would like to thank the German Research Foundation (DFG) for funding this research within the Collaborative Research Centre (SFB) 805 ‘Control of uncertainty in load carrying structures in mechanical engineering’ (TU Darmstadt), speaker Prof. Dr.-Ing. Peter F. Pelz.

Nomenclature

The first column of the following table shows the symbols utilized for physical and mathematical quantities. The second column shows the meaning of each quantity. The dimension of each physical quantity is denoted in the third column as a product of power terms, based on the generic quantities length (L), mass (M), time (T), amount of substance (N) and temperature (Θ). Dimensionless quantities are denoted by a plus symbol in the index, disturbing variables are marked by tilda or circumflex.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a$</td>
<td>thermal diffusivity coefficient</td>
<td>$L^2T^{-1}$</td>
</tr>
<tr>
<td>$A$</td>
<td>cylinder surface</td>
<td>$L^2$</td>
</tr>
<tr>
<td>$A_{\text{surf}}$</td>
<td>sorbent surface</td>
<td>$L^2$</td>
</tr>
<tr>
<td>$c_p$</td>
<td>specific isobaric heat capacity</td>
<td>$L^2T^{-2}Θ^{-1}$</td>
</tr>
<tr>
<td>$c_v$</td>
<td>specific isochoric heat capacity</td>
<td>$L^2T^{-2}Θ^{-1}$</td>
</tr>
<tr>
<td>$d$</td>
<td>characteristic diameter of sorbent material</td>
<td>$L$</td>
</tr>
<tr>
<td>$δ$</td>
<td>phase angle</td>
<td>$1$</td>
</tr>
<tr>
<td>$D$</td>
<td>diameter of the accumulator</td>
<td>$L$</td>
</tr>
<tr>
<td>$e$</td>
<td>inner energy</td>
<td>$L^2T^{-2}$</td>
</tr>
<tr>
<td>$E_A$</td>
<td>bond energy</td>
<td>$L^3M^{-1}T^{-2}N^{-1}$</td>
</tr>
<tr>
<td>$η$</td>
<td>efficiency</td>
<td>$1$</td>
</tr>
<tr>
<td>$h$</td>
<td>enthalpy</td>
<td>$L^2T^{-2}$</td>
</tr>
<tr>
<td>$i$</td>
<td>imaginary number</td>
<td>$1$</td>
</tr>
<tr>
<td>$k$</td>
<td>heat transmission coefficient</td>
<td>$MT^{-1}Θ^{-1}$</td>
</tr>
<tr>
<td>$M$</td>
<td>molar mass</td>
<td>$MN^{-1}$</td>
</tr>
<tr>
<td>$m_{\text{perm}}$</td>
<td>mass flow due to permeation</td>
<td>$MT^{-1}$</td>
</tr>
<tr>
<td>$n$</td>
<td>polytropic exponent</td>
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<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
<td>$1$</td>
</tr>
<tr>
<td>$p$</td>
<td>pressure inside the accumulator</td>
<td>$L^{-1}MT^{-2}$</td>
</tr>
<tr>
<td>$p_0$</td>
<td>referenced pressure</td>
<td>$L^{-1}MT^{-2}$</td>
</tr>
<tr>
<td>$p_{\text{oil}}$</td>
<td>oil pressure</td>
<td>$L^{-1}MT^{-2}$</td>
</tr>
<tr>
<td>$p_a$</td>
<td>ambient pressure</td>
<td>$L^{-1}MT^{-2}$</td>
</tr>
<tr>
<td>$q$</td>
<td>sorbent charge, surface specific mass of agglomerated gas molecules</td>
<td>$L^2M$</td>
</tr>
<tr>
<td>$Q$</td>
<td>volume flow</td>
<td>$L^2T^{-1}$</td>
</tr>
<tr>
<td>$Q_0$</td>
<td>heat flow</td>
<td>$L^2MT^{-2}$</td>
</tr>
<tr>
<td>$R$</td>
<td>specific gas constant</td>
<td>$L^2T^{-2}Θ^{-1}$</td>
</tr>
<tr>
<td>$s$</td>
<td>specific surface of the sorbent material</td>
<td>$L^1$</td>
</tr>
<tr>
<td>$S$</td>
<td>specific surface of the accumulator</td>
<td>$L^{-1}$</td>
</tr>
</tbody>
</table>
\[ t \quad \text{time} \]
\[ T \quad \text{gas temperature inside the accumulator} \]
\[ T_0 \quad \text{referenced temperature} \]
\[ T_a \quad \text{ambient temperature} \]
\[ V \quad \text{volume of the accumulator} \]
\[ V_0 \quad \text{referenced volume, volume of the accumulator at operation state} \]
\[ V_{\text{solid state}} \quad \text{solid state volume of the sorbent material} \]
\[ V_{\text{cavity}} \quad \text{cavity volume of the sorbent material} \]
\[ \gamma \quad \text{isentropic exponent} \]
\[ \varepsilon \quad \text{sorbent porosity} \]
\[ \lambda \quad \text{thermal conductivity coefficient} \]
\[ \vartheta \quad \text{porosity} \]
\[ \theta_0 \quad \text{referenced density} \]
\[ \tau \quad \text{thermal relaxation time} \]
\[ \Omega \quad \text{stimulation frequency} \]

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Polymer composites materials for water hydraulic seat on/off valves

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High pressures and harsh working conditions in hydraulic systems has made us sceptical about suitability of plastics for its components. Nevertheless in some cases it can become a sufficient substitute for expensive steels. In water hydraulic components, where demanding surface contacts are slowing its development, polymers can be a solution. Focus of our research is on implementing polymers into a moving contact in high speed water hydraulic on/off valve where high friction and wear occur. In this article we are presenting friction coefficient and wear rate of some engineering polymers immersed in water for different time periods. PEEK and POM showed comparable results regardless their price difference.

Keywords: water hydraulics, friction coefficient, wear rate, polymers, composites

Target audience: Tribology, Polymers, Digital Hydraulics

1 Introduction

Hydraulic, as a widely used technology, has not yet explored all the possibilities of polymers. They are already used as a material for seals and minor parts, such as handles, covers, caps, etc., but still the dominant material for making of the key elements is steel. In oil hydraulics, working mediums do not cause corrosion and have good lubricating properties thus steels are compatible materials. But oils are not fire resistant, they are expensive, they are not suitable for under water tool systems and they are not environmentally friendly. Water hydraulics therefor can represent good alternative but normal steels are not appropriate because of corrosion sensitivity. Another problem arises from water properties that differ from oil. Water’s kinematic viscosity is around 100 times lower what makes it easier to flow through small gaps and to create turbulences. Vapor pressure is a lot higher thus ensuring steady pressure is important, and it’s working temperature is limited to only 50 °C.

Listed above effects the construction process and costs of water hydraulic components. In terms of material selection, new solutions are researched in order to achieve lower friction, lower weight and, if possible, lower costs of production and costs of raw materials.

In literature several material groups are presented as a solution for different water hydraulic components, stainless steels, as most common ones, engineering ceramics, ceramic coatings, carbon coatings like DLC (diamond like carbon), nitride coatings, polymers and polymer composites.

Our application is water hydraulic is seat on/off valve where two types of contact occur, impact seat contact and sliding contact. First requires a material that is not fragile and has good impact strength and the second requires low friction coefficient. Ceramics and coatings have good wear resistance, high hardness, good chemical stability and low adhesion but, low impact strength in combination with water causes cracking and creating damaging debris. Material groups that are not so fragile are polymers and polymer composites. Literature suggests different polymers which shows low friction coefficient when sliding against harder material. Most common are PEEK, POM, PTFE, PPS, PI, PA, and their composites with graphite fibers, glass fibers, carbon fibers, graphite nanotubes, etc. Polymers have different mechanical properties since they are viscoelastic materials. Thus different problems as water absorption and creep arises that need to be tested and evaluated.

In this paper preliminary results of PEEK, POM, PA6 and PA66 with 30 % glass fibers materials friction coefficient and wear rate will be presented based on water absorption. This article structure will be as follows, chapter 2 covers one stage on/off valve construction, chapter 3 includes basic polymer properties and their main problems when exposed to water applications, chapters 4 and 5 presents experimental work and obtained results and in the end in chapter 6 summary and conclusions are presented.

2 One stage seat on/off valve

Simple construction is what makes it a suitable component to start the research of appropriate materials. It is assembled from housing, electromagnet with a coil and moving armature, poppet and a poppet seat. Return mechanism is made of spring (see Figure 1).

![Figure 1: Scheme of a simple hydraulic seat on/off valve, 1- armature, 2- housing, 3- poppet, 4- return spring, 5- cover.](image)

Electromagnetic actuator uses a coil to induce magnetic field which creates a magnetic force that pushes the armature and consequently the poppet. With the loss of a current, coil loses its force and poppet moves back with a help of a return spring. While the poppet is pressed on its seat, the valve is off and no fluid flow is present, when the poppet is pushed down with the help of a magnetic force, the valve opens and allows fluid to pass.

First problem arises between poppet and housing. In theory, the poppet floats in the housing, but with a high water pressure poppet is coincidentally pushed towards housing in all directions what causes friction and the loss of energy. Second, wear occurs on a valve poppet which hits the seat every time valve is turned off. Impacts damage poppet’s conical surface that leads to leakage and increases pressure loss.

3 Properties of polymers

In order to introduce polymers into water applications, some differences in behavior of materials needs to be considered. Polymers behave differently and the discrepancy between metallic and polymeric friction is due to the differences in the elastic-plastic behavior of metals and the viscoelastic behavior of polymers. Factors that affects friction and wear of polymers are sliding speed, temperature, counter face roughness, applied load and contact pressure, material properties, lubrication and fatigue of polymers.

Dependence of friction coefficient to the sliding speed its complex and hard to define. With speed, wear increases due to increase in the contact surface temperature. Thermoplastics have a critical sliding speed over
which wear rate slightly reduces due to melting and thermal softening. This critical point is a consequence of adhesive friction (see Figure 2) at the instantaneous contact interface and thus cannot be measured //24/.

Temperature, as mentioned, changes during sliding thus mechanical properties of polymers show transition from glassy to rubbery state. Some analytical and experimental models were obtained by various researchers. Models take into account different correlations between temperature, friction coefficient, sliding speed, contact pressure, semi contact length and width, environmental and/or contact temperatures and normal load. Since some of these parameters are difficult or impossible to define modelling is not always a solution //24/.

Counterface roughness has a strong correlation with tribology of polymeric materials in case counterface is a metal surface with higher hardness. During sliding, as the metal surface roughness decreases, friction coefficient decreases until minimum roughness value is reached. Further, decrease in roughness causes high friction. Turning point arises when adhesion forces prevail abrasive wear. Some materials shows vice versa trends where with increasing of counterface roughness friction decreases, but wear increases //24/.

![Figure 2: Display of four wear types //29/.

With increasing contact pressure and load, it is well known that increase in contact temperature, softening and plastic deformation of polymers occur since plastic deformations change real area of contact thus friction and wear are increased. For many thermoplastics experiments have shown that for high loads from 10 N to 100 N, friction coefficient is constant. For lower loads from 0.2 N to 1 N plastic deformation governs the sliding process with increase of friction coefficient. Thus transition from elastic to plastic contact plays a big role in friction and wear. Relation between friction force $F$ and applied normal load $L$ can be expressed with equation (1).

$$F = \mu L^n$$

(1)

Where $\mu$ is friction coefficient and $n$ is exponential constant that is different for different polymers //24/.

As already understood material properties are crucial in material behavior. Why polymers are so difficult to track and model is due to constant mechanical and physical property change. Mechanical properties as yield strength, hardness, elastic modulus and impact strength have a big effect on glass transition properties. They decrease with increasing temperature thus wear increases. Physical properties of thermoplastics on the other hand, govern chemical composition and interfacial energy. Both determine the rate of adhesion. Polymers with lower surface energy tends to transfer to materials with higher surface energy. Transferred film therefor depends also on physical properties //24/.

Another influential parameter to be considered when applying polymers in water hydraulics is atmospheric humidity. Water molecules diffuse into free volume of the amorphous phase of polymers what leads to plasticization, swelling and softening. Diffusion also reduces the attractive forces between polymer chains. Thus the amount of atmospheric humidity not only washes away transfer film that was supposed to create on the counterface but also has a pronounced effect on the wear and friction of polymers. According to literature the wear rate under water lubrication was always higher //19-24/.

No matter the conditions, wear of polymers is with time increasing due to fatigue of the material. During working cycles materials are also affected by cycle frequency, loading wave form, stress ratio, molecular weight of polymer, etc. The rate of fatigue can be improved with higher rate of crystallinity but it can also be exacerbate under harsh operating conditions //24/.

In small hydraulic seat on-off valve it is impossible to measure working parameters such as contact surface temperature and normal load to the friction surface since small gap tolerances, analytical models are therefore difficult to apply //25/. In order to measure friction coefficient and wear rate of material pairs that interests us, we chose standard tribological test which parameters are closest to working parameters in seat on/off valves. The apparatus and working parameters are presented in following chapter.

## 4 Experimental work

Tribological behavior of several unfilled and one filled polymer material sliding against 316 L stainless steel in water is studied. Coincidental touching is hard to measure and impossible to repeat therefore we chose standard tribological apparatus (TE 77) with a reciprocal movement to simulate the movement of a poppet in a housing.

### 4.1 Materials

Preliminary selected polymers are suggested by literature and are well known in other engineering applications like in pumps, bearings, etc.

#### 4.1.1 Polyoxymethylene

POM is a semi-crystalline polymer with high crystallinity ($70-100\%$), good friction and wear properties, high strength, and good chemical stability. It is widely used to replace the traditional metals and ceramics in microelectronic packaging, aerospace, automotive, and biomedical applications. Its low water absorption makes it suitable also for water applications.

#### 4.1.2 Polyetheretherketon

PEEK as a typical high performance semi-crystalline thermoplastic polymer, has received significant attention. This is due to its high mechanical strength and elastic modulus, high melting temperature, chemical inerterness, high toughness, easy processing and wear resistance. On the other hand, in tribological applications, because of the corrosive problems of the metals in water applications, polyetheretherketone (PEEK) and polyetheretherketone composites are preferred as rubbing materials. Therefore PEEK polymer material plays more important role as a bearing and slider material especially under water environment //22/.

#### 4.1.3 Polyamides

Polyamides (PA6 and PA66) are also semi-crystalline polymers used for many engineering parts undergoing friction and wear (bearings and gears) //26/. Their friction and wear properties are attributed to the presence of hydrogen bonds in polyamide molecular chains //27/. Even though they absorb more water than other chosen polymers they are low cost and thus approachable.

![Diagram of four wear types](image-url)
Last polymer composite in preliminary measurements is PA66 with 30 % of glass fibers. PA66 is the hardest and toughest among Polyamides. It absorb slightly less water than PA6 and it has better wear resistance. With addition of glass fibers composite shows increased strength, rigidity and service temperature.

The counter surface in all experiments was AISI 316 L, also known as marine grade stainless steel. Table 1: Characteristics of polymer materials from technical data. Table 1 presents characteristics of these selected materials.

All materials were bought in a company EX-MEGA d.o.o. already shaped in 1 m long bars of diameter 10 mm. Material technical data holds no information about production process and storing conditions.

4.2 Test machine and specimens

Measuring apparatus was high frequency friction machine TE 77 presented on a Figure 3. It has a water bath container inside which counterface stainless steel 316 L specimen was mounted. Polymer specimen was mounter into a sample holder which is connected to drive shaft via clamping assembly.

4.3 Experimental procedure

As described, wear and friction coefficient of polymers is a function of variety of contact and environmental factors, parameters of our measurements are presented in a Table 2. Frequencies of water hydraulic seat on/off valves are currently in a range of 10 Hz what is relatively low for applications as digital hydraulics where frequencies evolve around 100 Hz and more. Measurements were carried out at a constant frequency of 40 Hz for a 36 000 cycles due to friction coefficient stabilization. Normal force is unmeasurable therefore we used similar pressures and suggested in a literature, 15 MPa Hertzian pressure //19/-23/. Therefor 5 N was applied via weighting lever and was kept constant.

<table>
<thead>
<tr>
<th>Repetitions per material</th>
<th>3x</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency</td>
<td>40 Hz</td>
</tr>
<tr>
<td>Number of cycles</td>
<td>36 000</td>
</tr>
<tr>
<td>Approximate time</td>
<td>15 min</td>
</tr>
<tr>
<td>Normal force</td>
<td>5 N</td>
</tr>
<tr>
<td>Max. displacement</td>
<td>2,4 mm</td>
</tr>
<tr>
<td>Total distance</td>
<td>172800 mm</td>
</tr>
<tr>
<td>Medium</td>
<td>Demineralized water</td>
</tr>
<tr>
<td>Water temperature</td>
<td>21 °C</td>
</tr>
<tr>
<td>Irrigation</td>
<td>7, 14, 24, 31 days</td>
</tr>
</tbody>
</table>

Table 2: Experimental conditions.

Polymer specimen holder was moving in a reciprocal direction with a maximum distance of 2,4 mm what is the shortest distance possible with chosen apparatus (see Figure 5). Fluid for water bath and irrigation was demineralized water with room temperature to eliminate any on the measurements and to ensure repeatability. Every sample, when taken out of a bath, was dried with paper and weighted right after removing it.
from water and before it was used in tribological tests. Three repetitions were done with every sample and between every repetition sample was again weighted due to wear. Friction force was measured by a piezoelectric force gauge mounted under the water container.

