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Foreword

The International Conference on Hydraulics and Pneumatics organised by the Czech Association for Hydraulics and Pneumatics, which is part of the Czech Mechanical Engineering Society, and by the Department of Control Systems and Instrumentation of the Faculty of Mechanical Engineering at the VŠB-Technical University of Ostrava is a traditional meeting point of the specialists both from the industry and the universities with more than forty five years long tradition. The 24th conference organised this year again in Prague, in the seat of the Czech Mechanical Engineering Society, part of which is the Czech Association for Hydraulics and Pneumatics. The co-organizer of the conference, the Department of Control Systems and Instrumentation at the VŠB-Technical University of Ostrava, was involved in the network of foreign workplaces, former Fluid Power Net International (FPNI). This partnership has made it possible to establish the international scientific committee of the conference. I would like to express my great thanks to all the members of the scientific committee for their demanding work while reviewing the papers, which significantly contributed to the quality course of the conference.

I believe that the conference programme will be interesting and useful for all participants and the meeting will support the mutual understanding and the establishment of personal contacts necessary for a successful cooperation and partnership in research and development also outside the Czech Republic.

prof. Ing. Petr Noskievič, CSc.
Chairman of the Scientific Committee
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OPENING SESSION
Simulation of Drive for Electric-Hydraulic Excavator

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Abstract: The article describes some methods used in modeling and simulation of a drive of electric-hydraulic excavator. The paper introduces basic model schemes of the drive system, the simulation models of the electric motor system, of the pump driven by the electric motor, examples of simulation results – time courses of model quantities received either by means of Matlab/Simulink or a set of differential equations.

The electric drive powered by electric accumulators as a single source of energy is a crucial innovation in contradiction to the traditional drive of excavator by an internal combustion engine.

Keywords: simulation, electric, hydraulic, drive, excavator

1 Introduction
The research objective was to prepare materials for solving substitution of internal combustion engine by electric motor in drive of an excavator. For the drive, an electric synchronous motor with a rotor equipped with permanent magnets (PMSM) was used. The innovated machine can work in closed space, enclosed premises of hospitals, protected areas, etc. because surroundings are not burdened with exhaust gases and noise Mathematical modeling of individual subsystems and also of the entire excavator enabled to improve the properties of designed construction. Except for hydraulic motors, all hydraulic components were removed and a completely new hydraulic system was added.

2 Torque and power analysis of excavator
To simulate the electric motor, the power and torque behaviour during respective excavator working cycles was analysed. A decision to focus on digging and travel cycles was made based on the manufacturer’s experience with these excavators. The standard version of the compact excavator uses a diesel engine of 9.9 kW; a new electric drive should have at least the same power. To obtain torque characteristics, hydraulic system parameters were measured as we decided that hydraulic cylinders, the slew, and travel hydraulic motors would remain unchanged. The following parameters were measured: the pressure and flow in all pumps and the temperature of hydraulic oil. The speed of the motor using OBD (On-Board Diagnostics) was also measured.

The most important measurements were carried out during digging and straight travel of the excavator. A digging test was performed with approximately 90° turn to empty the bucket. Cycle times of hydraulic cylinders were also measured. During the cycle time measurements,
no additional load was applied on front attachment. Cycle times of the excavator with electric drive (Fig. 1) should correspond to the original solution. To save energy and use the advantage of dynamic characteristics of electric drive, it was decided to set the speed of the electric motor based on the sum of required speed from joysticks and pedals signals. If there are no speed requirements, the motor stops.

Figure 1 - Excavator during digging test

Manufacturer’s hydraulic system (all data were measured in this system) consists of one dual axial piston pump and the additional gear pump. A combined displacement of this solution is 2x6 cm³/rev and 4.5 cm³/rev. A dual axial pump has also a torque limit to reduce its displacement when a critical torque is reached to prevent the engine stall (Fig. 2).

Figure 2 - Drive system with diesel engine, dual axial piston pump and additional gear pump

The original pump has been replaced in our new solution by a single axial piston pump with variable pressure control for better flexibility. Displacement of the new pump is 18 cm³/rev. Directional valves were also changed to an electrically operated Rexroth closed centre LUDV valves from the originally hydraulically or mechanically operated open center directional valves. The designed new system (Fig. 3) with the electric motor consists of the following main parts: battery charger, battery, control system, electric motor with controller, LUDV
control block and hydraulic motors. The belt gear is a part of the subsequent model of the pump.

Comparison of Fig. 2 and Fig. 3 shows which components were removed and added related to the standard system. Torque of the new system obtained from the measured data was calculated based on pressure, displacement and speed of the engine (Fig. 4, 5). To reduce the torque required to achieve the maximum system pressure of 25 MPa with 18 cm$^3$/rev pump, a synchronous belt drive with 5:3 speed reduction was added.

Figure 3 - New system with electric motor and single axial piston pump

Figure 4 – Torque characteristics during digging test (diesel engine)
3 Model of electric drive of excavator

Modeling of individual subsystems [2] and also of the entire excavator enables to improve the properties of designed construction including the significant subsystem of electric drive of the pump. Control of the electric motor and pump, their connecting into the system of the excavator drive can be seen in Fig. 6.

3.1 The electric motor as a subsystem

This chapter deals with modeling of permanent magnet synchronous motor (PMSM) [3] in the excavator pump application. As mentioned above, the former combustion engine was replaced by an electric PMSM chosen according to a large set of measured data gathered in the former application with the combustion engine. The model is then used for verification of dynamic and static requirements on the electric drive given by the excavator operational diagram. While the chosen PMSM was realized as a prototype, proper measurement of motor parameters has to be carried out to build a realistic model of the motor. The well-known techniques of measurement of PMSM parameters are sufficiently described in [3]. According to these measurements, the following PMSM parameters were obtained and summarized in the following table.
Table 1. PMSM parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stator resistance $R_s$ [mΩ]</td>
<td>3.3</td>
</tr>
<tr>
<td>Stator inductance in $d$-axis $L_d$ [µH]</td>
<td>100</td>
</tr>
<tr>
<td>Stator inductance in $q$-axis $L_q$ [µH]</td>
<td>50</td>
</tr>
<tr>
<td>Rotor inertia $J_m$ [kg m$^2$]</td>
<td>0.00095</td>
</tr>
<tr>
<td>Number of pole pairs $p_p$ [-]</td>
<td>4</td>
</tr>
<tr>
<td>Motor flux $\Psi_F$ [Wb]</td>
<td>0.0214</td>
</tr>
</tbody>
</table>

A complete control structure of PMSM with field-oriented control (FOC) is shown in Fig. 7. A cascade control structure with the speed control loop and the inner current loops is used. The dynamics of motor inverter is also considered along with decoupling techniques for better performance of current control loops. The motor speed and load torque are set according to the real data obtained from the former excavator with a combustion engine.

The model of PMSM motor is realized according to the following set of equations (see in [3] where physical symbols including units can be found):

$$u_d = R_i i_d + L_s \frac{di_d}{dt} - \alpha L_i i_q$$

$$u_q = R_i i_q + L_s \frac{di_q}{dt} + \alpha L_i i_d + \omega \Psi_f$$

$$T_m = \frac{3}{2} [p_p \Psi_f i_q + (L_d - L_q) i_d i_q]$$

$$J \frac{d\omega_{mech}}{dt} = \frac{J}{p_p} \frac{d\omega}{dt} = T_m(t) - T_L(t)$$

A simulation result is shown in Fig. 8. The required speed is kept on the constant value of 2500 rpm. The motor torque changes dynamically according to the excavator operation. The results show that the motor is sufficiently designed for the required pump speed and load torques. There is still a lot of reserve in the amplitude of stator voltage (40 V) as well as in the stator current (300 A) to ensure the required torque and speed from both the dynamic and static point of view.