5 Results and discussion

As a follow up, tracking of water absorption was observed due to easier understanding of polymers chemical composition. As expected, PA6 absorption percentage was the highest followed by PA66. For PEEK and POM the line almost become constant and from all, PEEK absorbed the smallest amount of water during our measurements nevertheless it has the same water absorption % as POM in technical data (see Table 1).

Figure 5: Movement direction of polymer specimen on a stainless steel disc.

5.1 Friction

Friction was measured after every irrigation time under the same conditions. Figure 7 and Figure 8 represents friction coefficient during measuring cycles for 14 and 31 days of water absorption.

Stabilization of friction coefficient was achieved for all polymers and composite except for PA6 whose value was still dropping after 15 min. With higher water percentage this effect was even bigger and coefficient value was dropping even slower.

PEEK and POM have almost the same friction coefficient at lower water absorption percentage what can also be seen in material technical data but the values in reciprocal movement are almost double, 0.66.

PA 66 + 30 GF friction coefficient has more steady curve and reaches lower values as a unfilled PA6 but it can be seen that PA 6 is approaching the same values with time.
Summary and conclusion

The main contribution of this paper is acknowledgement of water absorption influence on tribological properties of polymers and composites. Tribological behaviour of polymers POM, PEEK, PA6 and a composite PA66 with 30 of glass fibers was monitored in water lubricated sliding contacts with pre-immersed specimens for different periods of time.

For all materials tested we can conclude that due to polymeric visco-elastic behavior and ability of water absorption, the time of immersion have significant influence on tribological properties thus full saturation has to be considered when choosing polymeric material for water applications. Also sliding direction have an influence on friction coefficient and consequently on wear rate of polymers and composites therefore standard rotational pin-on-disc tribological measuring process cannot be a measure for tribological parameters in applications with reciprocal movement as present in our water hydraulic seat on-off valve.

PA6, as expected, absorbed the highest percentage of water thus its friction coefficient was dropping and stabilizing slower with water percentage increase. This can be identification of continuous adhesion and abrasion since material was getting softer. Nevertheless it was still approaching the same value as achieved by PA 66 + 30 GF thus we can question the benefits of glass fibers when polymers are saturated.

PEEK and POM absorbed the less percentage of water therefore their friction coefficient was more stable and both were able to maintain lower values than observed with referential stainless steel. Considering the price differences of this two polymers, POM’s price to friction coefficient ratio is sufficient for or needs.

Preliminary results give us guidelines for our work to follow. For further tribological measurements we will exclude all engineering polymers with high water absorption percentage regardless their low cost benefits. Tribological properties of unfilled polymers and some composites with carbon fibers will be measured at full saturations based on ISO 62. For carbon composites, debris due to wear can cause contamination therefore it will be monitored and evaluated after the measurements. Before approaching implementation in final application we will also test the most suitable polymers and composites to impact stresses that occurs in water hydraulic seat on-off valve.

5.2 Wear

Wear was measured with weighting after every measurements. As presented in Figure 10, PA6 has the highest wear. The lowest wear was achieved by PA 66 + 30 GF therefore fibers do protect softer polymeric matrix but in this case do not contribute to lower friction. As for POM wear increases and decreases with higher water absorption percentage, PEEK is showing a big decrease.

Figure 10: Average wear rate in dependence of water absorption time.

6 Summary and conclusion

The main contribution of this paper is acknowledgement of water absorption influence on tribological properties of polymers and composites. Tribological behaviour of polymers POM, PEEK, PA6 and a composite PA66 with 30 of glass fibers was monitored in water lubricated sliding contacts with pre-immersed specimens for different periods of time.

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Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>F</td>
<td>Friction force</td>
<td>[N]</td>
</tr>
<tr>
<td>μ</td>
<td>Friction coefficient</td>
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</tr>
<tr>
<td>n</td>
<td>Exponential constant</td>
<td>[/]</td>
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<tr>
<td>L</td>
<td>Normal load</td>
<td>[N]</td>
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</table>

References


Field tests of the DOT500 prototype hydraulic wind turbine

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To reduce turbine mass, maintenance requirements, complexity, and thus the Levelized Cost of Energy (LCOE) for offshore wind, the Delft Offshore Turbine (DOT) concept combines individual hydraulic drive train wind turbines with a centralized generator system. In 2015 DOT built and tested a large-scale prototype, by retrofitting a 600kW wind turbine with a hydraulic drive train using commercial off-the-shelf components. The goal was to showcase a proof of concept from a technological and controllability point-of-view. This paper presents the results of building and testing the DOT500. Its drive train has an oil-hydraulic stage and a water-hydraulic stage. The method of rotor torque control with spear valves is novel and proves to be a substitute for conventional implementations.

Keywords: offshore wind, fluid power transmission, water hydraulics

Target audience: offshore wind industry, water hydraulic component suppliers

1 Introduction

1.1 Background DOT hydraulic wind turbine concept

The drive train of horizontal-axis wind turbines (HAWTs) generally consists of a rotor-gearbox-generator configuration in the nacelle, which enables each wind turbine to produce and deliver electrical energy independent of other wind turbines. While the HAWT is a proven concept, the turbine rotational speed decreases asymptotically and torque increases exponentially with increasing blade length and power ratings. The increased loads primarily affect the gearbox-generator combination, which makes it a maintenance critical and high mass component in the turbine /1/. The complete wind turbine support structure is designed to carry this mass, which in turn leads to extra material, mass and thus cost of the wind turbine /2/. Offshore wind turbines are getting ever larger, resulting in lower rotation speed and higher torque at the rotor axis. With that, the case for compact hydraulic power trains is becoming ever stronger.

In an effort to reduce turbine mass, maintenance requirements, complexity, and thus the Levelized Cost of Energy (LCOE) for offshore wind, hydraulic drive train concepts have been considered in the past /3/. However, so far none have been commercially realised.

The DOT concept consists of a seawater hydraulic network where every wind turbine rotor is directly coupled to a positive displacement pump and electricity generation is centralized, creating an offshore hydro-powerplant, as schematically represented in Figure 1. This system also enables multiple wind turbines to be controlled collectively which distinguishes it from earlier proposed hydraulic drive trains. The main benefits of this drive train system with respect to the current state-of-the-art are:

- High torque to mass ratio, i.e. mass reduction of the wind turbine nacelle by more than 50% /4/
- Simplification of offshore wind power electronics, a significant contribution to turbine downtime /5/

Figure 1: The DOT concept: a seawater hydraulic network where every wind turbine rotor is directly coupled to a positive displacement pump and electricity generation is centralized

1.2 The necessary first step of development

So far, all experiments had been done in a controlled indoor environment. Hence the first challenge after founding the company was to build and test a hydraulic drive train outdoors, in a real world and full-scale wind turbine.

In a collaboration between DOT and the TU Delft, a project was set-up in 2015 to retrofit a second-hand 600kW Vestas V44 onshore turbine into a full-scale DOT hydraulic wind turbine.

Figure 2 shows the typical power flow from wind to electricity. The wind turbine rotor converts wind power to mechanical power in the form of (high) torque and (low) rotation speed. A major challenge for the development of DOT is the commercial unavailability of low-speed, high-torque water pumps to connect directly to the rotor. However, for oil hydraulics such machines are available, which resulted in the idea to add an oil circuit which acts as a hydraulic gearbox, as schematically represented in Figure 3. This idea was already presented at the IFK 2014 in Aachen along with the simulated power performance /6/ of a 500kW DOT. Hence, to speed up development and showcase the practical feasibility of the hydraulic drive train, this intermediate concept was implemented using off-the-shelf components. Based on previous research, the design for a 500kW hydraulic drive train was made to fit the wind turbine. The reconfigured turbine was dubbed the DOT500. The hydraulic drive train was modelled theoretically and numerically prior to executing an extensive in-field test plan; using this set-up, two alternative torque control strategies were evaluated, and rotor and generator power optimizations were performed.

Figure 2: Wind power flow schematic
One of the few manufacturers in the world that supplies pumps with a range of operation covering those of wind turbine rotors is Hägglunds, part of Bosch Rexroth Group. For the DOT500 the CB840 motor was selected to function as pump, directly coupled to the turbine rotor. An overview of the final setup of the DOT500 drive train is shown in Figure 5. Oil hydraulic piping (three hoses with 2-inch inner diameter) was dimensioned for pressure loss of less than 1% and runs roughly 40 meters from the pump in the nacelle via a swivel (for continuous yaw-rotation) to the tower base. Here, a Bosch A6VLM oil motor converts hydraulic power into mechanical power. At this location, the inline filters are placed along with a hydraulic power unit (HPU), which controls the pressure in the oil feed lines (three 3-inch hoses). Filled up, the piping thus contained roughly 1000 liters of inherently biodegradable oil with viscosity 68cSt at 40°C. At normal operating conditions, the oil temperature is around 50°C with viscosity of around 43cSt.

The oil motor is coupled to a KAMAT 80120G water pump. Two 3-inch hoses direct the high-pressure water flow away from the wind turbine to the nearby Pelton turbine-generator. At the Pelton turbine, two spear valves control the two nozzle areas and thereby the pressure in the both the water and oil hydraulic circuits, and hence the braking torque experienced by the wind turbine rotor.

Due to the lack of a grid connection, power from the generator was either re-used for boost and control systems, or dissipated into heat by a break-resistor. The water exiting the Pelton turbine is caught in a reservoir. A centrifugal pump sends the water via a filter back to the water pump at the tower base.

An overview of the properties of the rotor and the drive train of the DOT500 are given in Table 1.

<table>
<thead>
<tr>
<th>Turbine properties</th>
<th>Oil circuit properties</th>
<th>Water circuit properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor diameter</td>
<td>44m</td>
<td>Nominal pressure oil</td>
</tr>
<tr>
<td>Nominal power</td>
<td>600kW</td>
<td>Nominal oil flow rate</td>
</tr>
<tr>
<td>Nominal speed</td>
<td>28rpm</td>
<td>Stroke volume 1</td>
</tr>
<tr>
<td>Nominal torque</td>
<td>205kNm</td>
<td>Stroke volume water motor</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Pelton wheel pitch diameter</td>
</tr>
</tbody>
</table>

Table 1: DOT500 rotor and drive train properties.

At nominal conditions for the DOT500 set-up, the water circuit experiences a flow of approximately 3500l/min. A significant design constraint is the required minimum flow for efficient performance of the Pelton turbine. This effectively limits the design pressure for the water hydraulic circuit at 80bar. If the flow is insufficient, the water jet (partly) disperses into mist before it hits the buckets on the Pelton wheel, thus reducing the efficiency of the energy conversion process /7/. Thus, if more flow were available, the design pressure could be increased, whilst maintaining maximum Pelton turbine efficiency.

2 Overview of main drive train components

At rated conditions the wind turbine produces approximately 600kW of mechanical power at the rotor, at rotational speed and torque of 28RPM and 205kNm, respectively.
3 Design for safe operation

The most important safety feature in any wind turbine is that one should always be able to stop the rotor. In modern wind turbines, this is done by pitching the blades to the vane position.

An additional method for stopping, enabled by the hydraulic configuration and implemented DOT500 wind turbine, is by increasing the system torque. For the event that pitch actuation would not work, a large cartridge valve was included. By activating this valve, the flow in the high-pressure side of the pump manifold is choked, thereby increasing the pressure and thus system torque, decelerating the rotor. The challenge of implementing this component was to avoid choking the flow too much too fast, which could lead to an extreme pressure surge and high forces on the blade roots due to rapid rotor deceleration.

The emergency stopping procedure was handled via a three-staged approach:

1. Pitch emergency stop: an emergency valve is opened in the pitch hydraulic power unit. The blades are pitched towards feather (90 degrees) at maximum pitch rate.
2. Hydraulic emergency stop: if the rotor does not decelerate sufficiently fast within a predefined time period after initiation of step 1, the hydraulic emergency stop is activated. The high-pressure oil line is choked, leading to a controlled increase in pressure, which leads to an increased system torque. The system torque is increased up to the point where it exceeds the aerodynamic torque from the rotor, leading to rotor standstill. To avoid excess breaking torque, the choke setpoint is automatically regulated as function of the operating pressure. Hence this method is applicable throughout the operational envelope of the wind turbine.
3. Electronic parking brake: once the rotor is at standstill, this step is initiated. All electronic power to the turbine is cut-off. The last step is never activated when the rotor is still running, as all turbine control is lost when power is lost.

The health of the turbine was continuously monitored via a staged alarm handling protocol. The three main turbine states were: Idle (turbine idling, waiting for input), Standby (turbine ready to operate, all boost systems active) and Power Production (turbine operational, producing electricity). Forward and backward switching between states is done via four transient states, in which the elapsed time is monitored and compared to a threshold value: (1) start-up of oil and water boost systems, (2) rotor start-up sequence, (3) rotor stopping sequence and (4) shutdown of oil and water boost systems.

4 Results

4.1 Nacelle mass reduction

From the original drive train, the generator and the gearbox were replaced by an oil-hydraulic pump, see Figure 6. This led to a nacelle mass reduction from 21.4 to 14.5 tons (more than 32%), excluding mass savings from removal of obsolete structural steel.

Figure 6: The original nacelle with gearbox and generator (left) and the DOT500 nacelle with pump (right)

4.2 Controllability

The DOT500 wind turbine operation is controlled in two ways:

1. Aerodynamically, using the blade pitch control system
2. Hydraulically, using the spear valve-nozzle combination at the Pelton turbine

The second is a feature unique to the DOT concept and unprecedented.

Part of the test program was the validation of the hydraulic torque control strategy in real life conditions. Two spear valve torque control strategies were evaluated. The first (A) was based on operating at maximum rotor power, i.e. aerodynamic efficiency, extracting as much power as possible from the wind. The other (B) strategy was based on the idea of passive nozzle control /8/, which was achieved by operating at or near maximum rotor torque. The spear valves are set at a predetermined set position. As a consequence, the available mechanical power from the rotor decreases, but overall system power increased as a result of operating in more favorable efficiency envelopes of the individual system components. The difference in these strategies is manifested below-rated wind speed.

At start-up, the blades are pitched to 45 degrees to gain torque whilst the spear valves are fully opened, minimizing system torque to speed up the rotor. In below-rated operating conditions, the spear valve position is used to control the system torque, (function of pressure determined by spear valve area) as function of the rotor speed. In this control region, the blade pitch is fixed at 6 degrees, the so called fine-pitch angle. In near-rated conditions, the spear valves are actively controlled to regulate the rotor towards rated speed. For above-rated conditions (600kW rotor power), the corresponding spear valve minimum area setting is maintained, fixing the rated system torque. Blade pitch control ensures the rotor speed is regulated to its nominal value of 28 RPM.

The in-field tests proved that in the below-rated region, maximum rotor power control can be achieved, but that operating the rotor at maximum torque with passive nozzle control yielded more power output and thus higher total system efficiency. In above-rated conditions, the control strategy stabilizes the turbine and succeeds in maintaining rated rotor speed and power using active pitch control. More details on the controls of the DOT500 are found in /9/.

A concern raised beforehand was the possible occurrence of stability problems due to the large volume of oil in the piping. However, this was never observed. Two reasons for this are the large mass moment of inertia of the rotor in relation to the fluid inertia in the piping and the damping effect caused by the relatively low efficiency of the oil motor.

4.3 Power performance

Figure 7: Left: experimentally derived rotor power curve of the DOT500 turbine. The grey dots represent field recorded data, binned per 0.5 m/s wind speed in the blue line. Results are compared to the original Vestas V44 data sheet (red) and theoretical maximum power (green). Right: the power vs pressure and flow in the water hydraulic circuit.
Figure 7 shows the rotor power of the DOT500 hydraulic wind turbine over its entire operational envelope for maximum rotor power control. The turbine is able to start from a cut-in wind speed of 4.5 m/s and operates with active spear valve control near optimal aerodynamic efficiency up to a wind speed of approximately 11 m/s. The right side of the figure shows the operational envelope in terms of pressure, flow and mechanical power at the Pelton turbine axis.

The fluid power transmission system between rotor and generator experienced severe losses. This is due to the mismatch of component efficiency envelopes and the addition of the oil circuit. Figure 8 shows the results of a drive train component efficiency analysis. It is evident that in each power conversion step energy is lost, as was expected.

4.4 Technical issues encountered

Two notable technical problems were encountered during the commissioning and testing of the DOT500:

1. A low-pressure oil hose failed in the turbine nacelle. During each turbine operational cycle, the nacelle oil hoses are filled and partially emptied after turbine shut down, due to gravity and leakage via the main pump. This led to a fatigue related failure of the inner steel part of the hose. After installing a new hose and wrapping it in SpiroFlex to prevent further fatigue, operation was continued. The oil mitigation equipment and procedures were effective in containing the spill.

2. Part of the piping of the water circuit was made of uncoated low-grade steel. This led to corrosion, contaminating the water circuit with rust. As a result, the filter in the water circuit rapidly saturated. This was mitigated by setting up a second filtering circuit for the water in the reservoirs.

6 Summary and Conclusion

The DOT500 became a fully functional hydraulic wind turbine, with fully automated and safe operation. After retrofitting the V44 drive train, a 32% nacelle mass reduction is attained. The novel spear valve torque control technology enables active regulation of the rotor speed. The total power transmission efficiency was predictably low, as a result of the double hydraulic circuit.

During turbine commissioning, a supervisory control scheme was developed, together with a turbine fault detection system, enabling safe operation of the turbine under all conditions. The developed DOT hydraulic wind turbine control strategies proved to be safe and stable over the full turbine operating range. Active and passive spear valve control are a feasible control substitute for industry standard generator torque control.

The DOT500 drive train was built with an oil pump in the nacelle. The next step for DOT is to develop a low-speed high-torque seawater hydraulic pump, that can be coupled directly to the rotor of a turbine /10/. This will enable the construction of a wind turbine with a single (sea)water hydraulic drive train.

6 Acknowledgements

The research presented in this paper was part of the DOT500 ONT project, which was conducted by DOT in collaboration with the TU Delft and executed with funding received from the Ministerie van Economische zaken via TKI Wind op Zee, Topsector Energie.

References

This paper deals with light-emitting phenomena in hydraulic components, which are closely linked to cavitation. Both the micro-diesel effect and the gas discharge have been optically investigated within plane models of a valve and a pump section, respectively. The gas discharge is caused by an electrostatic charge of the oil or of the component. One result of the investigations is an overview of the areas of occurrence and the minimum necessary operating conditions of the phenomena. The form of appearance of both phenomena is also shown. Furthermore, the impact of electrically insulating materials is presented. In addition some measurements of the temperatures in close proximity to the phenomena are presented.

Keywords: Micro-Diesel Effect, Cavitation, Valves & Pumps, Electrostatic Discharge

1 Introduction

Cavitation is an omnipresent process in the field of hydraulics. A local pressure drop causes evaporation and/or degassing of the liquid. The consideration of this phenomenon is important in the design and operation of many hydraulic components. Cavitation is often accompanied by mostly unwanted effects such as noise, vibrations, efficiency losses and erosion.

The micro-diesel effect is another phenomenon, which can be caused by cavitation in hydraulic components. This process is defined by the self-ignition of a gas and vapour containing bubble. Thus, the micro-diesel effect is another load for hydraulic components, which can be caused by cavitation. A typical example of a micro-diesel based overload are scorched sealings. But not only components can be affected by this phenomenon, also the quality of oil will suffer under accelerated ageing. If the micro-diesel effect is ignored, all these described negative impacts can lead to additional costs for the operator of the hydraulic system. But this is the crucial challenge. So far, there is no easy way to predict the micro-diesel probability. You can’t find any tool within commercial CFD software to judge the probability of micro-diesel occurrence.

Taking a look into literature, there are only few contributions concerning the micro-diesel effect. Lipphardt /1/ has conducted basic micro-diesel experiments in the 1970s. He was investigating the conditions which are necessary to trigger the self-ignition of a single bubble in mineral oil. The required minimum pressure of rate increase for self-ignition of a single bubble in mineral oil. The required minimum rate of pressure increase of 110.000 bar/s is one main result of his work. The experiments were conducted under much idealised conditions, which makes the translation of the results into components like valves and pumps difficult. Ideal spherical bubbles like Lipphardt investigated are hardly to be found due to high velocities and turbulent conditions. Schmitz /2/ has taken up the results from Lipphardt and was developing a simulation model. This is useful for a better understanding of the experimental results. It shows for example, that smaller bubbles need higher rates of pressure increase for self-ignition. But a translation of the model to predict the micro-diesel effect within real components is also difficult, because the modelled bubbles are ideal spherical. Thus, some equations which were used are not valid for strongly deformed bubbles. The experiments of Loheertz /3/ were the only ones investigating the micro-diesel effect in components with inner flow. It is a very simple geometry of a 90 degree flow deflection. But for the development of a simulation model of the micro-diesel effect from the experiments, there is insufficient information available. The operating conditions to obtain first micro-diesel events and a detailed position of the emitted light are not mentioned explicitly. Thus, the results are also inappropriate for model development.

The present contribution takes up the mentioned weaknesses and will deliver detailed boundary conditions of the micro-diesel effect within models of real components. Therefore a typical valve and a pump geometry were chosen, where the occurrence of the micro-diesel effect was expected. During the micro-diesel experiments a further light-emitting phenomenon has been observed, which could not be assigned to the micro-diesel effect. This intensive blue light was induced by a gas ionisation within the analysed models. To reach this, among other conditions, a high voltage is necessary. This voltage is caused by an electrostatic charge in the model. Electrostatic charge is a current topic in the world of hydraulics, in particular in filter technology /4/. Improved hydraulic oils lead to high electrostatic potentials and following electrostatic discharges, which can destroy a filter. The present contribution shows recordings of different types of electrostatic discharges and will deliver operating conditions for such a discharge. These information can be used to develop models for an electrostatic discharge prediction in further works.

2 Basics of the micro-diesel effect, gas ionisation and electrostatic charge

2.1 The micro-diesel effect

The micro-diesel effect is defined by the self-ignition of a gas and vapour containing bubble. Therefore it is crucial that a gas bubble has to contain oxygen and flammable components. The oxygen is typically provided by air evolution within the cavitation process, which can take place at various points in a hydraulic system. Common hydraulic fluids are consisting of various flammable hydrocarbons, which makes a micro-diesel effect possible.

For the self-ignition of the bubble a high temperature is necessary. Depending on the chemical composition, hydraulic fluids typically have an ignition temperature between 300 and 400 °C /5/. This liquid-dependent ignition temperature can be reached by a rapid compression of the bubble. The therefore minimum necessary critical pressure increase can be estimated considering an adiabatic compression:

\[ \Delta p_{\text{crit}} = p_1 \left( \frac{5}{p_1 T_{\text{ign}}} \right)^{\kappa} - p_0 \]

The pressure \( p_0 \) and temperature \( T_0 \) are the conditions in the bubble at the beginning of compression. In case the pressure increase was too slow, the adiabatic assumption of equation (1) is not valid and the necessary temperature would not be reached. This is caused by a high heat transfer from the inner bubble into the liquid. Lipphardt has found a minimum rate of pressure increase of 110.000 bar/s to provoke an ignition of a bubble in mineral oil. This lower limit has been found for big bubbles with a volume of approximately 180 mm³. For smaller bubbles (< 100 mm³) higher rates are necessary. The translational velocity of the bubble in the flow has also an impact on the heat transfer. High slip velocities are increasing the heat transfer. The typical combustion period of a bubble in mineral oil is less than 2.5 ms /1/. Furthermore, the mixing ratio within the bubble and the ignition delay time determine whether or not the bubble will ignite.

2.2 Gas ionisation and electrostatic charge

Ionisation defines the process of forming an ion from an atom or molecule. Its negative or positive charge of these elementary particles will be achieved by losing or gaining electrons. The loss of an electron can be caused by a collision between elementary particles or through interaction with light. Further mechanism of ionisation are also possible. For the separation of an electron from an atom or a molecule a minimum energy is necessary. This ionisation potential depends on the type of atom (material) and the excited state. The separation of further
electrons from an atom is possible, but compared to the first separation, higher ionisation potentials are needed /6/.

The ionisation potential can be delivered in case of an ionisation by collision by the kinetic energy of the collision partner. This is typical for a field ionisation, whereby charged particles are accelerated by an electric field. Accelerated by the same field, electrons can reach a higher kinetic energy than positive ions due to their lower mass. Consequently, a collision with fast electrons is the dominant ionisation process in an electric fields. If the electric field energy was high enough, this type of ionisation would cause a chain reaction. The result of every field ionisation event is a further free electron, which can be accelerated and thus is able to ionise further atoms or molecules /6/.

To ensure the ionisation of a gas, a combination of a high electric field and a low gas pressure is necessary. At high pressures the distance between adjacent gas molecules is small. In the same electrical field a collision without enough kinetic energy has a higher probability than at low gas pressures. In this case the acceleration path of the electron is too short and the kinetic energy will be transferred to the atom/molecule by an elastic impact without ionisation.

Positive gas ions are not stable. They recombine with electrons and the previously required ionisation energy will be released by emitting photons. In gases different types of gas discharge are basically possible. They are mainly depending on gas pressure and electric field, but further conditions have an influence. At low pressures a glow discharge is most likely /7/. A glow discharge in air emits photons which are visible as blue light. Therefore for example in a geissler tube, an absolute pressure of 1 mbar and a voltage of some kV are necessary /6/. A combination of a lower voltage and smaller distance between the electrodes can also yield to a glow discharge. But according to Paschen’s law the minimum voltage for an air discharge is approximately 330 V. At higher pressures and corresponding higher voltages an arc discharge is more probable. This type of discharge appears as lightning.

The presented phenomena of glow discharge can also occur in hydraulic components. Two conditions can be typically found in every cavitation zone: free gas and a low pressure. The necessary high voltage can occur due to an electrostatic charge of the oil and the components. Modern hydraulic oils have a low electrical conductivity and therefore tend to charge electrostatically /8/. Critical are fluids with an electrical conductivity lower than 1000 pS/m. A charge of hydraulic components is also possible, if they consist of isolating materials or if there is no electrical grounding of conductive materials. In addition to the mentioned influencing material properties further factors determine about the level of electrostatic charge. The geometry and size of components as well as the flow velocity are influencing the charge. High flow velocities are typically critical, because the associated friction is the reason for electrostatic charge. In literature voltage measurements are documented, which verify the high electrostatic charge of oil passing a filter. Voltage peaks till 17 kV are the result of these measurements /4/. This voltage is high enough to trigger an air discharge and thus all necessary conditions for a gas discharge are possible within hydraulic systems.

3 Overview of the test rig as well as the investigated valve and pump models

The presented test rig and geometries originally were developed to investigate cavitation and cavitation erosion. The design was not optimized to provoke the phenomenon of micro-diesel or gas discharge. Both are peripheral phenomena, which could be repeatedly observed during cavitation measurements.

3.1 Geometries of spool valve and axial piston pump model

An overview of the investigated models is given in Figure 1. Both are two-dimensional models with a model depth of 10 mm. An optimum optical access for visualization is assured through acrylic glass windows, which border the flow chamber in the third dimension.

The spool valve model represents a single control edge, which can be found in every spool valve. Between inlet and outlet a constant throttle gap of 1 mm height is positioned. The piston representing element of the valve model is chamfered in the control area. At high volume flows and low outlet pressures a cavitation area arises in the valve chamber downstream from the gap. This area is not only interesting for the visualization of cavitation, but also for the presented phenomena of chapter 2.

The second model is a section of an axial piston pump, where a high cavitation risk at reversing between delivery and suction stroke is typically. Thereby, the control groove of the valve plate is the dominant geometry, which throttles the flow. The high pressure difference between valve plate kidney and cylinder cavity causes a cavitation jet into the cylinder cavity. This one is very intensive in the considered pump at 9 ° overlap. For that reason, the investigated plane geometry was derived in this moment. Details of the origin geometry and some cavitation measurements can be found in the contribution of Wustmann /9/.

![Spool valve model](image1)

![Axial piston pump model](image2)

Figure 1: Overview of the investigated models

3.2 Test rig

Both models can be investigated with the same test rig, which is viewed in Figure 2. A change between the models is easily possible by a replacement of a few parts in the plane basic model. The volume flow is delivered by two screw-spindle pumps, which are characterized by a very low pulsation. Two speed variable motors enable the control of the volume flow. The maximum volume flow rate is 90 l/min. A variable throttle valve in the return line realizes a variation of the back pressure. High back pressures at same pressure difference can reduce or eliminate cavitation. A heat exchanger is used to keep the oil temperature constant. Various sensors are installed to measure the volume flow, inlet and outlet pressures as well as several temperatures at different positions. The temperature sensors $T_1$ and $T_2$ are only part of the valve model. These are positioned in areas, where a high micro-diesel impact is expected.

A visualization of the light appearances is possible through the windows of the basic model. To record the light phenomena with high light sensitivity, a monochromatic high speed camera is used. Although the camera allows a maximum frame rate of 650,000 fps, the presented recordings were taken with only 24 fps. This allows a higher exposure duration of 42 ms, which permits the visualisation of weak lights. The maximum spatial resolution of the camera is 1280x800. An impact of extraneous lights at the recordings can be precluded, because the optical setup is installed in a dark room. All presented measurements have been conducted with the mineral oil “HLP 46” of the manufacturer ELASKON.
4 Experimental results

In this chapter examples of different light phenomena will be presented. These recordings are selected frames of a few measurements in both presented geometries. An assignment of each appearance to the phenomena of micro-diesel effect or gas discharge will be discussed. Furthermore, the operation conditions at the inception of the phenomena will be presented. This overview can be used in further works for a development of a prediction model of the different phenomena. Measurements of the near wall temperatures in close proximity to the phenomena are also a part of this chapter. These temperatures show the load of the real component, which is caused by these phenomena.

4.1 Micro-diesel effect and gas discharge within the valve model

In the valve model both phenomena, the micro-diesel effect and gas discharge, have been observed. Figure 3 shows two examples of the occurrence, which have a clear difference in position and intensity. During measurement 1 two areas of low light intensity have been observed. At stagnation point of the open jet was a stable diffuse light, which is identified as a gas discharge of air. The reason of the assignment to the gas discharge is the observed blue appearance of the phenomenon in this area (compare Figure 5). This blue light is typically for a glow discharge of air. In a second area of measurement 1 a further weak light could be observed in the single frame. This one cannot be clearly assigned to a single phenomenon. The continuity of the light points to a gas discharge, but a clear blue light at this position is hard to be found in the available color recordings (compare Figure 5). In contrast, an analysis of an image series shows a lot of very small single events in that area. This points to the micro-diesel effect and will be clarified later in this section.

In measurement 2 two very intensive lights could be observed in the middle of the free jet. These continuous lights can be clearly assigned to the gas discharge, because of the intensity, continuity and typical position. The reason for the big difference between the appearances of measurement 1 and 2 is not clear. Both measurements have nearly the same operating conditions and were conducted within 20 minutes. Thus, a change of influencing factors like fluid properties or atmospheric pressure is unlikely within this time. The only one obvious difference between the measurements was the different acceleration of the volume flow. In measurement 1 the volume flow was linear increased from 0 to 80 l/min within 52 s. During measurement 2 for the same increase only 22 s have passed. Thus, this acceleration difference could be the reason for the difference appearance of the phenomena, but further measurements are necessary to confirm this hypothesis.
A comparison of the determined operating conditions shows, that the micro-diesel effect starts at lower pressures than the gas discharge. This is the case at all three determined points of micro-diesel phenomena. Furthermore, there is a big difference between the necessary minimum pressure of gas discharge inception between measurement 1 and 2. Once again, the difference in the acceleration of the volume flow above mentioned could be the reason for this.

### 4.2 Temperature measurements in the valve model

Two temperature sensor have been installed in close proximity to the phenomena of measurement 1 to determine the temperature load of the component. Both sensors are a type T sheath thermocouple with a diameter of 1 mm. These sensors have been mounted flush to the wall. The mounting positions of the sensors T1 and T2 is shown in Figure 5. Contrary to the first measurements, the upper component of the valve model consists instead of copper, of acrylic glass. Because of its low thermodynamic conductivity, this component was chosen for the integration of the thermocouple. As visible in Figure 5, this change of material has an impact on the temperature load of the component. Both sensors are a type T sheath thermocouple with a diameter of 1 mm.

<table>
<thead>
<tr>
<th>Effect / Measurement</th>
<th>Inlet pressure $p_1$ [bar]</th>
<th>Outlet pressure $p_2$ [bar]</th>
<th>Volume flow $Q$ [l/min]</th>
<th>Temperature $T_1$ [°C]</th>
<th>Temperature $T_2$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Micro diesel @ A</td>
<td>9,1</td>
<td>1,3</td>
<td>20,6</td>
<td>21,9</td>
<td>22,0</td>
</tr>
<tr>
<td>Measurement 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Micro diesel @ B</td>
<td>12,4</td>
<td>1,4</td>
<td>25,1</td>
<td>21,9</td>
<td>22,0</td>
</tr>
<tr>
<td>Measurement 1</td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Micro diesel @ C</td>
<td>17,8</td>
<td>1,5</td>
<td>30,3</td>
<td>21,8</td>
<td>22,0</td>
</tr>
<tr>
<td>Measurement 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>Gas discharge</td>
<td>28,6</td>
<td>1,6</td>
<td>38,8</td>
<td>21,8</td>
<td>21,9</td>
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<tr>
<td>Measurement 1</td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas discharge</td>
<td>17,7</td>
<td>1,4</td>
<td>31,3</td>
<td>25,1</td>
<td>28,4</td>
</tr>
<tr>
<td>Measurement 2</td>
<td></td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>

**Table 1: Operating conditions at the first events of the different phenomena**

During further temperature measurements, the backpressure $p_2$ was increased at constant motor speed of both pumps. This pressure increase from 3 to 19 bar has provoked a reduction of volume flow of only about 4.5 %, as visible in Figure 7. This is caused by a simultaneous increase of the system pressure $p_1$. While the temperatures $T_1$ and $T_2$ are approximately unaffected, there is a striking decrease of the temperature $T_{D2}$ in the area of the micro-diesel. This behaviour cannot be explained by a reduction of viscous losses. The hydraulic energy, which is converted in thermal energy, is only reduced slightly from 14 to 12 kW.

The cavitation behaviour during this measurement can deliver an explanation for this significant temperature decrease. At the start of this measurement there is massive cavitation downstream of the gap. An increase of the backpressure at same volume flow reduces the cavitation intensity stepwise, till cavitation is completely eliminated. This cavitation free operating condition was reached after 9 s. This was optical reviewed, but can also be identified by the noiseless signal of backpressure $p_2$ in the period of maximum backpressure. Without cavitation, no compression of gas bubbles and accordingly no increase of the bubble temperature is possible. An explanation of the presented temperature decrease could be the loss of these bubble compressions. Since in the micro-diesel area many compression processes could be expected, these can lead to significant temperature changes in the whole area.

It should be mentioned that there is also highly likely an impact of acrylic components on measurements 1 and 2. During all measurements acrylic glass windows were used to get the necessary optical access. So it cannot be ruled out that the presented discharge phenomena are only induced by the design of the model.