**3.2 Simulation results of electric drive of excavator**

Some examples of simulation results of electric drive in the mode of speed control and with step changes in torque are given in Fig. 8 and Fig. 9. Excellent dynamics of PMSM controlled via FOC technique, where the speed is kept constant with minimal disturbances caused by steep torque changes in comparison with combustion engine, is shown in Fig. 9. Good
conformity between the model and the real drive resulted from measured properties of the real drive in the laboratory [3].

Figure 8 – Time courses of simulated quantities

4 The subsystem of pump

The pump is the only source of pressurized oil for hydraulic motors of arm, for travel and rotation in the excavator. In our model, the pump is driven by an ideal source of angular velocity controlled by signals from the block Signal Builder. More information concerning chapter 4 can be seen in [1]. The adjustable displacement of the pump was used for the prediction system with the power control. The main part of the considered pump model is in Fig. 10. Great attention was paid to the securing function of pressure controller with override, which protects the pump against overload: if too high pressure occurs, non-rotating control swash plate of the pump is set in such a manner that the pressure will remain under the given limit. A result of such a securing process is obvious from Fig. 11. It is valid for the set limit of 24 MPa and the pump speed of 3000 rpm. The standard system and the electric system are so qualitatively different that the comparison of their quantitative parameters is not performed.
Conclusions

In this paper, development of the new type of drive for electric-hydraulic excavator up to 2 tons is described. Modeling and simulation of drive have enabled to achieve a quality solution hardly attainable by the use of other methods. New methods of control of the electric-hydraulic system have been developed. Results of this research and development represent an energy-efficient excavator consisting of electronic and hydraulic equipment with better ecologic and dynamic properties in comparison with contemporary drives using internal combustion engines. Simulation results confirmed the expected benefit of the electric drive in
comparison with the combustion engine drive, especially in terms of the drive dynamics and operation economy.

Figure 11 – An example of load changes of pump and function of overload protection

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References
Abstract: The paper relates to the fluid force acting on the journal of the slide bearings and its movement at stable lubrication. These forces can be calculated using the Reynolds equation. The analysis is based on the solution of the simplified equation and the experimental verification with the use of the test rig. The assumptions are verified by the numerical solution of the Reynolds equation by the FVM method.

Keywords: journal bearings, Reynolds equation, fluid forces, Muszynska model

1 Introduction
An analysis of the behaviour of active vibration control systems requires describing a controlled system with the use of the linear equations that are suitable for the calculation of the system transfer functions and the controller adjustment [1]. For this purpose, there is a sufficient model designed by Muszynska [2] because it allows deriving linear transmission functions that are useful for analyzing the control circuit and calculating the bearing radial stiffness. The disadvantage of this model is that the parameters of the Muszynska model cannot be calculated from bearing dimensions and lubricant properties. This analysis, however, is aimed at the calculation of the stiffness and damping matrices of the motion equation with the use of the Reynolds equation to estimate the behaviour of the journal bearing at the extra high rotational speed. Instead of numerical integration of the Reynolds equation, the derivation of the formulas to calculate the model parameters is used. For verification of the numerical solutions of the full three-dimensional fluid flow model (Navier Stokes equations) using finite volume method is applied.

2 Instrumentations
As has been said, we are interested in high-speed hydrodynamic journal bearings. We have a test rig with a rotor on two sliding bearings with these parameters: The span of bearing pedestals is of 200 mm, the journal diameter is of 30 mm, the radial clearance is of 60 µm, and the length-to-diameter ratio is equal to about 0.77. The rotor drives an induction motor that is powered by a frequency converter up to 400 Hz so that maximum speed can be up to 23,000 RPM. A sketch of the test rig is shown in Figure 1.
The test rig was completed for tests of actively controlled bearings with the use of piezoactuators. However, this article focuses only on the function of conventional sliding bearings, namely the movement of the bearing journal in the bushing. For the experimentation, proximity sensors and a rotational speed sensor are the most important. The installation of the sensors is shown in Figure 2. During the tests, the oil temperature at the bearing outlet can be recorded. Run-up and coast-down can take up to two minutes, and therefore these variables do not change. The position of the journal is measured by a pair of the proximity probes which are capacitive sensors originated from the Micro-epsilon Company. The sensors are of the capaNCDT CS05 type with a measurement range of 0.5 mm. Measurement error is less than 1 micron. An advantage of the capacitive sensors is that it is not necessary to ground the shaft. Previously, we used the sensors based on the eddy current principle. The error of these sensors was 10 times greater. The position of the bearing journal is filtered by the Kalman filter. The covariance of the measurement error corresponds to the sensor error. The rotor rotational speed is evaluated from a tacho-signal in the form of the pulse string. The pulses are generated by the laser sensor of the VLS Series (Optical Speed Sensor) type with the measurement range up to 250,000 RPM.

3 Experiments

Special oil for high-speed spindle bearing of the OL-P03 type was used for testing (VG 10 grade, viscosity \( \mu = 0.027 \text{ Pa.s} @ 200 \text{ C} \)). Tests were carried out without preheating the lubricant at a normal temperature. The journal bearing cross-section is shown in Figure 3.

The operating conditions of the hydrodynamic bearing are described by the Sommerfeld number [3]

\[
S = (R/c)^2 \mu N/P,
\]

where 
\( N \) – a rotational speed of the rotor in rev/s, 
\( \mu \) – a dynamic viscosity in Pa.s, 
\( R \) – a radius of the journal, 
\( c \) – a radial clearance, 
\( P \) – a load per unit of projected bearing area (2RL), 
\( L \) – a bearing length.

The value of the Sommerfeld number for the given bearing size and the rotor mass of 0.83 kg is as follows \( S = 0.014 \times N \), where \( N \) is a rotational speed of the rotor in rev/s.
The magnitude of friction coefficient in the plain bearings was analyzed in the past by the McKee brothers [4]. It has been found that bearing friction is dependent on a dimensionless characteristic given by a ratio $\mu N/P$ whose parameters are defined above. If the rotor does not rotate or rotate slowly, there is only a very thin-film between the journal and the bushing. Boundary, or thin-film, unstable lubrication occurs with a considerably increased coefficient of friction. Many experiments show that the journal axis moves chaotically at low speeds or the journal starts to oscillate. Only when the specified speed limit is exceeded the lubrication becomes stable, and thick-film of the lubricant is formed. The limit value of the bearing characteristic for the boundary lubrication is described in the Budynas handbook [3]. Designers keep value $\mu N/P \geq 1.7 \times 10^{-6}$ (reyn x rev/s/psi), which is about five times the value the McKee brothers have determined. The measurement in our test rig shows the limit of the unstable lubrication at about 1,000 RPM, which corresponds to the value of the dimensionless characteristic $\mu N/P$ equalled to $3.8 \times 10^{-5}$ (Pa.s $\times$ rev/s/Pa) when using SI units for input parameters. Our estimate for the lower limit of stable lubrication corresponds to the recommendations in the handbook [3]. In active control experiments, the feedback is closed for stable lubrication.

The coefficient of friction decreases from a considerable initial value. Exceeding the speed limit leads to stable lubrication with a low coefficient of friction because the changes are self-correcting. By the viscous friction theory, the coefficient of friction is proportional to the speed of rotation as is shown in Figure 4.

An example of a gradual change of position of the bearing journal centre during an increase in speed up to 5,000 RPM at the constant increase rate is shown in Figure 5. The lubrication is unstable in the range up to about 1,200 RPM and is accompanied by oscillations. The reason for the oscillations is the step change of speed to about 300 RPM after switching on because it is not possible to increase the rotational speed continuously from zero. Hydrodynamic stable lubrication at stable motion is produced for rotational speed up to 5,000 RPM. Motion instability of the whirl type occurs when this speed of 5,000 RPM is exceeded. Fluid force makes sense to be modelled just for stable motion and lubrication.