The temperature profiles of measurement 3 are presented in Figure 6. An increase of the volume flow $Q$ is resulting in an increase of all temperatures, which are measured downstream of the gap ($T_{D1}$, $T_{D2}$, $T_3$). That was to be expected, because the throttle geometry converts hydraulic energy into thermal energy. The temperatures $T_{D1}$ and $T_{D2}$ are round 15 K higher than $T_3$, but this cannot be clearly referred to be a result of the presented phenomena. Due to a heat transfer between the mineral oil and the walls of the model, a lower temperature $T_3$ at the further downstream positioned sensor is also typical.

Furthermore, the maximum measured temperatures of $T_{D1}$ and $T_{D2}$ are too low to pose a risk for usual materials. The maximum measured temperature of 45 °C has a big difference to the ignition temperature of mineral oil, which is above 300 °C. But this measurement did not preclude the presence of the micro-diesel effect in the measurement areas. The response time of a thermocouple is several orders of magnitude higher than the duration of the burning process of the micro-diesel effect. Thus, the sensor cannot sense this fast process. Furthermore, the camera of the thermocouple is too big to sense the very small single event of an igniting bubble.

During further temperature measurements, the backpressure $p_2$ was increased at constant motor speed of both pumps. This pressure increase from 3 to 19 bar has provoked a reduction of volume flow of only about 4.5 %, as visible in Figure 7. This is caused by a simultaneous increase of the system pressure $p_1$. While the temperatures $T_1$ and $T_2$ are approximately unaffected, there is a striking decrease of the temperature $T_{D2}$ in the area of the micro-diesel. This behaviour cannot be explained by a reduction of viscous losses. The hydraulic energy, which is converted in thermal energy, is only reduced slightly from 14 to 12 kW.

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Because of the consistency of the lights, the high intensity and the relatively large size, the detected light-emitting phenomena are assigned to the glow discharge. This type of gas discharge is possible in areas of very low pressures, as introduced in chapter 2.2. Thus, the glow discharge delivers a field information about low pressure locations at the corresponding operating conditions. During the measurements no lights were detected, which points to the micro-diesel effect within the pump model.

Analogous to the procedure of the valve model, the operating conditions at first occurrence of the gas discharge have been analysed in the different areas of the pump model. As mentioned before, the volume flow was increased linearly during this measurement. The overview of this analysis is given in Table 2.

<table>
<thead>
<tr>
<th>Effect / Measurement</th>
<th>Inlet pressure $p_1$ [bar]</th>
<th>Outlet pressure $p_2$ [bar]</th>
<th>Volume flow $Q$ [l/min]</th>
<th>Temperature $T_1$ [°C]</th>
<th>Temperature $T_2$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas discharge @ 1</td>
<td>83,2</td>
<td>1,3</td>
<td>29,5</td>
<td>26,4</td>
<td>27,7</td>
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<td>139,5</td>
<td>1,7</td>
<td>42,0</td>
<td>26,5</td>
<td>29,3</td>
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<td>Gas discharge @ 3</td>
<td>154,8</td>
<td>1,7</td>
<td>47,0</td>
<td>27,2</td>
<td>32,0</td>
</tr>
</tbody>
</table>

Table 2: Operating conditions at the first events of gas discharge within the pump model

In comparison to the valve model, higher inlet pressures are necessary to provoke a gas discharge within the pump model. So far there are no further investigations to find the reason for this behavior. It is very probable, that the big deviation of the geometry has the greatest influence on it. Furthermore, the deviations in temperature, the history of the fluid (cavitation nuclei) and the lower acceleration of the volume flow can have an influence.

5 Summary and Conclusion

The present contribution deals with light-emitting phenomena in hydraulic components, which are closely linked to cavitation. The first one is the micro-diesel effect, which can be induced by a rapid compression of cavitation bubbles. The second investigated phenomenon was the gas discharge of air. Therefore, an electrostatic charge of the oil or of the component is necessary. Both light-emitting phenomena have been optically investigated within two plane models of hydraulic components. The first one was a model of a spool valve and the second one a model of the valve plate area of an axial piston pump.

An overview of the areas of occurrence and the minimum necessary operating conditions of the different phenomena is one result of this contribution. Some frames of the different phenomena are presented to show the form of appearance. Furthermore, the impact of electrically insulating materials on the appearance of gas discharge is shown. Another part of this contribution is the measurement of the temperatures in close proximity.
to the observed phenomena. Thereby, the temperature in the area of the micro-diesel has shown a striking course due to an increase of the back pressure.

This contribution can be used as basis for further investigations of the micro-diesel effect and of the gas discharge within hydraulic components. For one thing, this paper delivers detailed information for simulative investigation of the effects and secondly, some findings can be used as suggestion for further experimental investigations. The overall objective of the simulative investigations should be to enable a prediction of the phenomena. Even though an influence of the acrylic windows on the presented gas discharge cannot be ruled out, the problems of electrical discharge can become more important in the future. The reason for this is the increasing trend of lightweight design of hydraulic components /10/ and the associated usage of electrically insulating plastics.

6 Acknowledgements

The project “Extension of the operating conditions of hydrostatic components operated with water based hydraulic fluids” (Ref.-No. AiF 18491 BR/1) was financed and supervised by the Forschungskuratorium Maschinenbau e.V., Lyoner Str. 18, 60528 Frankfurt. In the scope of the Programme to promote Industrial Collective Research it was funded by the German Federation of Industrial Research Associations (AiF) with means of the Federal Ministry of Economic Affairs and Energy (BMWi) on the basis of a decision by the German Bundestag.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>$p_0$</td>
<td>Start Pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>$p_1, p_2$</td>
<td>Inlet and Outlet Pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>$T_0$</td>
<td>Start Temperature</td>
<td>[K]</td>
</tr>
<tr>
<td>$T_1, T_2$</td>
<td>Inlet and Outlet Temperature</td>
<td>[°C]</td>
</tr>
<tr>
<td>$T_{11}, T_{12}$</td>
<td>Temperatures in close proximity to the phenomena (positioning see Figure 5)</td>
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<tr>
<td>$T_{ign}$</td>
<td>Ignition Temperature</td>
<td>[K]</td>
</tr>
<tr>
<td>$Q$</td>
<td>Volume flow</td>
<td>[l/min]</td>
</tr>
<tr>
<td>$\Delta p_{crit}$</td>
<td>Critical pressure increase of self-ignition</td>
<td>[bar]</td>
</tr>
</tbody>
</table>

References


An Investigation of the Effects of Fluid Composition on Aeration, Efficiency, and Sound Generation in an Axial Piston Pump

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In this investigation, hydraulic fluids of varying base oil and additive composition were evaluated in a dynamometer fitted with a reservoir that incorporated an aerator at the inlet, and a mass flow meter at the outlet. The effects of aeration on piston pump efficiency and air borne noise generation were evaluated. Hydraulic oils that entrained a greater volume of air demonstrated lower volumetric efficiencies and higher sound levels. The fluids differed in volumetric efficiency by as much as 8% and perceived sound level by as much as 50%. Based upon 2,500+ hours of testing in a high-intensity loader application, the performance benefits of the low aeration fluid were persistent.

Keywords: Fluid properties, air release, density method, sound analysis

Target audience: Mobile Hydraulics

1 Introduction
Reducing the volume of fluid in a hydraulic reservoir decreases the fluid residence time and increases susceptibility to aeration brought about by machine vibration, operation on slopes and flow surges from retracting cylinders. Aeration of the hydraulic fluid is undesirable because it impacts the efficiency and responsiveness of the hydraulic system. Aeration can also increase machine noise levels and susceptibility to cavitation damage. Cavitation is the dynamic process of gas cavity growth and collapse in a liquid /1,2/. Two forms of cavitation are generally recognized; vaporous and gaseous cavitation. Vaporous cavitation occurs when localized pressures in the hydraulic system drop below the vapor pressure of the fluid, causing the liquid to boil. Gaseous cavitation occurs when localized pressures in a hydraulic system drop below the saturation pressure of air in oil, causing the formation of bubbles in the fluid. Cavitation can cause significant surface damage in hydraulic components as shown in Figure 1. Hence avoidance of aeration is an important consideration in hydraulic fluid formulation and system design.

Aeration affects the suction dynamics of positive displacement piston pumps. /3/ As described by Harris /4/, when a pump opens to the suction port at top dead center, fluid within the chamber rapidly depressurizes. The resulting momentum from decompression causes pressure to oscillate, inducing air-release, cavitation and flow ripple, which in turn gives rise to fluid- and air-borne noise.

Because dynamic simulations of hydraulic components and systems requires an accurate accounting of fluid momentum, a variety of models have been proposed for describing the effects pressure on volume, density and gas evolution in hydraulic fluids. These models vary in complexity from basic Henry’s Law expressions that assume instantaneous air release to full cavitation models that incorporate separate time constants for air adsorption (compression) and release (decompression). /4-7/ In this study, the steady-state effects of aeration on pump performance for fluids of different base oil and additive composition were examined. The percentage volume fraction of entrained air (ae) in the fluid was determined using the general rule of mixtures as shown in Equation (1) where (ρ) is the density of the mixture, (ρg) is the density of the gas (air) and (ρl) is the density of the liquid. Expansion due to vacuum conditions at the pump inlet was estimated from the ratio of atmospheric (ρatm) to inlet pressure (ρin).

\[ a_e = \left( \frac{\rho - \rho_l}{\rho_l - \rho_g} \right) \left( \frac{\rho_{atm}}{\rho_{in}} \right) \times 100 \]  

Prior research has examined the effect of reservoir design on fluid aeration. Longhitano /8/ evaluated air release in reservoirs using an optical sensor to quantify and size air bubbles. The bubbles were less than 200µm in diameter. Internal tank design affected the air release process. Release was enhanced by maximizing the distance between the suction and the return lines as well as the strategic placement of baffle plates.

Wohlers /9/ examined heat-rejection and air release in hydraulic reservoirs that were fitted with a return-line filter. Air bubbles of varying diameter were injected at the pump inlet. A mass flow meter was used to quantify the inlet air flow rate. Aeration levels were determined using a temperature-compensated capacitive sensor. A multi-phase bubble flow simulation was used to evaluate the de-aeration performance of the filter-tank systems. The simulation correlated the number of bubbles de-aerated inside the tank to the number of bubbles injected. Bubble size was identified as important factor in de-aeration performance; as the bubble size decreased, filter-tank system optimization became less effective. The heat rejection rates of prototype reservoirs were measured in a wind-tunnel. The initial fluid temperature was 60°C and the terminal temperature was 30°C. The heat rejection model proved quite accurate. The results showed that the convective cooling afforded by the reservoirs was minimal; it took in excess of 150 minutes for the temperature to decrease 30°C. This is an important finding because the implication is that reducing the size and convective cooling capacity of the reservoir does not significantly increase the heat load on the cooling system.

Vollmer /10/ proposed a four-stage methodology for optimizing the air release performance of reservoirs. Initial constraints were established based upon available space, return and suction line positioning, flow rate, air fraction and reservoir inclination angles. Single-phase computational fluid dynamics (CFD) was used to estimate the fluid residence time. Baffles and inclined wire-gauzes were added to the model to achieve the desired streamlines. Once the internal streamlines were optimized, the two-phase CFD simulations were performed. Stokes law was used to predict bubble velocity with a mean bubble diameter of 1mm. As shown in Equation (2), the terminal bubble velocity (Vt) increases as a square of the diameter (d) and decreases linearly with the dynamic viscosity(µ) of the liquid. /11/

\[ V_t = \frac{1}{18} \frac{g d^2 (\rho_l - \rho_g)}{\mu} \]  

Viscosity, density, and bubble diameter are fundamentally affected by the composition of the hydraulic fluid. In this paper, we investigate the effects of hydraulic oil additive and base oil composition on aeration, pump efficiency, and noise generation. Fast air release times are required for compact fluid power systems.
2 Materials and Methods

2.1 Test Fluids

Five hydraulic fluids were examined in this study. Fluid “A” was a Group I mineral oil, fluid “B” was a polyalphaolefin (PAO) based Group IV synthetic, fluid “C” was a Gas-to-Liquid (GTL) Group III based synthetic, fluid “D” was a Group II mineral oil and fluid “E” was a GTL Group III based experimental synthetic hydraulic formulation. The properties of the test fluids are shown in Table 1. The air release properties of the fluid were evaluated in accordance with ASTM Standard Test Method D3427. In the D3427 procedure, compressed air is blown through test oil that has been equilibrated to 25, 50 or 75°C in a jacketed bath. /12/ The overall density of the fluid decreases with the presence of entrained air. The time required for finely dispersed air in the oil to decrease such that the density of the mixture is 99.8% of the original density of the fluid is measured as the air release time. Fluids with a viscosity range between 9.0 mm²/s and 90 mm²/s are evaluated at a bath temperature of 50°C. The upper limit for ISO VG 46 hydraulic fluid at 50°C is 10 minutes. /13/ Thus all the fluids (including fluid “E”) which exceeds the viscosity limit for ISO VG 46) complied with industry standards for air release.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base Oil Type</td>
<td>Group I</td>
<td>Group IV</td>
<td>Group III</td>
<td>Group II</td>
<td>Group III</td>
</tr>
<tr>
<td>Air release, minutes (ASTM D3427)</td>
<td>5.03</td>
<td>0.17</td>
<td>0.17</td>
<td>1.51</td>
<td>6.03</td>
</tr>
<tr>
<td>KV40°C, mm²/s (ASTM D445)</td>
<td>45.63</td>
<td>46.02</td>
<td>43.37</td>
<td>44.32</td>
<td>54.13</td>
</tr>
<tr>
<td>KV100°C, mm²/s (ASTM D445)</td>
<td>6.721</td>
<td>7.795</td>
<td>7.532</td>
<td>6.825</td>
<td>10.86</td>
</tr>
<tr>
<td>Viscosity Index (ASTM D445)</td>
<td>100</td>
<td>139</td>
<td>141</td>
<td>109</td>
<td>197</td>
</tr>
<tr>
<td>Density, g/cc @20°C (ASTM D4052)</td>
<td>0.8718</td>
<td>0.8310</td>
<td>0.8266</td>
<td>0.8639</td>
<td>0.8332</td>
</tr>
<tr>
<td>Density, g/cc @50°C (Inline densitometer)</td>
<td>0.8515</td>
<td>0.8120</td>
<td>0.8047</td>
<td>0.8419</td>
<td>0.8097</td>
</tr>
</tbody>
</table>

Table 1: Fluid Properties

2.2 Fluid Field Testing

Test fluids “A” and “B” were used in a Komatsu Loader operated at a gold and silver mine located in southern California. The mine employs conventional open pit mining methods and operates 24 hours per day, 365 days per year with a life expectancy of 11 years. The machines were operated at high kinetic intensity and accumulated more than 2,500 hours of service in less than 6 months. Fluid samples were collected during the field trial. Air release time, foaming tendency and elemental analysis were performed on the samples. At the end of the field trail the fluid was transferred into 400 litre drums and returned to the laboratory for dynamometer testing.

2.3 Test Methods

The fluids were evaluated using a modified ISO 4409-2007 procedure. /14/ This standard test method prescribes the hydraulic circuit configuration, instrumentation, operating conditions, and test procedures required for evaluating the steady-state performance of hydraulic pumps, motors and integral transmissions. Normally this test method is used for rating the performance of hydraulic components. In this study, test fluids were evaluated and the hydraulic components were held constant in order to compare the susceptibility of fluids to aeration and the resulting impact upon pump efficiency and noise generation.

A Bosch- Rexroth A10VSO pump was used in this study. The A10VSO is an open loop axial piston pump that is commonly used in off-highway, construction and material handling equipment. The test pump differed from the common commercial machine in that it had high-precision electronic pump displacement control. The power to}

![Figure 2: Simplified circuit schematic showing the air-injection system.](image)

The pump outlet pressure, inlet oil temperature, rotational frequency and swashplate angle were controlled using a Gantner Automation system. The outlet pressure was 207 Bar (3000 psi), the fluid temperature was 50°C (122°F) and the pump rotational frequencies were 1200, 1700 and 2200 rpm. The swashplate angle was set at full displacement. The pump was enclosed in a chamber lined with foam-type acoustic insulation and the pump sound emissions were measured with a Larson-Davis 831 meter. Air was dispersed into the hydraulic fluid within the return line of the circuit using a custom diffuser. A mass flow controller supplied air at a flow rate of 18.9 standard liters per minute. Fluid density, pump flow rate, torque and sound levels were recorded with the aerator on and off for 60s intervals to assess the air release behaviour of each fluid. The test sequence consisted of twelve different combinations of speed, pressure and aeration states. Each sequence was evaluated a minimum of three times and more than 150,000 data points were collected. Large data sets were required to ensure that the air ingress and release rates had reached equilibrium and pump performance was steady state. Pressure sensors, flow meters, thermocouples and the torque transducer were selected to maintain ISO 4409 Permissible Variation Class A performance. Instrument output was collected using an IP Motion data acquisition system. The instrument list is shown in Table 2.

![Diagram](image)
The density of the fluid exhibits periodic changes corresponding to the temperature fluctuations that occur as the flow of water to the heat exchanger turns on and off. The oil temperature exhibited a periodic variation. This was due to cyclic flow of the cooling water through the heat exchanger. As the heat exchanger cycled on and off, the fluid experienced thermal expansion and contraction. The fluctuation in density of the non-aerated fluid mirrored the pump inlet temperature, indicating that the density sensor had sufficient bandwidth to detect thermal expansion and contraction. Thermal cycling did not impart periodic changes in density when the fluid was aerated the addition of air caused an increase in the variability. The amount of air in the hydraulic fluids was determined from the mean density measurements using Equation (1). The 95% confidence interval for the mean free air content (v/v %) at an outlet pressure of 207 Bar and a rotational frequency of 2200 rpm are shown in Fig. 5. The mean aeration level was less than 0.5% volume with the aerator was off.