![Figure 5 – The run-up of a journal bearing](image)

Notice that the centre of the journal rises to the level of the centre of the bearing bore and gradually approaches this centre so that the small eccentricity gradually decreases to zero as is shown on the bottom panel of Figure 6. The data for this orbit was approximated by the 5-degree polynomial in the time interval which begins at the 3rd second and ends at the 12th second. The difference between thin and thick film lubrication is also evident on the upper panel of Figure 6, which depicts an orbit plot for the entire measurement time.
4 Mathematical model

4.1 Equations of motion

The bearing journal can be considered as a rigid body rotating within the bearing housing at an angular velocity $\Omega$. For simplicity, it is assumed that the rotation axis does not change its direction. Fluid forces are caused by the hydrodynamic pressure generated in the oil film, whose total mass relative to the journal and rotor is negligible. The oil pumped by the rotating journal surface produces an oil wedge that lifts up the bearing journal so that it does not touch the inner walls of the housing. The coordinate system of a cylindrical journal bearing is shown on the left side in Figure 7. The planar motion of the bearing journal at the $x$ and $y$ coordinates can be described by two motion equations arranged into a matrix equation

$$
\begin{bmatrix}
M & 0 \\
0 & M
\end{bmatrix}
\begin{bmatrix}
x''(t) \\
y''(t)
\end{bmatrix}
+ 
\begin{bmatrix}
B_{XX} & B_{XY} \\
B_{YX} & B_{YY}
\end{bmatrix}
\begin{bmatrix}
x'(t) \\
y'(t)
\end{bmatrix}
+ 
\begin{bmatrix}
C_{XX} & C_{XY} \\
C_{YX} & C_{YY}
\end{bmatrix}
\begin{bmatrix}
x(t) \\
y(t)
\end{bmatrix}
= 
\begin{bmatrix}
F_X(t) \\
F_Y(t)
\end{bmatrix},
$$

(2)

where

- $M$ – a mass of the rotor,
- $F_X$ – a force acting on the journal in the horizontal direction,
- $F_Y$ – a force acting on the journal in the vertical direction,
- $C_{UV}$ – a stiffness coefficient,
- $B_{UV}$ – a damping coefficient.

In addition to force components in the horizontal and vertical directions, the force balance will be solved in other possible directions. Force in the direction of the line of the centres is denoted as a direct force $F_D$ while force which is perpendicular to the line of centres is denoted as a quadrature force $F_Q$. Both these forces balance the gravity force $G$ as is shown in Figure 7. The system is described by two motion equations, and therefore the total order of the system is four. This system may become unstable even for positive values of physical parameters.

![Figure 7 – A cross-section of a hydrodynamic bearing](image)

4.2 Muszynska model

The motion equation of the rotor with the journal bearing in coordinates $x$ and $y$ was designed by Muszynska. The derivation is based on the design of the formula to calculate the already mentioned direct and quadrature forces. Compared to Eq. (2), the stiffness and damping matrices are designed in such a way that the oil film is replaced by a spring and a dashpot system that rotates at an angular velocity $\Omega$, where $\lambda$ is a dimensionless parameter, which is slightly less than 0.5. The stiffness of the spring is designated by $K$ and the damper has a damping factor $D$

$$
\begin{bmatrix}
M & 0 \\
0 & M
\end{bmatrix}
\begin{bmatrix}
x''(t) \\
y''(t)
\end{bmatrix}
+ 
\begin{bmatrix}
D & 0 \\
0 & D
\end{bmatrix}
\begin{bmatrix}
x'(t) \\
y'(t)
\end{bmatrix}
+ 
\begin{bmatrix}
K & D\lambda\Omega \\
-D\lambda\Omega & K
\end{bmatrix}
\begin{bmatrix}
x(t) \\
y(t)
\end{bmatrix}
= 
\begin{bmatrix}
F_X(t) \\
F_Y(t)
\end{bmatrix}.
$$

(3)

The derivation of the motion equation is described, for example, in [2]. No analytical way to calculate the value of unknown model parameters has been proposed. These parameters can be estimated only from a comparison of measurement and simulation.
4.3 Analytical solution of the Reynolds equation

The theory of hydrodynamic bearing is based on a differential equation derived by Osborne Reynolds. Reynolds equation is based on the following assumptions: The lubricant obeys Newton’s law of viscosity and is incompressible. The inertia forces of the oil film are negligible. The viscosity \( \mu \) of the lubricant is constant, and there is a continuous supply of lubricant. The effect of the curvature of the film concerning film thickness is neglected. It is assumed that the film is so thin that the pressure is constant across the film thickness. The shaft and bearing are rigid. Furthermore, it is assumed that the thickness of the oil film depends on the other two coordinates, namely the coordinate \( z \) along the axis of rotation and the location on the perimeter of the journal which is described by the angle \( \theta \) as is shown on the right side in Figure 7. If the radius of the bearing journal is equal to \( R \), then the most general version of the Reynolds equation for calculation of the oil pressure distribution is as follows [5]

\[
\frac{1}{R^2 \partial \theta} \left( h^3 \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial z} \left( h^3 \frac{\partial p}{\partial z} \right) = 6\mu \Omega \frac{\partial h}{\partial \theta} + 12\mu \frac{\partial h}{\partial t}.
\]

(4)

There is no analytical solution for the Reynolds equation. During operation, the journal axis shifts from the centre of the bearing bushing to the distance of \( e \), called eccentricity, which is related to a radial clearance \( c \). Variable is called an eccentricity ratio \( n = e/c \). The film thickness as a function of \( \theta \) is defined as follows \( h = c(1 + n \cos \theta) \). The oil film moves in adjacent parallel layers at different speeds, and shear stress results between them. The oil layer at the surface of the journal moves at the peripheral velocity of the journal while the oil layers at the surface of the bearing bushing don’t move (at zero velocity). The surface of the journal moves at a velocity of \( U = R \Omega \). Reynolds equation will be solved for the steady state and independence of the pressure distribution on the coordinate of \( z \)

\[
\frac{d}{d\theta} \left[ h^3 \frac{dp}{d\theta} \right] = 6\mu UR \frac{dh}{d\theta}.
\]

(5)

On double integrating using the integration constants \( K_1, K_2 \), and \( p_0 \), see [5], we get

\[
p(\theta) = \frac{6\mu UR}{c^2} \int \left[ \frac{d\theta}{(1+n \cos \theta)^2} + \frac{K_1 d\theta}{c(1+n \cos \theta)^3} \right] + p_0.
\]

(6)

The solution must meet the boundary condition \( p(\theta = 0) = p(\theta = 2\pi) = 0 \), which gives

\[
\frac{6\mu UR}{c^2} \int_{\theta = 0}^{\theta = 2\pi} \left[ \frac{1}{(1+n \cos \theta)^2} + \frac{K_1}{c(1+n \cos \theta)^3} \right] d\theta = 0 \Rightarrow \frac{K_1}{c} = \frac{\int_{\theta = 0}^{\theta = 2\pi} 1/(1+n \cos \theta)^3 d\theta}{\int_{\theta = 0}^{\theta = 2\pi} 1/(1+n \cos \theta)^2 d\theta}.
\]

(7)

On simplifying, we get a formula for calculating the first integration constant \( K_1 \) equal to

\[
K_1 = 2c(2n^2 - 1)/(n^2 + 2).
\]

Extreme oil pressure values as a function of attitude angle \( \theta \) are achieved if \( dp/d\theta = 0 \) at \( K_1 = -h = -c(1 + n \cos \theta) \). The first integration constant is related to the thickness of the oil film at the perimeter of the journal, where the maximum and minimum oil pressure is achieved

\[
h_m = (h)_{p=\text{min}} = (h)_{p=\text{max}} = -K_1 = 2c(1 - n^2)/(n^2 + 2).
\]

(8)

The attitude angle where the maximum and minimum pressure occur is given by \( \cos \theta_m = -3n/(n^2 + 2) \). The formula for calculating of the pressure distribution is as follows

\[
p(\theta) = \frac{6\mu UR}{c^2} \left[ \frac{n(2+n \cos \theta)}{(n^2+2)(1+n \cos \theta)^2} \right] + p_0 = \frac{6\mu UR}{c^2} \beta(\theta, n) + p_0.
\]

(9)

The first integration constant was selected to meet the mentioned boundary condition. The oil pressure distribution on the journal for \( n = 0, 0.1, 0.2, \ldots, 0.9 \) is shown in Figure 8. Note that the integration constant \( p_0 \) has not any effect on the force excited by the oil pressure.
4.4 Fluid force

The forces acting on the journal in the centre of gravity along the bearing length of $L$ can be calculated for the direction of the line of the centres and the perpendicular direction. Force in the direction of the line of centres is denoted as a direct force $F_D$ while force which is perpendicular to the line of centers is denoted as a quadrature force $F_Q$. Both these forces balance the gravity force $G$ as is shown in Figure 7.

\[ F_D = \int_{0}^{2\pi} p_{\theta} \cos(\pi - \theta) \, d\theta = \frac{6\mu U R^2 L}{c^2} \int_{0}^{2\pi} \beta(\theta, n) \cos(\pi - \theta) \, d\theta = F_0 \beta_D(n) \]
\[ F_Q = \int_{0}^{2\pi} p_{\theta} \sin(\pi - \theta) \, d\theta = \frac{6\mu U R^2 L}{c^2} \int_{0}^{2\pi} \beta(\theta, n) \sin(\pi - \theta) \, d\theta = F_0 \beta_Q(n). \] (10)

where \[ F_0 = 6\mu UR^2L/c^2 = 6\pi SG \] – a force factor.

Note that according to formula (9) the pressure on the part of the journal surface is negative, which is, in fact, a relative negative pressure. Since the pressure distribution is anti-symmetric, without mathematical evidence, it is clear that these formulas can be applied. Only quadrature force $F_Q > 0$ acts on the bearing journal and the direct force is zero $F_D = 0$, as is shown on the upper panel in Figure 9. For speeds ranging from 1250 to 5000 RPM, the force factor $F_0$ varies from 0.46 to 1.8 kN. However, this force is reduced by multiplying the eccentricity ratio $n$ in the range of one thousandth (0.001) to their fractions (0.0004). The balance of forces $F_D$, $F_Q$, and $G$ allows to calculate an attitude angle $\alpha = \arctan(\beta_D(n)/\beta_Q(n))$, see the right panel in Figure 4.

The presence of direct force can be explained, e.g. by the cavitation or the inability to achieve high vacuum, but the mathematical model is more complicated [6]. The lubricant flows through the bearing, but in the part of the bearing journal circumference where the pressure is below the barometric pressure, the lubricant can also be sucked. The magnitude of the negative pressure for $\pi < \theta < 2\pi$ is multiplied by a factor $\gamma$. Therefore the total force is given by the sum of integrals (10) as follows
\[ \frac{\bar{F}_D}{F_Q} = \int_{0}^{\pi} (...) \, d\theta + \gamma \int_{\pi}^{2\pi} (...) \, d\theta, \] (11)

The effect of negative pressure reduction is demonstrated in Figure 9. Negative pressure is limited to 1% of the magnitude of positive pressure for the angle interval of $0 < \theta < \pi$. The formulas for the calculation of the quadrature and direct forces contain the same force factor $F_0$ and hence the dependence on the peripheral speed $U$ and therefore on the rotor angular velocity. The coefficients $\beta_Q(n)$ and $\beta_D(n)$ differ considerably. The diagrams confirm the linearity of the quadrature and direct force to eccentricity ratio up to 0.6. The $\beta_Q(n)$ and $\beta_D(n)$ coefficients can be approximated in this range as a linear function
\[ \beta_Q(n) \approx qcn = qe \]
\[ \beta_D(n) \approx dcn = de, \] (12)

where $q$ – determines the quadrature stiffness $C_Q = 6\mu UR^2 L/c^2 \times q$,
\[ d \] – determines the direct stiffness $C_D = 6\mu UR^2 L/c^2 \times d$. 

![Figure 8 – Pressure distribution along the angular coordinate](image-url)
The stiffness in the directions of the Cartesian coordinates $x, y$, and the attitude angle $\alpha$ which is defined in Figure 2 can be obtained by substitution
\[
x(t) = -e \sin \alpha \\
y(t) = +e \cos \alpha.
\] (13)

The vector of the direct and quadrature forces depends on the coordinates $x, y$ according to the following formula
\[
\begin{bmatrix}
-C_D e \sin \alpha + C_Q e \cos \alpha \\
C_Q e \sin \alpha + C_D e \cos \alpha
\end{bmatrix}
= \begin{bmatrix} C_D x(t) + C_Q y(t) \\
-C_Q x(t) + C_D y(t) \end{bmatrix}
= \begin{bmatrix} C_D & C_Q \\
-C_Q & C_D \end{bmatrix}
\begin{bmatrix} x(t) \\
y(t) \end{bmatrix}.
\] (14)

The cross-coupled stiffness $D\lambda\Omega$ according to the Muszynska model corresponds to the expression $6\mu UR^2L/c^2q$. The direct stiffness $K$ is orderly less than the cross-coupled stiffness; however, the analytical calculation of the stiffness matrix shows the dependence on the rotational speed.

The damping matrix can be derived based on its relationship to the stiffness matrix according to the model that was designed by Muszynska. The motion equation for the rigid rotor in the plain bearing is as follows
\[
\begin{bmatrix} M & 0 \\
0 & M \end{bmatrix}
\begin{bmatrix} \ddot{x}(t) \\
\ddot{y}(t) \end{bmatrix}
+ \begin{bmatrix} C_Q/\Lambda\Omega & 0 \\
0 & C_Q/\Lambda\Omega \end{bmatrix}
\begin{bmatrix} \dot{x}(t) \\
\dot{y}(t) \end{bmatrix}
+ \begin{bmatrix} C_D & 0 \\
-C_Q & C_D \end{bmatrix}
\begin{bmatrix} x(t) \\
y(t) \end{bmatrix}
= \begin{bmatrix} F_x(t) \\
F_y(t) \end{bmatrix}.
\] (15)

The sum of direct and quadrature forces must compensate for the gravitational force that does not depend on the rotational speed. The suitability of this model is confirmed by Cavalca [7].

### 4.5 Force balance

As shown in Figure 2, three static forces acting on the bearing journal, namely direct force $F_D$, quadrature force $F_Q$, and gravitational force $G$. The following equation is a condition of equilibrium of these forces $G^2 = F_D^2 + F_Q^2$. On substituting, we get
\[
(\beta_D(n))^2 + (\beta_Q(n))^2 = 1/(6\pi S)^2.
\] (16)

The expression on the right side of the previous formula can be calculated from the Sommerfeld number $S$ for a given rotation speed $N$, and the load $P$ per unit of projected bearing area. Since both coefficients $\beta_Q(n)$ and $\beta_D(n)$ depend on the eccentricity ratio $n$, this unknown quantity can be determined just like the attitude angle $\alpha$, see Figure 10, and the journal centre coordinates as a function of the speed of rotation.
5 Numerical solution of the Reynolds equation

The basic prerequisite for deriving a simplified fluid force calculation was the existence of cavitation or the inability to achieve the high vacuum due to the sucking out of lubricant from the outside of the bearing. This assumption has been verified by the numerical solution of the general Reynolds equation (4), including the assumption of pressure compliance with barometric pressure at the edge of the bearing bushing. In order for the input data for the calculation to be realistic, the actual eccentricity for a given speed was deducted from the measurement in Figure 5. Due to the rotational speed and flow velocity then the Reynolds number was determined for the flow in the gap, the value of which was very low. The flow was classified as laminar. A significant result was the distribution of pressure on the bushing walls and the bearing journal. Since the minimum pressure is approaching the vacuum pressure (zero absolute pressure), the flow does not show cavitation. The results of the simulation calculations confirm that the gravity force compensates for the quadrature force, which is many times larger than the direct force.