The pump efficiency was calculated based upon the definitions listed in ISO 4391. Volumetric efficiency ($\eta_v$) was determined by dividing the actual pump flow rate ($Q_a$) by the theoretical pump flow rate ($Q_o$). The theoretical pump flow rate is defined as the pump displacement per revolution times the rotational frequency of the pump ($N$) as shown in Equation (3). Displacement ($V$) was determined from the maximum ratio of pump outlet flow rate to rotational frequency under steady-state conditions.

$$\eta_v = \frac{Q_a}{Q_o}$$

The pump overall efficiency was determined by dividing the pump output power ($P_o$) of the pump by the input power ($P_i$) of the pump as shown in Equation (4) where ($Q_o$) is the actual pump flow rate, ($p$) is the pump outlet pressure, ($N$) is the rotational frequency and ($T$) is the torque.

$$\eta_o = \frac{P_o}{P_i} = \frac{Q_o \cdot p}{N \cdot T}$$

Hydro-mechanical efficiency ($\eta_{hm}$) was derived from the volumetric and overall efficiency using Equation (5).

$$\eta_{hm} = \frac{\eta_v \cdot \eta_o}{\eta_v}$$

### Table 2: Published maximum ratings and errors for dynamometer sensors

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Sensor</th>
<th>Rating, max.</th>
<th>Error, max.</th>
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</thead>
<tbody>
<tr>
<td>Inlet mass flow</td>
<td>Micro Motion CMF200M</td>
<td>87,100 kg/hr</td>
<td>±0.05%</td>
</tr>
<tr>
<td>Outlet volume flow</td>
<td>Max Machinery G240</td>
<td>240 L/min</td>
<td>±0.3%</td>
</tr>
<tr>
<td>Case drain flow</td>
<td>Max Machinery G045</td>
<td>45 L/min</td>
<td>±0.3%</td>
</tr>
<tr>
<td>Pressure inlet</td>
<td>GEMS</td>
<td>7 bar</td>
<td>±0.25%</td>
</tr>
<tr>
<td>Pressure outlet</td>
<td>GP50</td>
<td>350 bar</td>
<td>±0.5%</td>
</tr>
<tr>
<td>Shaft speed</td>
<td>BEI Encoder HS35F</td>
<td>6000 rpm</td>
<td>±0.05%</td>
</tr>
<tr>
<td>Temp inlet</td>
<td>OMEGA Type J</td>
<td>220 °C</td>
<td>±1.1°C</td>
</tr>
<tr>
<td>Torque input</td>
<td>HBM T40B</td>
<td>500 N-m</td>
<td>±0.03%</td>
</tr>
<tr>
<td>Sound level</td>
<td>Larson Davis 831</td>
<td>140 dB</td>
<td>±0.1 dB</td>
</tr>
<tr>
<td>Air mass flow</td>
<td>Aalborg GFC 37</td>
<td>20 L/min</td>
<td>±1.5%</td>
</tr>
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</table>

3.1 Aeration

Time-series plots of density and temperature are shown in Figure 3. In this example, the pump was operating at an outlet pressure of 207 Bar and a rotational frequency of 2200 RPM. The density of the fluid was lower when the fluid was aerated (aerator on). At the beginning of the transition period between aerator states, the density of the fluid increased or decreased depending upon the direction of the transition. In order to ensure that the system was in a steady-state condition, the first 180s of data was excluded from the statistical evaluation of pump performance.
surfactants inhibit coalescence since they produce similar surface tension gradient forces in the surface that are responsible for stabilizing Marangoni flows or immobilizing the gas-liquid interface. /17, 18/ As shown in Table 3, Group I base oils have higher sulfur content, lower saturates content and lower viscosity index than Group III base oils. /19/ GTL Group III and PAO Group IV base oils contain ≥99.9% saturated hydrocarbons. These base stocks coalesce bubble rapidly because the produce large bubbles with a narrow size distribution. In comparison, Group I base oils of the same viscosity stabilize a broader bubble diameter distribution of smaller bubble sizes, resulting in higher air release times. It is however important to acknowledge that even with high saturate content base oils, it is possible to extend air release times through an unjustified selection of additives. The air entrainment level of fluid “E” was 5-times higher than fluid “B” even though both fluids were formulated with Group III base stocks. The difference can be primarily ascribed to additive chemistry, especially silicone based anti-foams and certain viscosity modifiers.

<table>
<thead>
<tr>
<th>Base Oil Category</th>
<th>Sulfur (wt%)</th>
<th>Saturates (wt%)</th>
<th>Viscosity Index (VI)</th>
<th>Fluids</th>
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<tr>
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<td>≤0.03</td>
<td>≥90</td>
<td>80-120</td>
<td>A</td>
</tr>
<tr>
<td>Group II</td>
<td>≤0.03</td>
<td>≥90</td>
<td>80-120</td>
<td>D</td>
</tr>
<tr>
<td>Group III</td>
<td>≤0.03</td>
<td>≥90</td>
<td>≥120</td>
<td>C &amp; E</td>
</tr>
<tr>
<td>Group IV</td>
<td>All Polyalphaolefins (PAOs)</td>
<td></td>
<td></td>
<td>B</td>
</tr>
<tr>
<td>Group V</td>
<td>All Others Not Included in Groups I-IV</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3: Base oil categories per the API Engine Oil Licensing and Certification System. /19/

3.2 Efficiency

The volumetric efficiency of the pump was determined from the volumetric flow rate using Equation (3). As shown in Figure 5, the average volumetric efficiency of the pump was approximately 93% when the aerator was off. The volumetric efficiency of the pump decreased by 9.4% for fluid “A” and 7.5% for fluid “E” when the aerator was on. Fluids “B” & “C” changed by 2%. A comparison of Figures 4 and 5 reveals that decrease in volumetric efficiency when the aerator was on (2.0 to 9.4%) was greater than the volume of air measured in the fluids (0.91 to 5.36%). It is theorized that air volume expansion at the pump inlet increased the apparent effect of aeration on volumetric efficiency.

Figure 5: Aeration of the fluid decreased pump volumetric efficiency and increased case drain flow losses.

As shown in the graph on the right side of Figure 5, air entrainment also affected the case drain flow losses of the pump. Entrained air can reduce the density and the viscosity of lubricants. Determining the relative impact of viscosity and density under two-phase flow conditions is difficult. Regardless, Figures 5 provides evidence that aeration increased both compressibility and case drain flow losses.

The overall efficiency of the pump was determined from torque, speed, pressure and flow measurements using Equation (4). As shown in Figure 6, the average overall efficiency of the pump was approximately 86% when the aerator was off. The overall efficiency of the pump decreased by approximately 7% for fluid “A” and 6% for fluid “E” when the fluids were aerated. The overall efficiency of fluids “B” and “C” changed by approximately 1%. The decrease in overall efficiency brought about by fluid aeration was comparable to the decrease in volumetric efficiency.

Figure 6: Aeration decreased overall and increased hydromechanical efficiency.

The hydromechanical efficiency values shown in Figure 6 were determined using equation (5). In physical terms, the hydromechanical efficiency of a pump is largely a function of the pump displacement, outlet pressure and viscous drag. When the fluid was aerated, the effective displacement of the pump was reduced as evidenced by a decrease in the pump volumetric efficiency. The reduction in effective displacement also lowered the amount of torque required to rotate the pump. Furthermore, aeration reduced the effective viscosity of the fluid. Thus an apparent increase in hydromechanical efficiency was evident when the fluid was aerated. Since aeration of the fluid decreased volumetric efficiency more than it increased mechanical efficiency, a net decrease in overall efficiency was observed when the fluid was aerated.

3.3 Sound Measurements

Aeration of the fluid caused cavitation and contributed to broad band noise generation by the pump. The equivalent A-weighted steady-state sound level of the pump was measured in 60s intervals at a distance of 10 cm with and without aeration. As shown in Figure 7, aeration increased the pump sound level in all 5 fluids. Fluids “A” and “E” were the most susceptible to aeration and exhibited the largest increase relative to its baseline value. The mean sound level of fluid “A” was approximately 6 dB(A) greater than that of the other fluids, which corresponds to 50% louder in terms of human perception.

Figure 7: Noise generation by pump with and without aeration.
Sound-level measurements are useful for comparing noise intensity but they do not contain frequency information. The frequencies (Hz) of sound vibration are important because the human auditory system functions like a bandpass filter. As a result, critical bands at roughly 1/3 octave intervals determine the psychoacoustic effects of noise. /20/ One-third (1/3) octave analysis was used to identify the range of frequencies responsible for the higher sound level generated by aerated fluid. Audio spectra were digitally filtered and divided into 36 frequency bands centered around 1,000 Hz ranging from 6.3 to 20,000 Hz. As shown in Figure 7, sound level at 3150 Hz increased from 85 to 95 dB(C) when the aerator was turned on. This increase in the high frequency sound level accounts for the harsh noise associated with pump cavitation.

3.4 Fluid Field Testing

Samples of hydraulic fluids “A” and “B” were collected from two Komatsu Wheel Loaders during a field trial. Foam tendency/stability, air release time, neutralization number, emission spectroscopy and other routine oil analysis tests were performed. The foaming tendency was measured using the ASTM D892 Standard Test Method for Foaming Characteristics of Lubricating Oils. /21/ The D892 test is conducted in three sequences. In Sequence I, the oil sample is equilibrated at 24°C. Air is bubbled through oil for 5 minutes to measure foaming tendency. The oil is allowed to settle for 10 minutes to measure foam stability. In Sequence II, the fluid is equilibrated at 93.5°C and the foam tendency/stability is measured as described above. In Sequence III the fluid temperature is returned to 24°C and the tendency/stability test is repeated. Various levels of foaming tendency are permitted by industry standards, but stable foam is generally not tolerated. Neither fluid “A” nor “B” generated stable foam in the field trial after 2,500 hours of service. However, the foaming tendency of fluid “A” exhibited considerable fluctuation as shown in Figure 8.

As shown in Figure 9, the air release time for fluid “A” changed relatively little throughout the course of testing. The mean air release time for fluid “B” increased relative to new fluid but remained lower than that of “A”. Given the fluids were in service for in excess of 2,500 hours, the consistency of the air release characteristics was unanticipated.

3.5 Dynamometer Testing of Fluids from the Field Trial

At the end of the field trial the fluid was transferred into 400 litre drums and returned for dynamometer testing. Susceptibility to air entrainment and the impact of aeration upon pump performance were evaluated using the same methods that were used to test the new fluids. As can be seen in Figure 10, the volume of air entrained by the used fluids was higher than that of the new fluids. Air entrainment increased by 1.2% in fluid “A” and 1.5% in fluid “B”. Nonetheless, fluid “B” remained twice as effective at releasing air.

Aging of the fluid did not cause a deterioration in pump volumetric efficiency when the aerator was off. The volumetric efficiency of the pump remained around 93%. However, aging of the fluid reduced volumetric efficiency of “A-used” by 1.5% and “B-used” by 3% when the aerator was on. Thus, dynamometer testing indicates that aerated fluid “B” was still more efficient than fluid aerated “A” by approximately 4% after 2,500 hours of service.

4 Summary and Conclusions

A methodology for studying aeration of hydraulic fluids and the effects of entrained air on pump efficiency and sound generation was presented. Five fluids of varying base oil and additive composition were compared.

- Fluids “B” and “C” demonstrated fast air release times in the ASTM D3427 test method and low levels of aeration in dynamometer testing. Both formulations used synthetic base stocks that contained greater than 99.9% saturated hydrocarbons.
- Fluids “A”, “D” and “E” complied with the limits for the ASTM D3427 air release test as defined in ASTM D6158. However, these fluids exhibited high levels of aeration in dynamometer testing. Fluids “A”, “D” and “E” used conventional and high-purity synthetic base stocks.
- The average volumetric efficiency of the pump was similar for all fluids when the aerator was off. When the aerator was on, the average volumetric efficiency for fluids “A”, “D” and “E” decreased by 9.4%, 4.4% and 7.5% respectively. The average volumetric efficiency fluids “B” and “C” decreased by 2.0%. Compressibility losses alone did not account for the observed decrease in volumetric efficiency. Increased pump case leakage was observed with the highly aerated fluids.
• The hydromechanical efficiency of the pump slightly improved when the fluid was aerated. The increase in mechanical efficiency was less than the decrease in volumetric efficiency brought about by aeration. Consequently, aeration reduced the overall efficiency of the pump.
• The mean sound level of the pump increased by 7.8 dB(A) for fluids “A” and “E” when the aerator was on. In terms of human perception, the pump was more than 50% louder. Aeration increased broadband high frequency sound levels between 2500 and 4000 Hz, which accounts for the harsh noise associated with pump cavitation.
• Based upon benchtop testing of field samples, the foaming and air release performance of fluids “A” and “B” decreased slightly during 2,500+ hours of high-intensity use in a front-end loader. Dynamometer testing of the used fluids confirmed these results. Nonetheless, fluid “B” retained a 5% advantage in volumetric efficiency relative to fluid “A” when aerated.

These results demonstrate the importance of additive and base oil selection in the formulation of hydraulic fluids for quiet, compact and efficient fluid power systems.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>Leq(A)</td>
<td>A-weighted sound level decibels</td>
<td>[dB(A)]</td>
</tr>
<tr>
<td>Leq(C)</td>
<td>C-weighted sound level decibels</td>
<td>[dB(C)]</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
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References

Influence of transient effects on the behaviour of translational hydraulic seals

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In common practice a hydraulic cylinder undergoes permanent acceleration and deceleration. In general this transient behaviour is neglected in the simulation of hydraulic seals, especially regarding the fluid film where stationary conditions are assumed. In order to gain a detailed understanding of the dynamic sealing process, a finite element based, elastohydrodynamic simulation model for hydraulic seals has been developed, including transient effects /1/. In this paper the influence of these transient effects on the behaviour of a hydraulic seal is investigated. The influence is studied under different system conditions in order to examine to which extend the consideration of transient effects in a simulation of hydraulic seals is inevitable.

Keywords: Hydraulic Seals, Transient Behaviour, Friction, FE-Simulation, Breakaway Force

Target audience: Tribology, Components, Design Process

1 Introduction

Seals are crucial machine elements for example in hydraulic cylinders. However, especially in regard to dynamic seals, the theoretical understanding of the sealing mechanism is still insufficient. Besides experiments, a physically based simulation model can increase the understanding of the sealing mechanism. In common practice the concept of Blok is often used to calculate the sealing process /2/. Here the fluid pressure is assumed to equal the solid contact pressure (that is provided, e.g., by a FE-calculation). Based on the fluid pressure the gap height (or fluid film thickness) in the contact regime is calculated with the (inverse) Reynolds equation. Using this method any solid contact, that will occur at least for very low velocities, is neglected. Thus, the method of Blok is not applicable for the purpose of this study.

A second approach is a direct combination of the fluid and solid contact calculation. For a certain height the solid and the fluid pressures are calculated. The pressures lead to a deformation of the seal and thus to a changing gap height. In this elastohydrodynamic lubrication (EHL) approach the computational effort is much higher but solid and fluid contact interaction can be considered. A comparison of both methods is, e.g., presented by Wohlers /3/.

In order to gain a detailed understanding of the dynamic sealing process, a finite element based, EHL simulation model for hydraulic seals was developed. The simulation is embedded in the software ABAQUS. Contact and fluid calculations are physically motivated and include transient effects. Aim of this study is to examine to which extend the consideration of those transient effects is inevitable for an exemplary contact problem of an O-ring seal sliding on a hard, randomly rough surface.

This paper is structured as follows: In the next chapter the simulation model is briefly described. Chapter 3 provides a detailed investigation of three different transient effects. The simulation results are compared to experimental measurements in chapter 4. Finally a summary and conclusion is given in chapter 5.

2 Simulation model

In this chapter the simulation model is briefly described. Details of the simulation method are provided in /1/.

In this simulation an O-ring seal cord (diameter: 5 mm, length: 40 mm) is squeezed into contact with a rigid counter surface (Rod), as illustrated in Figure 1. The rod’s velocity is linearly increased (acceleration = 3 mm/s²) at a constant system temperature of 20 °C. The boundary surface is fixed in x-direction. Solid and fluid contact forces are calculated at the seal surface.

![Figure 1: Illustration of the simulated sealing contact /1/](Image)

The structure of the simulation model is shown in Figure 2. The contact and fluid interaction as well as the resulting friction forces are directly implemented in the FE-Software ABAQUS using the concept of user subroutines /4/. Thus the interdependency of contact forces and deformation of the seal is considered in every single calculation step.

![Figure 2: Structure of the EHD-simulation model /1/](Image)
Influences the fluid pressure build-up. Therefore, in this study three major transient influences on the behaviour of translational hydraulic seals are investigated:

- Influence of the actual relative velocity in the contact
- Influence of the non-constant coefficient of solid friction
- Influence of the time dependent change of the sealing height

For all comparisons a benchmark simulation is defined: The coefficient of solid friction is constant (μ = 0.8). For the calculation of the fluid film the rod velocity is chosen (vRod). Also the stationary Reynolds equation (4) is applied for the calculation of the fluid film:

$$\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) = \frac{\partial}{\partial x} \left( h v_x \right)$$

3.1 Influence of the actual relative velocity

Only the pre-breakaway phase (1) is influenced by the chosen velocity. In this phase it affects the fluid pressure build-up and therefore the sealing gap height, as shown in Figure 4.