6 Conclusions

The calculation of the fluid force in the steady-state acting on the bearing journal using the Reynolds equation shows that the stiffness matrix corresponds to the stiffness matrix according to the Muszynska model. This finding can only be applied for a rotational speed range where lubrication is stable, which is of course not at zero or low starting speeds. From the Reynolds equation, unlike the Muszynska model, it follows that the elements on the main diagonal of the stiffness matrix are also dependent on the speed of rotation. The conclusions apply only to the range of magnitudes in which the formulas can be linearized. The numerical solution of the Reynolds equation confirms the occurrence of cavitations. The theoretical models are always unreliable and require comparative measurement on the test rig.

7 References


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Study on a Helmholtz Type Hydraulic Silencer with Variable Resonance Mechanism

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Abstract: Pressure pulsation is caused by the flow ripples from a positive displacement hydraulic pump. It transmits throughout fluid power equipment and causes unwanted excitation of the mechanical parts. In many practical applications, a Helmholtz type hydraulic silencer may be used to attenuate such pulsation. It is the preferred solution on account of its simple structure and high attenuation performance. However, the distinctive drawback of this silencer is that it is effective only within a narrow range of the attenuating frequency. Therefore, the silencer is only suitable for use in the hydraulic systems running at constant pump rotational speeds. The purpose of this research is to develop a novel silencer for hydraulic systems that have a fixed displacement pump driven under variable speed. Firstly, a mechanism for adjusting the resonant frequency has been proposed. This works by changing the volume of the silencer. Secondly, a prototype of the novel silencer was designed using a new distributed parameter system model, in which the vessel volume was modelled as an annular fluid line. Finally, the attenuation characteristics of the prototype silencer were verified by measuring the transmission loss and insertion loss under various conditions, and comparing them with the calculated results.

Keywords: Helmholtz type hydraulic silencer, pressure ripple, displacement pump, variable rotational speed, transmission loss

1 Introduction

The flow ripple from the positive displacement pump generates pressure pulsation due to mutual interference with pipelines, valves, hydraulic actuators and so on. It may cause vibration and noise of hydraulic equipment. One way to attenuate the pulsation is to install a silencer in the hydraulic circuit. The Helmholtz type hydraulic silencer is often used in mobile and industrial machineries because of simple structure and high damping effect of the pulsation.

On the other hand, an energy-saving type hydraulic system that controls the pump rotational speed by an inverter or a servo motor has attracted attention instead of the variable displacement type pump. Since the Helmholtz type hydraulic silencer utilizes the resonance phenomenon, it is only in the vicinity of the resonance frequency that the damping effect can be expected. Therefore, there is a disadvantage that a sufficient damping effect cannot be obtained for a hydraulic system in which the rotational speed of the pump fluctuates.

In acoustic Helmholtz type resonators, many proposals have been reported in which the resonance frequency is matched with the frequency by mechanically manipulating the volume of the torso and the cross sectional area of the throat even when the frequency of the noise changes[1]-[3]. However, the variable resonance mechanism for these acoustics can not be
directly applied to the hydraulic system from several viewpoints. Our group has been experimentally and theoretically studied influences of the dimensional configuration on the attenuation characteristics of Helmholtz type silencers\cite{4}-\cite{7}.

The purpose of this research is to develop and academically study a novel silencer for hydraulic systems that have a fixed displacement pump driven under variable speed. Firstly, a mechanism for adjusting the resonant frequency has been proposed. This works by changing the volume capacity of the silencer. Secondly, a prototype of the novel silencer is designed using a new distributed parameter system model, in which the volume capacity in a vessel is modelled as an annular fluid line. Finally, the attenuation characteristics of the prototype silencer are verified by measuring the transmission loss under various conditions, and comparing them with the calculated results.

2 Helmholtz Silencer with Variable Resonance Mechanism

2.1 Silence structure

The Helmholtz type hydraulic silencer is basically composed of a vibration system based on mass and spring. In the lumped model, the mass and spring correspond with hydraulic oil in the neck and compressibility of the vessel volume respectively. If this resonance frequency is designed to match the fundamental frequency of the pressure ripple from the displacement pump, it can be effectively attenuated. The fundamental frequency $f_f$ is defined as a product using the pump element number $z$ and rotational speed $N$.

$$f_f = \frac{zN}{60}$$

(1)

As shown in Figure 1, the structure of the proposed Helmholtz type hydraulic silencer consists of two volumes and two necks. The volume [1] is connected to the main line through the neck [1]. The length $L_1$ of volume [1] can be adjusted by moving the piston. The subscript $k$ ($k=1, 2$) of the length $L$ and diameter $d$ is corresponding to [1] and [2] of the volumes and necks, respectively. In order to minimize the load resistance force for sliding, a neck with small diameter, neck [2], is placed between the volume [1] and volume [2]. This acts to nearly equalize the pressure on each side of the piston. The rod packing and piston packing seal the external and internal leakages respectively. This silencer seems like a two-stage Helmholtz type hydraulic silencer having two neck portions and two capacity portions, so that it has two resonance frequencies. However, only one of them can set the volume of the capacity part to a desired value. Therefore, this paper describes a single-stage silencer that can vary only the resonance frequency at the neck [1] and volume [1] by adjusting the volume length [1].

![Figure 1- Structure of Helmholtz type hydraulic silencer](image)

Figure 1- Structure of Helmholtz type hydraulic silencer
In order to design the dimensions of a Helmholtz type hydraulic silencer (Figure 1) having a variable resonance mechanism, this section shows a mathematical model dealing with the capacity part as an annular conduit. This damping characteristic is evaluated by transmission loss $TL$ which is a characteristic value peculiar to the silencer. The transfer matrix $T_{V,k}$ of the capacitive part adopts the following equation which is obtained by analyzing the axial plane wave motion theory under the boundary condition of the annular conduit\[^\text{8}\].

$$T_{V,k} = \begin{bmatrix}
\cosh\left(\frac{G_k s}{c_a} L_k\right) & Z_{V,k} \sinh\left(\frac{G_k s}{c_a} L_k\right) \\
\frac{1}{Z_{V,k}} \sinh\left(\frac{G_k s}{c_a} L_k\right) & \cosh\left(\frac{G_k s}{c_a} L_k\right)
\end{bmatrix} \tag{2}
$$

Where, the suffix $k$ is $k = 1, 2$ for the capacitive parts [1], [2]. In addition, $s$ is the Laplace operator, and $L_k$ is the length of the capacitance section. In the annular conduit, $Z_{V,k}$ and $G_k$ are complex coefficients representing the characteristic impedance and the transient viscous resistance, respectively. These parameters are expressed as follows using the density $\rho$ of the hydraulic fluid, the inner diameter $D$ of the cylinder tube and the piston rod diameter $d_r$.

$$Z_{V,k} = \frac{4 \rho c_s G_k}{\pi (D^2 - d_r^2)} \tag{3}$$

$$G_k = 1 + \frac{1}{(1-m)^2} + \frac{3}{2(1-m)^3} \sigma^2 - \frac{1-2m + m^2}{8m(1-m)^2} \sigma^3 \tag{4}$$

The sound speed $c_a$ in Eqs.(2) and (3) in the annular oil conduit is represented by the bulk modulus $K$ of the hydraulic fluid, the sound speed $c$ in the oil in the rigid pipeline, the wall thickness $h$ of the tube, the modulus of longitudinal elasticity $E$ and the Poisson's ratio $\eta$.

$$c_a = c \left[ 1 - \frac{K}{E} \frac{m^2}{1-m} \left( \frac{D}{2h} + 1 - \eta \right) \right] \tag{5}$$

Where, $m$ is the diameter ratio between the piston rod and the cylinder tube, and the coefficient $\sigma$ can be expressed using kinematic viscosity $\nu$.

$$m = \frac{d_r}{D} \tag{6}$$

$$\sigma = \frac{D}{2} \sqrt{\frac{s}{\nu}} \tag{7}$$

On the other hand, plane wave theory in a rigid pipe is applied to the neck portion. Transmission matrix $T_{N,k}$ of the neck portion is formed by connecting the length $L_k$ of the capacity portion to the length $l_k$ of the neck portion in the equation (2), the characteristic impedance $Z_{V,k}$ of the annular conduit, the complex representing the unsteady viscous resistance in the annular conduit. It is obtained by substituting the coefficient $G_k$ with the characteristic impedance $Z_{N,k}$ and the complex coefficient $\zeta_k$ expressing the transient viscous resistance in the rigid pipe line\[^\text{9}\].

$$Z_{N,k} = \frac{4 \rho c_s \xi_k}{\pi d_r^2} \tag{8}$$

$$\xi_k \approx 1 + \sqrt{\frac{4v}{d_r^2 s}} + \frac{4v}{d_r^2 s} \tag{9}$$
Where \( d_k \) is the diameter of the neck parts \([1], [2]\).

Figure 2 shows the transfer matrices on the silencer. The transfer matrices of the two volumes \([1], [2]\) and two necks \([1], [2]\) are respectively denoted by \( T_{V,1}, T_{V,2} \) and \( T_{N,1}, T_{N,2} \). Consequently, the transfer characteristic \( T_S \) is described by the dot product of these transfer matrices as shown in the following equation.

\[
T_S = T_{V,1} \cdot T_{V,2} \cdot T_{N,1} \cdot T_{N,2}
\]

Putting the pressure pulsations \( P_{in}, P_{out} \) and flow ripples \( Q_{in}, Q_{out} \) in the entrance and exit of the silencer respectively, the relationship with the transfer characteristic is given as follows.

\[
\begin{bmatrix}
P_{in} \\
Q_{in}
\end{bmatrix} = T_S \begin{bmatrix}
P_{out} \\
Q_{out}
\end{bmatrix}
\]

Since the Helmholtz type hydraulic silencer is branched and connected to the main pipe, the transmission loss \( TL \) is expressed as follows using the matrix component of \( T_S \).

\[
TL = 20 \log_{10} \left( \frac{1}{2} \frac{1 + Z_m T_{S,21}}{1 + Z_m T_{S,11}} \right)
\]

Where, \( Z_m \) is the characteristic impedance of the main pipe and can be obtained by replacing the diameter \( d_k \) of the neck portion with the inner diameter \( d_0 \) of the main pipe in the equation (8).

![Figure 2- Transfer matrices on silencer](image)

### 2.2 Determination of dimensions of prototype silencer

In this study, we assume an external gear pump with a number of teeth \( z = 10 \), which changes the pulsation source of the hydraulic system to be damped at \( N = 1440 \) to 2400 min\(^{-1}\). Therefore, the resonance frequency \( f_r \) of the silencer should be chosen as \( f_r = f_2 = 240 \) to 400 Hz as shown in Eq.(1). As described above, since this single stage silencer has the capacity parts \([1] and [2]\) and the neck parts \([1] and [2]\), it is actually a two-stage Helmholtz type hydraulic silencer, it therefore has two resonance frequencies. When there are two resonance frequencies, it is difficult to design the silencer so as to satisfy the resonance frequency range \( (f_r = 240 \text{ to } 400 \text{ Hz}) \) due to the interaction with each other. Therefore, by making the diameter \( d_2 \) of the neck part \([2]\) as close to zero as possible, the mutual influence by the two resonance frequencies must be reduced.

It is known that the transmission loss value \( TL_r \) at the resonance frequency in the experiment is generally lower than the value calculated by the mathematical model\(^{10}\). This is due to the modelling error that arises because it does not consider the loss at the neck of the silencer. Therefore, it is assumed that the transmission loss is about 70% of the calculated value by the
mathematical model. From the above discussion, the transmission loss value at the resonance frequency is determined to be $TL_r = 22\text{dB}$. Each part dimension was selected from a double-acting hydraulic cylinder in the previous JIS (Japanese Industrial Standards) of which is conventionally used in Japan as much as possible. The parameters shown in Table 1 are used on the calculation of the damping characteristics. Table 1 also shows the dimensional specification of the prototype silencer which finally determined by Eqs. (2) to (12).

Table 1- Specifications of numerical parameter and prototype silencer dimension

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>$\rho = 875 \text{kg/m}^3$</td>
</tr>
<tr>
<td>Kinematic viscosity</td>
<td>$\nu = 3.20 \times 10^{-2} \text{m}^2/\text{s}$</td>
</tr>
<tr>
<td>Speed</td>
<td>$e = 1380 \text{m/s}$</td>
</tr>
<tr>
<td>Diameter of main line</td>
<td>$d_o = 21.0 \text{mm}$</td>
</tr>
<tr>
<td>Bulk modulus</td>
<td>$K = 1.66 \text{GPa}$</td>
</tr>
<tr>
<td>Modulus of longitudinal elasticity</td>
<td>$E = 206 \text{GPa}$</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>$\nu = 0.3$</td>
</tr>
<tr>
<td>Diameter of neck I</td>
<td>$d_1 = 10 \text{mm}$</td>
</tr>
<tr>
<td>Length of neck I</td>
<td>$l_1 = 90 \text{mm}$</td>
</tr>
<tr>
<td>Diameter of neck II</td>
<td>$d_2 = 2 \text{mm}$</td>
</tr>
<tr>
<td>Length of neck II</td>
<td>$l_2 = 50 \text{mm}$</td>
</tr>
<tr>
<td>Length of front and rear cover</td>
<td>$L_r = 47 \text{mm}$</td>
</tr>
<tr>
<td>Length of volume I</td>
<td>$L_i = 80-242 \text{mm}$</td>
</tr>
<tr>
<td>Diameter of tube</td>
<td>$D = 80 \text{mm}$</td>
</tr>
<tr>
<td>Length of tube</td>
<td>$L_t = 250 \text{mm}$</td>
</tr>
<tr>
<td>Length between front cover and neck I</td>
<td>$L_{1} = 21.5 \text{mm}$</td>
</tr>
<tr>
<td>Thickness of tube</td>
<td>$h = 7 \text{mm}$</td>
</tr>
</tbody>
</table>

3 Experimental Apparatus

Figure 3(a) shows the schematic diagram of variable resonance mechanism. The mechanism consists of the pump rotational speed detector, A/D and D/A converters, PC, linear actuator and displacement sensor. The target length $L_o$ of the volume [1] is determined from the detected rotation speed $N$ of the pump. It is experimentally obtained by the relationship between the length $L_1$ of the volume [1] and the resonance frequency $f_r$ as shown in Figure 5(a) described next chapter. The displacement of the piston is adjusted by a linear actuator connected to the piston rod, and then controlled by performing the position feedback so that the volume length $L_1$ becomes the target length $L_o$. The specifications of the linear actuator are estimated from the maximum speed $v_p$ of the piston and the load force generated on the piston rod. Since the movable range of the volume length is about 200mm, the maximum speed of the piston is $v_p = \pm 50 \text{mm/s}$. The direction is defined as positive, when the volume length $L_1$ increases.