Due to the lower relative velocity, the fluid pressure and the gap height are decreased. Even though this phase is dominated by solid friction, this results in a slightly higher breakaway friction force, as illustrated in Figure 5.

Figure 4: Fluid pressure and gap height in the sealing gap. Pressure and height are influenced by the chosen velocity: piston rod velocity (vRod) or actual relative velocity (vRel).

Due to the lower relative velocity, the fluid pressure and the gap height are decreased. Even though this phase is dominated by solid friction, this results in a slightly higher breakaway friction force, as illustrated in Figure 5.
3.2 Influence of the non-constant coefficient of solid friction

The simulation model for the solid friction is based on the simple Coulomb friction model. For the implementation in the FE model, the subroutine UINTER is used. Here the (normal and tangential) stresses $\sigma$ between the contacting surfaces are defined for every contacting node $i$. For this comparison two different versions for the calculation of the tangential stress are implemented:

For sealing simulations in general Amonton’s law of friction is applied (version A). The tangential stress $\sigma^\tau$ is directly proportional to the applied normal stress $\sigma^N$ (Equation 5 A). The proportionality factor (or coefficient of friction) $\mu$ is constant in this version. $\delta$ is the contact slip.

$$\sigma^\tau = \text{sign} (\delta) \cdot \sigma^N \cdot \mu$$

(5 A)

In reality this assumption of a constant coefficient of friction does not hold for the contact between rubber and a hard rough surface. In fact, the tangential stress in the contact is proportional to the area of real contact $A^\text{real}$. The area of real contact depends on the contact pressure, as illustrated in Figure 6. For small contact pressures, real contact only occurs at few asperities. For rubber contacts the ratio of the real area of contact in regard to the apparent area of contact grows nonlinear. The actual correlation (Figure 6 – right) is calculated with Perssons model of contact mechanics /5/. It is influenced, e.g., by the surface of the contact partners and their material parameters.

In the previously described model the idle time dependent influence on the area of real contact are not significant. At higher contact pressures (e.g. for pressurised rod seals) the influence is assumed to be considerable. The consideration of an increased area of real contact due to idle time leads to an increased critical breakaway force (equation 8):

$$\tau^\text{crit idle} = \tau^\text{crit real} \cdot \frac{A^\text{real}}{A^\text{apparent}}$$

(6 B)

The amount of local slip until the critical shear stress is reached, is described by a critical tangential slip $\delta^\text{crit}$ /11/. Thus, in this initial pre-breakaway phase, the contact stress can be described with equation 7 for both models.

$$\sigma^\tau = \tau^\text{crit} \cdot \delta^\text{crit}$$

(7)

For the comparison, the coefficient of friction ($\mu = 0.8$) is chosen to match the breakaway force of the physical calculation based on the shear stress in the contact for a normal load of 10 N.

In Figure 7 the calculated friction force is shown for two different normal loads. For the applied system conditions, the influence is negligible. Only a small deviation is visible for the normal load of 31.1 N. For the low contact pressures in this application and their little variation (0.6 MPa - 1.2 MPa) the correlation of contact pressure and area of real contact is approximately linear. Thus, the assumption of a constant $\mu$ is applicable here.

Amonton’s law only holds for a linear correlation of contact pressure and area of real contact. Thus, a second version (version B) of the solid contact friction calculation is implemented:

$$\sigma^\tau = \text{sign} (\delta) \cdot \tau^\text{crit real} \cdot \frac{A^\text{real}}{A^\text{apparent}}$$

(5 B)

The contact shear stress in the area of real contact $\tau^\text{crit real}$ is assumed to be constant in this study. A constant value of $\tau^\text{crit real} = 3.5$ Mpa is assumed. In reality it varies with temperature and relative velocity. According to the model of Coulomb, a critical contact shear stress $\tau^\text{crit}$ has to be exceeded before macroscopic slip in the contact region occurs. As long as $\tau^\text{crit}$ is not exceeded only microscopic slip occurs. In the simulation, this critical contact shear stress $\tau^\text{crit}$ is calculated for each node $i$ according to equations 6 A or 6 B, respectively.

$$\tau^\text{crit} = \sigma^\tau \cdot \mu$$

(6 A)
Figure 8: Friction force as a function of the rod velocity for two different normal loads. In the contact stress based calculation (version B) an idle factor $\kappa_{\text{idle}} = 1.2$ is applied.

3.3 Influence of the time dependent change of the sealing height

Especially in phase 2, the sealing height changes rapidly. In Figure 9 the sealing height and the velocity normal to the contact is plotted for two instants of time just before and directly after breakaway occurred.

Before breakaway, the seal is dragged with the moving piston rod, resulting in a negative velocity in the inlet zone of the sealing gap. At breakaway the seal flips in the opposite direction, resulting in a change of sign. For the two marked positions (I) and (II) the time dependent change of fluid pressure and $y$-velocity is plotted in Figure 9. Two different simulations are compared. The stationary (stat) model is based on the stationary Reynolds equation (equation 4) while in the transient (trans) simulation the time dependent change of the gap height is considered (equation 2).

In Figure 10, the influence of the transient term in the Reynolds equation is shown for the minimum sealing gap height and the mean fluid pressure. It is especially apparent in the oscillation of the minimum sealing height during breakaway ($t = 0.5 - 1.0 \text{ s}$).

In Figure 11, the minimum sealing gap height and mean fluid pressure as a function of time are plotted for the stationary and the transient calculation. The comparison shows a lower fluid pressure and a smaller sealing gap when the transient term of the Reynolds equation is considered. The increasing height during acceleration leads to a considerable decrease of the fluid pressure.

Figure 10: Velocity normal to the contact and fluid pressure as a function of time for two different positions in the inlet (I) and the outlet zone (II) of the sealing gap.

Figure 11: Minimum sealing gap height and mean fluid pressure as a function of time.

Figure 12: Fluid pressure and sealing gap height for the stationary and the transient calculation

The influence of the transient Reynolds equation on the friction force is shown in Figure 13. For a normal load of 10 N especially the breakaway force is slightly increased. At a normal load of 31.1 N the friction force just
after the breakaway is higher. In the long term only a minor influence can be detected for the investigated system conditions.

Figure 13: Friction force as a function of time for two different normal loads with and without consideration of the transient term in the Reynolds equation

In conclusion the calculation based on the transient Reynolds equation leads to a lower fluid pressure and a smaller sealing gap. Thus, the friction force is slightly increased.

In all previously shown plots of the fluid pressure in the sealing gap a zone is visible were the fluid pressure reaches the vapour pressure \( p^{\text{vap}} \). Due to high gradients of the Reynolds equation in the outlet zone of the sealing gap an actually (physically not possible) negative pressure is calculated. A common approach is the constraint of Gämble where any fluid pressure \( p^f < p^{\text{vap}} \) is set to the vapour pressure /3/, /10/. In the current simulation model this approach is applied, including its effect on the global stiffness matrix \( K \). The influence of this correction on the gap height is shown in Figure 14.

Figure 14: Fluid pressure and sealing gap height with and without the avoidance of negative pressures

Only a minor effect on the calculated height can be detected. The influence on the friction force is negligible even at a minimum pressure of -0.5 bar, as shown in Figure 15.

Figure 15: Friction force and fluid pressure as a function of the rod velocity. A calculation with (physically not possible) negative pressures has no detectable influence.

When taking the time dependent change of the sealing height into account, the minimum pressure is distinctly decreased during breakaway. With the current constraint this leads to numerically unstable simulations for higher normal loads. Even though the influence might be small, a different approach to avoid negative pressures is necessary for a physically motivated simulation model.

4 Comparison with experimental data

For a comparison with experimental data a seal tribometer at the Institute for Fluid Power Drives and Controls is used to measure the friction force. A seal specimen is pressed with a normal force \( F_N \) against a rotating steel disc. A sketch of the test rig is shown in Figure 16. The test rig is described in detail in /12/. For the comparison a test weight of 1 kg is chosen, leading to a normal force in the contact of 31.1 N (due to the lever arm at the test rig)

Figure 16: left: Sketch of the test rig and the corresponding FE model; right: Comparison of experimental and simulation results. The friction force is plotted as a function of the rod velocity.

In general the comparison reveals a very good agreement of simulation and measurement for both full stationary and full transient calculation. As effects of the surface structure on the fluid film are not considered in this study, the comparison should only be evaluated qualitatively. The biggest discrepancy is revealed regarding the rod’s velocity at which breakaway occurs. In the measurement the velocity is higher. This might be caused by the flexibility of the force measurement arm, which is not considered in the simulation. In the measurement
breakaway occurs at a total travel distance of approximately 1.2 mm, which seems to be to large to be caused by the deformation of the O-ring alone.

5 Summary and Conclusion

In his study, three major transient influences on the behaviour of translational hydraulic seals were investigated in order to examine to which extend the consideration of these transient effects in a simulation of hydraulic seals is inevitable. In particular, the simulation study, based on a physically based FE model for a translational sealing contact, reveals the following results:

- Influence of the actual relative velocity in the contact:
  When the actual relative velocity is used in the calculation, the fluid pressure and the gap height decrease. This results in a slightly higher breakaway friction force.

- Influence of the non-constant coefficient of solid friction:
  For the investigated system conditions, the influences of the consideration of the area of real contact are not significant. But at higher contact pressures (e.g. for pressurised rod seals) the influence is assumed to be considerable. The consideration of an increased area of real contact due to idle time leads to an increased breakaway force.

- Influence of the time dependent change of the sealing height
  The calculation based on the transient Reynolds equation leads to a lower fluid pressure and a smaller sealing gap. Thus, the friction force is slightly increased.

In conclusion the transient influences on the calculated friction force are small but detectable for the investigated system conditions. For a physically motivated description they must not be neglected. But from an engineering point of view they might not be inevitable. On the other hand, no major influences on the calculation time were detected when considering the transient effects.

6 Acknowledgement

The research work was performed within a Reinhart-Koselleck project funded by the German Research Foundation (DFG). We would like to thank DFG for the project support under the reference German Research Foundation DFG-Grant: MU 1225/36-1 and PE 807/8-1.

Nomenclature

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<th>Variable</th>
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<td>$h$</td>
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<td>Global stiffness matrix</td>
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<td>$R_8$</td>
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<td>RHS</td>
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<td>u</td>
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<tr>
<td>$v_x$</td>
<td>Velocity</td>
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Reduction of bearing load capacity due to measured wall slip

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The present work investigates the temperature dependence of the Navier slip boundary condition and the related reduction of load capacity of a bearing. In part (i), the Navier slip boundary condition is discussed and a modified Reynolds equation, including slip, is derived. Based on this modified Reynolds equation, the pressure distribution and the load capacity of a slider bearing are obtained. Part (ii) presents the Darmstadt Slip Length Tribometer, utilized for measuring the slip length of technical rough surfaces. Part (iii) shows the temperature dependent results of the slip length measurements and the effect on the load capacity of the slider bearing in comparison to the standard no slip boundary condition.

Keywords: Fundamentals, Journal Bearing, Sealing Technology, Slider bearing
Target audience: Fundamentals, Pumps, Valves, Seals

1 Introduction

Since the beginning of the 20th century, hydraulic sealings and journal bearings [1-3] are designed employing Reynolds lubrication theory. The Reynolds lubrication theory presumes the no slip boundary condition at the liquid-solid interface. Recent studies conducted by the authors show, that the so far assumed no slip boundary condition at the liquid-solid interface is not valid for most fluid power applications; cf. [4]. This effects the prediction of leakage flow and frictional behaviour of sealing systems as well as bearing capacity of journal bearings. Thus, considering slip at the liquid-solid interface is important for the design of hydraulic components.

The concept of wall slip was already discussed by Navier [5] and Stokes [6], when deriving the momentum equation for Newtonian fluids in the 19th century. Stokes favours the no slip boundary condition and justifies his hypothesis by a good agreement of the theory with experimental investigations of Poiseuille [7]. In contrast, Navier [5] formulates the slip boundary condition, with the slip velocity being the product of the shear rate and the slip length. Recent experimental studies of the authors at the Technische Universität Darmstadt show that the slip length is for oil/steel of the order of magnitude of 100 nm with a strong temperature dependence.

Regardless of this discussion, the no slip boundary condition is established over the centuries, based on the insufficient measurement techniques. However, in many technically important applications of fluid power technologies wall slip is not negligible. This is the case if the quotient of slip length and typical flow geometry is less than 10⁻². Thus, for typical hydraulic systems is reasonable to consider slip, if the gap geometries are of the order of magnitude of 10 μm.

With the exception of the slip length measurement results provided by the authors [4], existing investigations and test methods are limited to material pairings far from hydraulic applications. The deployed liquids commonly are water or aqueous solutions [8-21] and pure hydrocarbons [15-19, 22]. Investigated solids generally are glass [8, 10, 11, 14, 18-21, 23] and mica [9, 15-17, 22]. These surfaces are smooth at the atomic level with a maximum square mean roughness of 2 nm. Typical surfaces of hydraulic applications however have a mean roughness of the order of magnitude of 100 nm.

Replacing the liquid does not pose a challenge when applying existing slip length measuring methods for hydraulic applications. However, solids [8-23] and in particular the surface topographies cannot be varied as desired. Because surface roughness is quantified by means of integral methods. The DIN EN ISO 4287: 2010-07 standard specifies the required integral length for measuring the surface roughness. The characteristic length of the slip length measurement should be at least one order of magnitude larger than the length for determining the surface roughness, since otherwise local disturbances affect the slip length measurement. Above mentioned slip length measuring methods have a characteristic length from 10 μm to 100 μm. Thus, they are limited to surfaces with a roughness of the order of magnitude of 1 mm.

In order to be able to measure systematic slip lengths for technical systems of hydraulic applications the Darmstadt Slip Length Tribometer (DSLT) [24] is developed at the Chair of Fluid Systems of the Technische Universität Darmstadt in cooperation with the Fluid Power Association of the VDMA. This tribometer enables measuring slip lengths for technically rough surfaces with a mean roughness of 10 nm to 400 nm. Furthermore, for the first time it is possible to measure the slip length as a function of temperature, which was out of focus in earlier studies. For hydraulic applications, this temperature impact on the slip length is of major relevance due to typical operating conditions of hydraulic systems.

This article provides novel insights on the influence of slip on a hydraulic bearing system. It is organized as follows. In part (i), the Navier slip boundary condition is introduced and a modified Reynolds equation, including slip, is derived. Based on this modified Reynolds equation, the pressure distribution and the load capacity of a slider bearing are obtained. Part (ii) presents the Darmstadt Slip Length Tribometer, utilized for measuring the slip length of technical rough surfaces. Part (iii) shows the temperature dependent results of the slip length measurements and the effect on the load capacity of the slider bearing in comparison to the standard no slip boundary condition. The paper closes with a summary and a conclusion.

2 Navier slip boundary condition and Reynolds equation

This section provides the theoretical fundamentals of the slip length and the application to the well-known Reynolds equation. Afterwards, the modified equation is solved analytically for a slider bearing.

2.1 Navier's slip boundary condition

Interesting is the train of thought, which Navier causes to formulate the slip boundary condition in 1822 [25]: Navier interprets the processes at the wall as a dynamic equilibrium between the shear force of the liquid at the wall μ du/dh and the wall parallel adhesive forces. The adhesive forces are proportional to the slip velocity us, conforming to Stoke's law. Hence the balancing yields

\[ u_s \cdot \text{const} = \frac{\partial u}{\partial n} \]  

Helmholtz [26] interprets the constant in Navier's relationship by means of dimensional analysis as a length, the nowadays called slip length λ

\[ \lambda = \frac{\text{const}}{\mu} \]

Thus, (1) yields the purely kinematic form

\[ u_s = \lambda \frac{\partial u}{\partial n} \]

known today, no longer revealing Naviers original thought and dynamic nature of the boundary condition. Figure 1 illustrates the geometrical interpretation of the slip length by means of a simple shearing flow example. It shows the velocity profiles for no slip boundary condition (grey) and Navier slip boundary condition (black). The lower surface is fixed while the upper surface moves with constant velocity U at a distance h. In the case of no-slip, the
velocity of the liquid molecules at the wall is identical to the wall velocity. In the case of slip, there is a relative velocity between the wall near molecules of the liquid and the wall.

The relative velocity at the fixed wall \( u_w \) is greater than zero and the relative velocity at the moving wall \( u_e \) reduces the velocity relative to the wall velocity \( U \). Extrapolating the velocity profile down to zero and up to the surface velocity \( U \) yields the slip length as the perpendicular distance from the surface to the boundaries of the extrapolated velocity profile. Due to this, Helmholtz’s [26] geometric interpretation of an apparent gap opening due to wall slip is deduced. Slip velocity and shear rate are proportional and the proportionality constant is the slip length.

\[ u_i(x_2, t) = \frac{1}{2d} \frac{d^2 p}{d x_i^2} x_i^2 (h + \lambda_i + \lambda_u) - x_2 (h^2 + 2h \lambda_u) - 2h \lambda_i \lambda_u - h^2 \lambda_i + U \frac{(x_2 + \lambda_i)}{h + \lambda_i + \lambda_u} \]  

(7)

The volume flow per unit depth in the \( x_i \) direction, \( i = 1, 2 \) is obtained by integrating the velocity distribution across the lubrication gap height

\[ q_i(t) = \frac{U h}{2} \frac{1 + 2 \lambda_i/h}{1 + \lambda_i/h + \lambda_u/h} \frac{h^2}{\mu} \frac{d p}{d x_i} \frac{1 + 4 \lambda_i/h + 4 \lambda_u/h + 12 \lambda_i \lambda_u/h^2}{1 + \lambda_i/h + \lambda_u/h}. \]  

(8)

Still the Couette term and the Poiseuille term are superimposed. This is because the equation of motion (4) and the boundary condition (5), (6) are linear, due to negligible inertia for \( \alpha Re \ll 1 \). The continuity equation in integral form (integrated from \( 0 < l_2 < h(x_i, t) \)) yields for incompressible fluids

\[ \frac{d h}{d x_i} + \frac{dh}{dt} = 0. \]  

(9)

Inserting (8) into (9) yields the here for the first time given generalized Reynolds equation, taking a Poisson type partial differential equation

\[ \frac{\partial}{\partial x_i} \frac{h^3}{\mu} \frac{d p}{d x_i} \left(1 + 4 \lambda_i/h + 4 \lambda_u/h + 12 \lambda_i \lambda_u/h^2 \right) = \frac{\partial}{\partial x_i} \left[ 6U h \left(1 + 2 \lambda_i/h \right) \right] + \frac{\partial h}{\partial t}. \]  

(10)

Equation (10) is the well-known Reynolds equation for plane flows, but now taking wall slip into account. Integrating this equation gives the pressure distribution within the fluid film.