Figure 3(b) shows the overview of experimental apparatus. The hydraulic oil from the gear pump is delivered to the main line which is connected to the prototype silencer. The setting pressure of the main line is adjusted by the pressure relief valve and the oil temperature is kept at $40\pm 1\text{°C}$ by the oil cooler. A piezoelectric type pressure sensor (PCB Piezotronics 113B22) is installed downstream of the silencer. The distance from the outlet of the pump to the silencer connection is $L_{ps} = 1750 \text{mm}$, the distance of the pressure sensor from the silencer $L_{ss} = 1245 \text{mm}$ (49 mm from the relief valve to the pump side), the distance of the relief valve from the silencer is $L_{sr} = 1294 \text{mm}$. The signals from the pressure sensor are processed by the PC via the amplifier and A/D converter in order to analyse the attenuation characteristics.

![Figure 3- Variable resonance mechanism](image)
4 Results and Consideration

Figure 4 shows the comparison of the calculated value by the mathematical model with the measurement result by the 4-pressures 2-systems method \(^9\) on the transmission loss \(TL\). When the volume length \(L_1\) of the capacity part \([1]\) is longer \((L_1 = 242 \text{ mm})\) as shown in Figure 4(a), the measurement result of the silencer having the piston rod in the capacity part is in good agreement with the mathematical model of the annular conduit. As shown in this graph, the resonance peak appears in the low frequency region of about 25Hz. This phenomenon is caused by the neck portion \([2]\) provided to minimize the difference in force due to the pressure applied to both sides of the piston. However, it can be confirmed that the peak is suppressed low without affecting the damping characteristics of the silencer, since this diameter \(d_2\) is made small. On the other hand, as the volume length \(L_1\) of the capacitive section \([1]\) decreases \((L_1 = 80 \text{ mm})\), it can be observed that the measured resonance frequency is slightly lower than the calculated value as shown in Figure 4(b).

\[
\begin{align*}
\text{Figure 4- Transmission loss of silencer}
\end{align*}
\]

In order to clarify the tendency of the difference in the resonance frequency as described above, Figure 5 shows the attenuation characteristics of silencer. This figure represents how the resonance frequency \(f_r\) and transmission loss \(TL_r\) at resonance frequency vary with respect to volume length \(L_1\) of the capacitance section \([1]\). As seen in Figure 5(a), although a good agreement is seen at \(L_1 = 242\text{ mm}\), the measurement result of the resonance frequency \(f_r\) slightly decreases with respect to the mathematical model as the volume length \(L_1\) of the capacitance part \([1]\) decreases. In Figure 5(b), the experimental result of the transmission loss \(TL_r\) at the resonance frequency is smaller than the result of the mathematical model, regardless of the volume length \(L_1\) of capacity part. It can be confirmed that \(TL_r\) satisfies 15dB or more as given in design specification at any frequency. This is because the each dimension was determined so that \(TL_r = 22\text{ dB}\) by assuming the actually measured value of the transmission loss \(TL_r\) at the resonance frequency to be about 70% of the mathematical model based on the conventional experience.

\[
\begin{align*}
\text{Figure 5- Attenuation characteristics of silencer}
\end{align*}
\]
This section discusses whether the variable resonance mechanism of the prototype silencer has the desired performance. In order to match the fundamental frequency component $f_f$ of the pressure pulsation with the resonance frequency $f_r$ of the silencer, it is necessary to investigate the target length $L_o$ of the capacity part [1] with respect to $N$ in advance. As clarified in the previous section, when the capacity section length $L_1$ is shorter, the measurement result of the resonance frequency and the mathematical model differed. Therefore, in this report, the target length $L_o$ is obtained by interpolating the plotted points of the measurement result instead of the mathematical model. As shown in Figures 4(b) and 5(a), since the measured resonance frequency of the silencer at $L_1=80\text{mm}$ is $f_r=375\text{Hz}$, it is found that $f_r=400\text{Hz}$ in the original design specification cannot be obtained. Therefore, the variable resonance mechanism will be verified within the frequency range where a good match between the calculated value and the experimental value is obtained. In this paper, the performance of the prototype silencer is measured by changing the pump rotation speed $N$. Figure 6 shows the experimental results of the volume length $L_1$ with reference signal $L_o$ and pump rotational speed $N$. In stages i, iii and v of the constant speed area, the volume length $L_1$ corresponds well with the reference signal. The deviation is within the range of just ±1mm. On the other hand, in stages ii and iv, it is clear that the volume length tracks to the reference signal with almost no time lag.

Figure 6- Experimental results of the volume length $L_1$

Figure 7 shows the time history of pressure pulsation. The left-hand figure (a) shows the measured pressure at the pump rotational speed $N=1440\text{min}^{-1}$ with and without the silencer in the hydraulic circuit. It is found that the pulsation is reduced to about one-third by using the silencer. The right-hand figure (b) shows the measured pressure when the variable resonance mechanism for the silencer is operated and not operated in the hydraulic circuit. When using the variable resonance mechanism, the pump is driven at the rotational speed $N=2160\text{min}^{-1}$. On the other hand, when not using the variable resonance mechanism, the pump is also driven at the rotational speed $N=2160\text{min}^{-1}$, but the volume length is $L_1=242\text{mm}$ which corresponds to $N=1440\text{min}^{-1}$. It is clear that the variable resonance mechanism has an excellent effect.

Figure 7- Time history of pressure pulsation
5 Conclusion

In this report, a Helmholtz type hydraulic silencer having a variable resonance mechanism was developed for a hydraulic system in which the frequency of pressure pulsation varies as the pump rotation speed changes. That is, the silencer has a structure in which the resonance frequency can be matched with the frequency of the pressure pulsation by varying only the length of the capacity portion. The dimensions of each part were determined using a mathematical model of a silencer having an annular shaped capacity part so as to satisfy the required performance. The prototyped silencer was introduced into a hydraulic system whose rotational speed of the pump changes. As a result, it was demonstrated that the damping effect can be kept high by using the variable resonance mechanism. In the next stage, the research area on the practical application will be investigated in the various hydraulic systems.

References

HYDRAULIC COMPONENTS
Development of a Prototype Valve for a Hydro-mechanical Braking-torque Control used on Railway Disc Brakes

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Abstract: Hydraulic disc brakes are one of the most common brake types used in vehicles. In low-floor railway vehicles, they are used due to their compact design, high power density and good dynamic performance. The friction between the brake disc and the brake pads is subjected to various disturbances, which influence the generated braking torque of disc brakes. Usually, there is no closed-loop control of the generated braking torque and few countermeasures are taken to avoid the disturbances. Consequently, the braking effect is affected by those disturbances, which may lead to a degradation of the braking performance. A closed-loop control of the braking torque of disc brakes is subject of an ongoing research project conducted at the Institute for Fluid Power Drives and Systems at RWTH Aachen University. Within the project, a novel hydraulic disc brake system with closed-loop control of the braking torque is under development. During the braking process, the torque on the caliper acts on a supporting rod. It has been found that the resulting supporting force is proportional to the braking torque. Thus, this force is utilized as a torque feedback for a hydro-mechanical control unit. The functionality of the concept for spring-applied hydraulic brakes is described. The braking force of those brakes is generated by a pre-stressed spring and the brake is opened by a cylinder. Therefore, spring-applied brakes are inherently fail-safe and often preferred to active brakes.