2.3 Slider Bearing

In this section, the pressure distribution in the plane lubrication gap of a slider bearing is investigated, taking Navier’s slip boundary condition into account; c.f. Figure 2. The upper wall is inclined by the angle \( \alpha \) with respect to the \( x = y \) axis and moves at the constant velocity \( U \). The liquid is pulled into the narrowing gap, leading to a pressure increase. In sliding bearing technology, such an arrangement is also known as a slider bearing. So far, the load-bearing capacity for this arrangement is only examined for the kinematic no-slip boundary condition. In the following, the influence of wall slip is considered as well.

![Figure 1: Simple shearing flow for no-slip (grey) and slip (black).](image1)

The slip length is a characteristic quantity of tribology, characterizing each tribological system consisting of a liquid and a solid. The slip length depends on the molecular weight and the additives of the liquid, the surface material, the surface topography and the temperature. For identical solid-state pairings the slip length remains constant, \( \lambda_i = \lambda_u = \lambda \). This relationship is used in the experiment in section 5 for measuring the slip length.

2.2 Generalized Reynolds equation taking Navier’s slip boundary condition into account

So far, the Reynolds equation is derived and applied only assuming no-slip boundary condition. Since wall slip affects both the leakage flow of seals and the bearing capacity of plain bearings, it is necessary to solve the Reynolds equation while taking the dynamic slip boundary condition into account. Within this section the generalized Reynolds equation incorporating wall slip is derived in five steps.

Starting from the well-known assumptions of the Reynolds equation (cf. [27]), i.e. \( \alpha Re \ll 1 \), the Navier Stokes equations simplify to the linear differential equations

\[ \frac{d p}{d x_i} = \mu \frac{d^2 u_i}{d x_i^2}, \quad \frac{d p}{d x_i} = \mu \frac{d^2 u_i}{d x_i^2}, \quad \frac{d p}{d x_i} = 0. \]  

Integrating the linear differential equation twice provides the general solution of the velocity distribution in the lubrication gap. Applying Navier’s slip boundary condition for the velocities \( u \) at the fixed wall (\( y = 0 \)) and the moving wall (\( y = h \))

\[ u_i(x_2 = 0) = \lambda_i \frac{d u_i}{d x_2} \bigg|_{x_2=0}, \]  

\[ u_i(x_2 = h) = U_i - \lambda_i \frac{d u_i}{d x_2} \bigg|_{x_2=h(x_i)}, \]  

(5)

(6)

yields the special solution of the velocity distribution for \( i = 1, 2 \). Substituting the integration constants using the boundary conditions from Equation (5) and (6) provides the solution to the boundary value problem, with \( h = h(x_i, t) \),

\[ \frac{d^2 p}{d x_i^2} \left(1 + \frac{\lambda_i}{h} + 12 \frac{\lambda_i^2}{h^2} \right) = \frac{d}{d x_i} \left[ 6U h \left(1 + 2 \frac{\lambda_i}{h} \right) \right]. \]  

(11)
Integration of (11) leads to
\[
\frac{h^3}{\mu} \frac{dp}{dx} \left(1 + 8 \frac{h}{\mu} + 12 \frac{h^2}{\mu^2}\right) = 6Uh \left(1 + 2 \frac{h}{\mu}\right) + C_1.
\] (12)

The integration constant \(C_1\) is determined at the location \(x = \ell\) of maximum pressure, where \(dp/dx = 0\), giving the pressure gradient depending on the gap height \(h(x)\) and the gap height at maximum pressure \(H\)
\[
dp{h}{x} = 6UfL \frac{1 - \ell/h}{H^2 + 8\ell/h + 12\ell^2}.
\] (13)

With the gap height \(h(x) = h_1 - ax\) and substituting \(dx = -dh/\alpha\), the pressure distribution yields
\[
p(h(x)) = \frac{6UfL}{\alpha} \int \frac{1 - \ell/h}{H^2 + 8\ell/h + 12\ell^2} dh.
\] (14)

with the two integrals
\[
\int \frac{dh}{H^2 + 8\ell/h + 12\ell^2} = \frac{1}{4\alpha} \left[\ln(h + 2\ell) - \ln(h + 6\ell)\right] + C,
\] (15)
\[
\int \frac{dh}{H(h^2 + 8\ell/h + 12\ell^2)} dh = \frac{h}{2\ell^2} \left[2\ln(h - 3\ln(h + 2\ell) + \ln(h + 6\ell)\right] + C.
\] (16)

The integration constant \(C\) is determined by applying the boundary condition 
\[p(x = 0) = p(h_1) = 0.\]

The position of the maximum pressure \(H\) is determined by applying the condition 
\[p(x = L) = p(h_2) = 0,\]
resulting in
\[
H = 6\ell \left[\ln\left(h_1 + 2\ell\right) - \ln\left(h_2 + 2\ell\right)\right] - 3\ln\left(h_1 + 6\ell\right) + \ln\left(h_2 + 6\ell\right) + \ln\left(h_1 + 2\ell\right) - \ln\left(h_2 + 2\ell\right) + 3\ln\left(h_1 + 6\ell\right) - \ln\left(h_2 + 6\ell\right)
\] (17)

Hence, the pressure distribution is obtained as
\[
p(h(x)) = \frac{\mu L}{4\alpha^2} \left[2h_1 \ln\left(h_2/h_1\right) - 3(h_1 + 2\ell)h_1 + 6h_1 + 6\ell + 12\ell^2\right] \ln\left(h_1 + 2\ell/h_2 + 2\ell\right)
\] (18)

Integrating the pressure distribution from Equation (18) gives the load capacity of the slider bearing
\[
F(h) = \frac{\mu L}{4\alpha^2} \left[2h_1 \ln\left(h_2/h_1\right) - (3h_1 + 6h_2\ell + 6\ell + 12\ell^2) \ln\left(h_1 + 2\ell/h_2 + 2\ell\right) + (h_1 + 6h_2\ell + 6\ell + 36\ell^2) \ln\left(h_1 + 6\ell/h_2 + 6\ell\right) + (h_1 + 6h_2\ell + 6\ell + 36\ell^2) \ln\left(h_1 + 6\ell/h_2 + 6\ell\right)
\] (19)

The dimensionless load capacity is here the Sommerfeld number
\[
So = 5a = \frac{F(h)}{\mu L} = \frac{1}{4\alpha^2} \left[2h_1 \ln\left(h_2/h_1\right) - (3h_1 + 6h_2\ell + 6\ell + 12\ell^2) \ln\left(h_1 + 2\ell/h_2 + 2\ell\right) + (h_1 + 6h_2\ell + 6\ell + 36\ell^2) \ln\left(h_1 + 6\ell/h_2 + 6\ell\right)
\] (20)

3 Darmstadt Slip Length Tribometer: Function and measuring principle

This section gives the function and measuring principle of the Darmstadt Slip Length Tribometer (DSLT). The DSLT is an indirect measuring method for quantifying wall slip. The slip length is determined by the measurement of an integral quantity and a suitable model. The integral measured quantity is the friction torque between rotating and the stationary disk depending on the gap height. As a suitable model, the Reynolds equation is used, considering Navier's slip boundary condition.

The DSLT is a classical plate-plate tribometer (cf. Figure 3) measuring the friction torque transmitted from the rotating disk through the liquid film of height \(h\) to the stationary disk. The torque is measured with a reaction torque sensor with a response threshold less than 0.03 mN at the stationary disk. The distance is measured by means of capacitive distance sensors with a resolution of 4 nm. These sensors are integrated directly into the stationary disk. In order to allow a cardanic self-leveling of the two disks relative to each other, the lower one is supported by a jewell bearing. The adjustment of the lubrication gap is achieved by the axial spring stiffness and the feed pressure of the test liquid.

\[
F = \frac{\mu L}{4\alpha^2} \left[2h_1 \ln\left(h_2/h_1\right) - (3h_1 + 6h_2\ell + 6\ell + 12\ell^2) \ln\left(h_1 + 2\ell/h_2 + 2\ell\right) + (h_1 + 6h_2\ell + 6\ell + 36\ell^2) \ln\left(h_1 + 6\ell/h_2 + 6\ell\right) + (h_1 + 6h_2\ell + 6\ell + 36\ell^2) \ln\left(h_1 + 6\ell/h_2 + 6\ell\right)
\] (21)

In the lubrication gap, the pressure flow in radial direction and the drag flow in circumferential direction are superimposed. For small gap heights, the Reynolds number is of the order of magnitude of 0.1 and the tilt angle of the disks is smaller than 0.001°. Thus the Reynolds equation (8) for wall slip can be used, giving the Couette velocity profile \(u(r,\gamma) = \Omega(r + h)/h + \lambda_1 + \lambda_2\). With the velocity profile in the circumferential direction, the friction torque is determined by integrating the shear stresses. The inverse of the friction torque
\[
M^{-1} = \frac{h + \lambda_1 + \lambda_2}{\mu h \Omega}
\] (21)

is a linear equation. With the polar moment of area \(I_p = \int r^2 da\), the sum of the slip lengths at the stationary and the rotating disk can be obtained by determining the x-axis intercept for the curve of the equation.

Figure 4 illustrates schematically the relationship of equation (21) for Newtonian fluids. The inverse friction torque depends linearly on the gap height. As the no-slip boundary condition holds true, the linear curve of the inverse friction torque intersects the coordinate origin. As the rotational frequency of the rotating disks increases, the slope of the straight-line equation decreases. If wall slip occurs at the liquid-solid interface, the curve intersects the x-axis in the negative. This distance of the x-axis intercept to the coordinate origin is equal to the sum of the two slip lengths at the stationary and the rotating disk. As in the case of no slip, the slope of the linear curve decreases with increasing rotational frequency of the rotating disk, however the point of intersection with the absissa remains unchanged. Thus, the slip length for Newtonian fluid is independent of the shear rate.

In the conducted experimental studies, gap height and friction torque are measured; cf. markers in Figure 4. The slip length is obtained by determining the x-axis intercept of the best-fit line, which is obtained by means of the
least square fit method. This linear extrapolation is necessary since the measuring method is an indirect measuring method. However, this is not considered a disadvantage since the slip length is not a function of the gap height, as the linearity of the inverse friction torque shows. Rather, the systematic measurement over a gap height change of 10 μm offers two advantages: (i) the plausibility of the individual measurements is examined over a wide measuring range and (ii) the shear rate varies by an order of magnitude during the measurement, thus a shear rate independence can be verified at once.

Two planar disks made of surface hardened stainless steel (steel type 1.8519) with a diameter of 64 mm form the lubrication gap. Due to the hardened surface it possible to manufacture a surface flatness of 30 nm with a mean roughness of 10 nm to 100 nm by means of lapping. The disks utilized for the conducted studies have a mean roughness of 10 nm to allow comparison with the slip lengths of smooth surfaces published in literature. The diameter of the measuring disks constitutes the decisive advantage of the DSLT in comparison to the so far used measuring methods. As discussed in the introduction, the measurement geometries in the earlier used measuring devices are too small to quantify slip lengths for technically rough surfaces.

Technical surfaces are quantified according to DIN EN ISO 4287: 2010-07 using integral quantities, such as arithmetic mean roughness or average roughness. DIN EN ISO 4288: 1997 specifies the measuring section averaging the surface roughness. For technically rough surfaces with a mean roughness of the order of 100 nm, a single measuring section of 800 μm is required. The characteristic lengths of so far used measuring geometries for slip length measurement vary from 1 to 100 μm. These characteristic lengths are smaller than the lengths over which parameters for characterizing surface topographies are integrated. Based on this fact, it is reasonable to distinguish slip length measuring methods into local and integral measuring methods based on the effective length of the measuring geometry.

Measuring methods whose characteristic measuring geometry is smaller or of the same order of magnitude as the effective length that characterizes the surface roughness are considered as local measuring methods. Measuring methods whose characteristic measuring geometry is at least one order of magnitude greater than the characteristic length that quantifies the surface roughness are considered as integral methods. This ensures that the measurement is integrated via local effects and the slip length represents an average over the rough surface, analogous to the surface roughness itself.

4 Results

This section presents the results of the conducted slip length measurements utilizing the DSLT. Additionally, the influence of the slip length on the load bearing capacity of the slider bearing is discussed. The following section is subdivided into two subsections. First, slip length measurements at constant temperature are presented, depicting that wall slip exists in hydraulic systems. Due to the fact, that hydraulic systems are not operated at constant temperature, it is necessary for system design to know the slip length depending on the temperature. Thus, the authors verified an Arrhenius relation for the thermal characteristic of the slip length; cf. [4]. Second, with these temperature-dependent slip lengths, the influence of the slip length on the load capacity of a slider bearing is quantified.

4.1 Slip length measurements

Figure 5 shows the measurement of the inverse torque as a function of varying gap heights, showing the $M^{-1} - h$ curve for an alpha-olefin at constant temperature of 29.9 °C and constant rotational frequency of 2 Hz. The symbols mark the individual measurement points at which the torque was measured depending on the gap height. The different colors of the markers characterize the repeated measurements. Overall, the figure shows 20 measurement series. The best fit linear regression curve discussed in Figure 3 is determined individually for each measurement series and the slip length is determined from the intersection with the x-axis; cf. Figure 4.

Figure 6 gives a detailed view at the intersection of the regression curves with the x-axis. The measured slip length for an alpha-olefin at 29.9 °C averages to $\lambda = 540$ nm. The measurements can be repeated with a standard deviation of $\sigma = 50$ nm. Due to the 20 repetitions, the statistical uncertainty is reduced by the factor $\sqrt{20}$ (student-t-distribution). The statistical uncertainty of the measured value with $n = 20$ repetitions and a confidence interval of 95% is less than 24 nm and thus less than 5% of the measured mean value. The systematic errors are dominated by the distance measurement.

The distance measurement is critical for two reasons: On the one hand, the permissibility of the test liquid is temperature-dependent and on the other hand, the sensor position changes as a function of the temperature relative to the gap surface. This means that the distance sensor has to be calibrated for each test fluid depending on the temperature. This is done by means of interferometric layer thickness measurements as an absolute reference.

![Figure 5: Slip length measurements for PAO 6 at $\Theta = 29.9$ °C and constant rotational speed of 2 Hz. The figure shows 20 measurement runs. Each run is marked by a symbol. Only the last one marked by triangles is visible.](image)

Figure 5 shows the measurement of the inverse torque as a function of varying gap heights, showing the $M^{-1} - h$ curve for an alpha-olefin at constant temperature of 29.9 °C and constant rotational frequency of 2 Hz. The symbols mark the individual measurement points at which the torque was measured depending on the gap height. The different colors of the markers characterize the repeated measurements. Overall, the figure shows 20 measurement series. The best fit linear regression curve discussed in Figure 3 is determined individually for each measurement series and the slip length is determined from the intersection with the x-axis; cf. Figure 4.

4.2 Bearing capacity of the slider bearing

Figure 7 gives the pressure distribution in a lubrication gap for the slip as well as for the no slip boundary condition applying Equation (18). The geometry of the used slider bearing is shown in Figure 2. The gap height $h_1$ is 20 μm, the tilt angle $\alpha$ of the upper plate is 0.286° and this plate is moved at constant velocity $U$ of 0.3 m/s. The utilized fluid is an alpha-olefin with a dynamic viscosity $\mu$ of 0.039 Pa·s and a slip length $\lambda$ of 540 nm at 29.9 °C. The quotient of slip length and typical flow geometry $h_2$ is thus 0.027. Figure 7 exhibits, that the peak pressure is reduced due to slip by approximately 25 %. Integrating the pressure along the horizontal directions yields the load capacity per unit depth, which is also reduced by approximately 25%.

Hydraulic applications operate in a wide temperature range. Thus the thermal behaviour of the load capacity is of major interest. The temperature dependent dynamic viscosity and the temperature dependent slip length are obtained by means of Arrhenius relations; cf. [4]. Using these Arrhenius relations, Figure 8 gives the temperature depending load capacity of the above mentioned slider bearing. The presented results clearly show that difference in load capacity per unit depth for slip and no slip decreases with increasing temperature. This is reasonable since the activation energy for wall slip is smaller than the activation energy for shearin.
5 Summary

The presented article provides for the first time the application of the slip length at hydraulic systems. As a hydraulic system, a typical slider bearing is considered, quantifying the influence of wall slip by means of the pressure distribution and the load capacity.

At first, the slip boundary condition is introduced following the dynamic considerations of Navier. Based on this slip boundary condition the Reynolds equation considering wall slip is derived. Integrating the Reynolds equation gives the pressure distribution as well as the load capacity of a slider bearing.

For measuring the slip length of hydraulic applications the Darmstadt Slip Length Tribometer (DSLT) is used. The slip length for tribological system steel/oil/steel at a temperature of 29.9 °C is measured and accounts for 540 nm. The influence of slip on the slider bearing is quantified by means of pressure distribution and load capacity, showing that the consideration of slip reduces the load capacity of the considered geometry by approximately 25%.

6 Acknowledgements

The authors thank the Research Association for Fluid Power of the German Engineering Federation VDMA for its financial support. Special gratitude is expressed to the participating companies and their representatives in the accompanying industrial committee for their advisory and technical support.

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Advanced heat transfer model for piston/cylinder interface

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The piston/cylinder interface in axial piston machines requires both sealing and bearing functions. The fluid and structure coupled physical phenomena including the temperature distribution of the piston and cylinder block controls the gap fluid behavior, therefore, the dual functions of the piston/cylinder interface. Instead of addressing the heat transfer problem of the piston and the cylinder block separately as the former model, the proposed advanced heat transfer model solves the temperature distribution of both solid bodies together using the fluid domain heat transfer characteristic to assemble the two solid parts. Comparing to the former unconnected heat transfer model, the integrated model is found more robust and accurate especially at challenging operating conditions.

Keywords: Piston/cylinder interface; fluid structure and thermal interaction modelling, heat transfer

Target audience: Tribological interface design, Mobile hydraulics

1 Introduction

The fluid behavior of the piston/cylinder interface in axial piston machines is controlled by the dynamics and kinematics of the piston, the piston micro motion, and the deformations on the running surfaces of piston and cylinder bore under both the pressure and thermal load. Unlike the cylinder block/valve plate interface and the slipper/swashplate interface, the gap height of the cylindrical shaped lubricating gap is very sensitive to the dimensions of the piston and cylinder bore, more specifically, the outer diameter of the piston and the inner diameter of the cylinder bore. Due to the fact that clearance between the piston and cylinder bore is at the level of microns, the expansion and the shrinkage of the piston and the cylinder bore due to the thermal load notably influence the fluid behavior, therefore, the performance of the piston/cylinder interface.

The thermal load, which is the rate of energy dissipation due to the viscous shear in the fluid domain, the conduction through the thin fluid film, and the convection of the leakage flow, the compression and expansion of the gap flow, and the conduction from the running surface on the other side of the gap. All of the determining effects that are mentioned above changes with the behavior of the fluid, therefore, the deformation of the piston and cylinder bore. These complicated physical phenomena include fluid, structure, thermal, and the interaction in between, making the modeling study of the piston/cylinder interface in axial piston machines a challenging topic.

The first numerical pressure distribution simulation in the piston/cylinder gap is conducted by Van der Kolk (1972). The piston/cylinder interface in his model was simplified as a tilted journal bearing. The piston motion was firstly considered in the pressure distribution calculation by Yamaguchi (1976). The first non-isothermal fluid model for the piston/cylinder interface was introduced by Ivantysynova (1983) who coupled Reynolds equation with energy equation and calculated the fluid behavior with non-constant fluid properties. Ofema (2001) firstly solved the piston micro motion from the force balance between the external load and the fluid pressure force in the gap. Ivantysynova and Huang (2002) firstly published an elasto-hydrodynamic model for piston/cylinder interface that utilizes an influence matrix to calculate the pressure deformation of the solid part. This deformation is then superposed on the gap height and the pressure distribution can be solved by an iterative method. Pelosi and Ivantysynova (2009) firstly introduced a fluid-structure and thermal interaction model for the piston/cylinder interface which considered not only the deformation under the pressure load but also the thermal load on the piston and cylinder block solid body. The temperature distribution on the solid parts is calculated through a three-dimensional heat transfer model using the temperature distribution in the gap and the fluid temperature and heat transfer coefficient in the surrounding control volumes as boundary conditions. In order to attain these boundary conditions, the fluid temperature in the inlet, outlet and case volume must be measured until Shang and Ivantysynova (2015) developed a port and case flow temperature prediction model. Their model enables digital prototyping of the piston/cylinder interface without the support from the measurement.

In the thermal model of Pelosi and Ivantysynova (2009), the temperature distribution of the piston and the cylinder block was solved separately. There is no communication between the piston and cylinder block temperature even though the conduction through the thin fluid film makes the temperature of both solid bodies highly dependent on each other. In this article, an advanced heat transfer model is proposed that integrated the two solid bodies into a single system and solves the temperature distribution of both the piston and cylinder block simultaneously. The methodology to establish the communication between the two finite element system with their relative location and motion keep changing with time due to the reciprocating and spinning motion of the piston is explained in the article. The simulation results are then presented at the end of the article and compared to the previous model.

2 Piston/Cylinder Interface

In swashplate type axial piston machines, the stroke, the axial velocity, and the acceleration of the piston is controlled by the geometry of the piston/cylinder interface, the swashplate angle, and the shaft angular position and velocity.

The piston stroke follows:

\[ s_p = \frac{-d_a \tan \beta}{2(1 - \cos \varphi)} \]  

where \( d_a \) is the pitch diameter, \( \beta \) is the swashplate angle, and \( \varphi \) is the shaft angular position.

The piston axial velocity is derived from the derivative of the piston stroke:

\[ v_a = \frac{ds_p}{d\varphi} = -\frac{1}{2} d_a \tan \beta \sin \varphi \]  

where \( \varphi \) is the shaft angular velocity.

The piston axial acceleration then can be derived from the derivative of the piston axial velocity:

\[ a_a = \frac{dv_a}{d\varphi} = \frac{1}{2} d_a \tan \beta \cos \varphi \]  

Besides of the axial motion, the piston in the cylinder bore is also subjected to the relative spinning motion which is discovered by Renius (1974) and confirmed by Lasar (2003). The axial motion, spinning motion, together with the micro motion including the tilting and squeezing motion build up the fluid pressure that balances the external load of the piston.

Figure 1 shows a free body diagram of the piston in a swashplate type axial piston machine. The piston body is subjected to the displacement pressure force \( F_{disp} \), the axial inertia force of the piston/slipper assembly \( F_a \), and the axial friction \( F_f \) between piston and cylinder bore. The total axial force is balanced by the axial component of the reaction force \( F_{react} \) from the swashplate through the slipper and the ball joint applies on the piston body. As shown...
in Figure 1, due to the inclination of the swashplate, the reaction force \( F_{Rc} \) has another component pointing to the side of the piston. This \( F_{Rc} \), together with the slipper friction force \( F_{fs} \), applies on the center of the piston ball as the total resultant force \( F_{ak} \). This resultant force and the radial inertia force of the piston slipper assembly \( F_{rk} \) and their associate moments have to be balanced by the pressure distribution between the piston and the cylinder bore.

Figure 1: Piston free body diagram.

3 Piston/Cylinder Interface Fluid Structure And Thermal Interaction Model

In order to study the complicated physical phenomena behind the piston/cylinder interface and investigate novel piston/cylinder interface designs, a fluid structure and thermal interaction model was firstly developed by Pelosi and Ivantysynova (2009) and continuously updated by Pelosi (2012) and Pelosi and Ivantysynova (2012a, 2012b, 2013).

In the above mentioned model developed by Pelosi and Ivantysynova, the two-dimensional pressure distribution and the three-dimensional temperature distribution is solved in the finite volume method fluid domain model by using the Reynolds equation and the energy equation. The instantaneous pressure deformation of the piston and cylinder block solid body is calculated in the elastic deformation model which linearly scales the pre-constructed influence matrix according to the gap pressure that attained from the FVM fluid model. The pressure deformation is then superposed on the gap height of the piston/cylinder interface and fluid structure interaction problem as shown in Figure 2 is solved when the iterative loop converged. The thermodynamic model in the FVM fluid model then solves the three-dimensional fluid domain temperature distribution as heat transfer problem using the running surface temperature as the boundary condition and the energy dissipation and the enthalpy change due to compression and expansion as the source term. The temperature distribution is then used to update the fluid properties for the next time step calculation and used to derive the heat flux on the solid bodies running surface. The three-dimensional temperature distribution of piston and cylinder block solid bodies is calculated between each shaft revolution. By assuming that the solid body temperature distribution remains unchanged while the pump or motor operating at steady state, the instantaneous heat flux was averaged over one completed shaft revolution and used as a steady state boundary condition for solving the solid part temperature distribution. This solid part temperature distribution is used as the thermal boundary for the fluid domain temperature distribution calculation in the next shaft revolution, and used as the thermal load for the solid body thermal deformation calculation. The thermal deformation is then used for the finite volume fluid model in the next revolution. As shown in figure 2, the fluid structure and thermal interaction problem is solved when the solid part temperature reaches convergence over revolutions.

In general, there are two types of thermal boundaries used for the three-dimensional heat transfer model. The Neumann boundary is used for the running surface of the piston and the cylinder block solid bodies as shown in figure 3. The heat flux on the Neumann boundary is calculated from the temperature differential in the direction of the fluid film thickness and the thermal conductivity of the gap fluid:

\[
q = \lambda_{gap} \frac{\Delta T}{L_z}
\]  

(4)

The mixed boundary is used for the rest of the surfaces of the piston and the cylinder block solid bodies as shown in figure 3. The heat flux on the mixed boundary is calculated from the temperature difference between the fluid in the surrounding control volume and the surface temperature, and the heat transfer coefficient on the surface:

\[
q = \alpha_{surf} \cdot \Delta T
\]

(5)

Figure 2: Fluid structure and thermal interaction simulation scheme for piston/cylinder interface.

Figure 3: Thermal boundary conditions for piston and cylinder block solid bodies.
first order of the temperature differential but not the absolute temperature, the solved three-dimensional temperature distribution is very sensitive to the heat flux on the running surface. This can lead to a situation where the piston and cylinder block temperature will alternate from revolution to revolution without getting stabilized, i.e. the piston hotter than the cylinder block in the i-th revolution will lead to a piston colder than the cylinder block in the i-th revolution with repeating such cyclic behavior. This fluid and thermal interaction simulation scheme is unstable by nature and is difficult to achieve convergence without relaxing and saturating the heat flux between each iteration. Therefore, at challenging operating conditions, the accuracy of the simulation result must be compromised in order to converge the solid temperature distribution.

4 Advanced heat transfer model for piston/cylinder interface

The three dimensional solid body heat transfer model for the piston/cylinder interface mentioned in previous chapter is based on the finite element discretization of the solid bodies. The steady state diffusive form of energy equation as shown in Eq. 6 is used to determine the temperature distribution.

\[ \nabla \cdot (\lambda \nabla T) = 0 \]  
\[ (7) \]

The temperature at each nodal position then solved in a linear system:

\[ M_pT_p = q_p \]
\[ M_cT_c = q_c \]
\[ (8) \]

where \( T_p \) and \( T_c \) are the temperature field of piston and cylinder block, \( q_p \) and \( q_c \) are the thermal boundaries on the piston and the cylinder block solid bodies, and \( M_p \) and \( M_c \) are the matrices contain the conductivity between each node that constructed from the material properties of the piston and the cylinder block solid bodies and the position of each node.

Unlike the simulation scheme for the fluid structure and thermal interaction of the piston cylinder interface model mentioned in the previous chapter which solves the temperature distribution in the piston and the cylinder block solid bodies separately, the proposed advanced heat transfer model solves the dual-body temperature distribution in an integrated linear system.

In this integrated linear system as shown in Eq. 9, there are two extra matrices \( M_{kp} \) and \( M_{kb} \) contain the conductivity of the fluid film between the piston and the cylinder block.

\[ \begin{bmatrix} M_p & 0 \\ 0 & M_c \end{bmatrix} \begin{bmatrix} T_p \\ T_c \end{bmatrix} = \begin{bmatrix} q_p \\ q_c \end{bmatrix} \]
\[ (9) \]

In order to construct the matrix \( M_{kp} \), the relationship of the temperature at face \( i \) of the piston running surface, the temperature at the face \( j \) on the cylinder bore running surface, and the average heat flux between face \( i \) and face \( j \) in one shaft revolution must be found first. The instantaneous piston/cylinder interface mesh relative position is shown in figure 4.

For each single time step, the heat flux on piston face \( i \) can be determined by:

\[ q_i = \frac{\lambda}{h_{ij}} (T_{ sea } - T_j) \]
\[ (10) \]

where \( h_{ij} \) gap height and \( T_{ sea } \) is the cylinder bore surface temperature at the reference location. As shown in figure 4, the reference location is defined as the projection of face \( i \) on cylinder bore. Due to the relative motion of the piston, having a face exactly at the reference location is not possible. Therefore, the reference temperature \( T_{ sea } \) is the weighted average of the temperatures of the faces on cylinder bore running surface near the reference location.

Then, the instantaneous heat flux from face \( j \) to face \( i \) yields:

\[ q_i = \frac{w_{ij}}{h_{ij}} (T_j - T_i) \]
\[ (11) \]

where \( w_{ij} \) is the weight according to the distance between the location of face \( j \) and the reference location.
Each triangular face is defined by three nodes. As shown in figure 4, face $i$ is defined by node $i_1, i_2, i_3$ and face $j$ is defined by node $j_1, j_2, j_3$. Each node can be used to define multiple faces. The face temperature is defined as the average of the temperature of the three nodes. The heat flux on each face distributed to the three nodes evenly is shown in figure 5.

The relationship between the temperature at piston nodes $i_1, i_2, i_3$ and cylinder block nodes $j_1, j_2, j_3$ and the conduction heat flux from face $j$ to face $i$ yields:

$$
\sum_{i=1}^{n} \sum_{j=1}^{m} M_{ijn} \left[T_j[i_j] + \sum_{i=1}^{m} M_{ijn} \left[T_i[i_i] \right] \right] = q_{inc,i} \cdot A
$$

where $A_i$ is the area of face $i$.

Then, the matrix $M_{ijn}$ can be constructed as:

$$
M_{ijn} \left[i_i, j_j \right] = \frac{n}{60} \cdot \frac{\lambda_{oil}}{9} \cdot A_i \cdot B_{2K} [i_i]
$$

$$
M_{ijn} \left[i_i, j_j \right] = \frac{n}{60} \cdot \frac{\lambda_{oil}}{9} \cdot A_j \cdot B_{2K} [j_j]
$$

The matrix $M_{ijn}$ can be constructed using similar method.

In the fluid structure and thermal interaction model using the proposed advanced heat transfer model, thanks to the fact that the conduction between the two running surfaces is already included in the system, the only changing thermal boundary between each iteration is the energy dissipation and the fluid connection in the gap, which is less sensitive to the solid body temperature.

5 Simulation results comparison

In order to test the fluid structure and thermal interaction model with the advanced heat transfer model, a challenging operating condition is used with a tight clearance. The operating condition is 3600 rpm, 50bar, and 20% displacement. The high speed leads to high energy dissipation in the gap. The low pressure causes low gap flow, therefore, low fluid convection. The low displacement lead to low axial motion of the piston, therefore, concentrated heat flux.

Figure 6 shows the simulated piston temperature distribution from the fourth revolution to the ninth revolution using the previous heat transfer model. Obviously, the piston temperature was not converging in this iterative fluid structure and thermal interaction simulation process.

Figure 7 shows the simulated piston temperature distribution from the fourth revolution to the ninth revolution using the novel in this paper proposed advanced heat transfer model. The piston temperature converged on the ninth revolution and the temperature difference between each iteration shows a stable converging process.

6 Conclusion

The proposed advanced heat transfer model for the solid parts of the piston/cylinder interface that uses the integrated linear system to solve the temperature distribution in both solid bodies simultaneously improves the convergence of the iterative process of the fluid structure and thermal interaction model without compromise the result accuracy.

The methodology to construct the integrated linear system to solve the dual-body temperature distribution using finite element method is explained in this article. A challenging operating condition with high speed, low pressure, and low displacement was used to test the performance of the model. The simulation result comparison that presented at the end of the article demonstrates the improvement of this proposed advanced heat transfer model.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>$s_i$</td>
<td>Piston stroke</td>
<td>m</td>
</tr>
<tr>
<td>$d_e$</td>
<td>Pitch diameter</td>
<td>m</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Swash plate angle</td>
<td>deg</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Shaft angular position</td>
<td>deg</td>
</tr>
<tr>
<td>$v_i$</td>
<td>Piston axial velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>$\omega$</td>
<td>Shaft rotational velocity</td>
<td>rad/s</td>
</tr>
<tr>
<td>$a_i$</td>
<td>Piston axial accretion</td>
<td>m/s^2</td>
</tr>
<tr>
<td>$q$</td>
<td>Heat flux rate</td>
<td>W/m^2</td>
</tr>
<tr>
<td>$\lambda_{oil}$</td>
<td>Fluid conductivity</td>
<td>W/K/m</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
<td>°C</td>
</tr>
<tr>
<td>$z$</td>
<td>Direction of fluid film</td>
<td>m</td>
</tr>
</tbody>
</table>
\( \alpha_{\text{surface}} \) Surface heat transfer coefficient [W/K·m²]  
\( \Delta T \) Temperature difference between the fluid temperature and surface temperature [°C]  
\( \lambda_{\text{solid}} \) Solid body thermal conductivity [W/K·m]  
\( \mathbf{M} \) Conductivity matrix  
\( \mathbf{T} \) Vector of temperature [°C]  
\( \mathbf{q} \) Vector of heat flux [W/sm²]  
\( A \) Face area [m²]  
\( t \) time [s]  

References