In the scope of this paper, a simulation model of a spring-applied disc brake is introduced and the behavior of the conventional spring-applied disc brake is investigated. Based on those results, necessary requirements on the control unit are deduced and a suitable prototype for the investigation on a test rig is developed. Finally, the behavior of the control unit is analyzed using the simulation model.

Keywords: spring-applied disc brake, hydro-mechanical braking-torque control, valve design, railway application

1 Introduction

The friction coefficient in the contact between brake pads and brake disc is subjected to various influences such as temperature, wear on the brake pads or wet or frozen friction contacts [1]. Currently, most commonly open-loop braking systems are in use and only passive countermeasures, such as optimization of the friction materials, are taken [2]. Within a research project at the Institute for Fluid Power Drives and Systems (ifas) regarding a self-energizing hydraulic brake for railway vehicles (SEHB), Petry et. al. introduced a closed-loop control for the braking torque of the SEHB and described the potential and advantages of a closed-loop brake system [3]. From this work, a concept for a closed-loop control of the torque of a conventional brake has been deducted [4]. In a preliminary study, it has been found that the supporting force of the brake caliper is proportional to the braking torque. Thus, it has been proposed to utilize the supporting force as a feedback variable for a
hydro-mechanical control unit. The control unit can be implemented by integrating a hydraulic valve in the supporting rod. On the spool, the supporting force is balanced with a set pressure. As a result, the respective brake pressure is generated. This concept has been further investigated by Bordovsky et al. [5].

So far, all work has been theoretical and only active brake systems have been regarded. Even though, spring-applied brakes are often preferred to active brakes due to their inherent fail-safe behavior. An application of the closed-loop braking torque control for a spring-applied brake is still missing.

This paper applies the concept of a closed-loop control of the braking torque with the supporting force as a feedback variable to a spring-applied disc brake. Furthermore, a prototype valve for an upcoming experimental verification on a test rig and on a reference vehicle is developed.

2 Conventional brake system

In this paper, a novel concept of a hydro-mechanical braking-torque control is analyzed on an existing disc brake HYS 258 by the company Hanning & Kahl GmbH & Co KG (see Figure 1). It is a hydraulic spring-applied brake system with a floating caliper, which is mounted on a bogie via a sliding pivot joint \( B \) and a supporting rod connected to the pivot joint \( A_0 \). Such systems are used in low-floor vehicles as fail-safe brakes since they provide braking power by themselves without any other input energy.

Figure 2 shows a functional schematic of the original brake system. When closing the brake by reducing the braking pressure \( p_B \), the brake pad I is moved against the brake disc. Then, the brake pad I pulls the caliper including the brake pad II against the opposite side of the brake disc, which results in a braking force, respectively a braking torque. This torque acts on the brake disc as well as the caliper and is taken up by the supporting rod.

2.1 Simulation model

A simulation model of the original brake is implemented in the software AMEsim to assess the system behavior. Using a three-dimensional mechanical library, a simplified kinematic model of the brake system is built based on the functional schematic (see Figure 2). The hydraulic system is simplified for the purpose of the simulation. An ideal pressure source is assumed, and a proportional pressure reducing valve, which controls the pressure in the hydraulic actuator, is modelled as a first-order-delay element with an orifice. The valve characteristic is implemented according to the manufacturer data. A pipe connecting the valve to the hydraulic actuator is modelled by considering its resistance and capacitance. The model of the hydraulic actuator is composed of a mass-envelope module, which represents the piston and the caliper masses respectively, a return spring, and a piston with a moving body.
Compared to the similar model presented in the publication [5], this model is extended by a complex braking-torque calculation. In previous work, e.g. [4], it has been assumed that the friction force acting between brake pads and the brake disc acts in the center of their contact area and that there are no additional forces or torques transmitted. However, in reality, friction forces arise over the whole contact area and act in the direction of the local friction speed. Thus, it is necessary to integrate the local friction torque over the whole contact area to evaluate the overall torque. This effect has been described for example by Haag [6]. Though, he disregarded the translational relative movement between the brake disc and brake pads. For the regarded system, such movement is generated by the suspension of the vehicle. Since this has to be taken into account for the investigated brake system, Haag’s formulas are extended by corresponding terms.

Figure 3 illustrates friction speeds acting on a pad-disc contact area for a displaced brake disc related to a fixed brake pad. In any point on this area, the friction speed is composed of the rotational speed component \( \vec{v}_{rot} \) and the translational speed component \( \vec{v}_t \) of the brake disc. The center of the brake disc \( M' \) is displaced from its ideal position \( M_0 \) in the direction of \( x \) and \( z \)-axis by distances \( M'_x \) and \( M'_z \).

The local torque \( t_B \) related to the pivot joint \( B \) can be expressed according to (1)

\[
t_B = p \cdot \frac{\mu}{|\vec{v}_{tot}|} \cdot (l_x v_{tot,x} + l_z v_{tot,z})
\]

where \( p \) is the local contact pressure, \( \mu \) the local friction coefficient, \( l_x \) and \( l_z \) are the local levers related to the point \( B \). Furthermore, \( v_{tot,x} \) and \( v_{tot,z} \) are the local \( x \) and \( z \)-components of the vector field \( \vec{v}_{tot} \), which is composed of the rotational and the translational velocity fields \( \vec{v}_{rot} \) and \( \vec{v}_t \) as follows

\[
\vec{v}_{tot} = \vec{v}_{rot} + \vec{v}_t = -R' \cdot \omega \cdot \left( \frac{\sin \beta}{\cos \beta} \right) + (v_{t,x} \hat{x} + v_{t,z} \hat{z})
\]

with the angle \( \beta \)

\[
\beta = \arctan \left( \frac{R \sin \delta - M'_z}{R \cos \delta + M'_x} \right)
\]

and the local friction radius \( R' \)

\[
R' = \sqrt{(R \sin \delta - M'_z)^2 + (R \cos \delta + M'_x)^2}
\]

The overall torque \( T_B \) related to the pivot joint \( B \) can be calculated by integrating the local torque \( t_B \) over the whole pad-disc contact area.

\[
T_B = \int t_B \, dA = \int \int t_B \cdot R \, dR \, d\delta
\]

The local torque on the brake disc \( t_{M'} \) can be written as
\[ t_{M'} = \frac{\vec{v}_{\text{tot}} \vec{R}}{|\vec{v}_{\text{tot}}|} \mu p R' \]  

with the unit normal vector \( \vec{n} \)

\[ \vec{n} = \left( -\sin \beta, -\cos \beta \right) \]  

The overall torque \( T_{M'} \) can then be found by formula (8).

\[ T_{M'} = \int t_{M'} dA = \int \frac{\vec{v}_{\text{tot}} \vec{R}}{|\vec{v}_{\text{tot}}|} \mu p R' dA \]  

Because no analytical solution could be found for those expressions, they were numerically calculated. Since this could not be done directly in AMEsim, the calculation was implemented using Modelica within AMEsim.

2.2 Behavior of the open-loop system

Main influences and disturbances affecting the braking process are identified using the simulation model. To examine the behavior of the original open-loop system, a parametric study is set up using the Monte-Carlo approach. The following variables are evaluated: the maximum braking torque, the brake reaction time, and the ratio between the braking torque and supporting force (hereinafter referred to as factor \( \alpha \)). The braking torque was normalized with the mean braking torque obtained from the Monte-Carlo run.

Figures 4 – 6 depict results of the parametric study in the form of response surfaces. Two major influences on the maximum braking torque are identified. As expected, the friction coefficient has the main influence on the braking torque (see Figure 4). Furthermore, the braking torque is lowered by a rising clearance \( b \) between brake pads and the brake disc because the spring actuator has to overcome this clearance before generating a braking force. Due to this fact, the reaction time of the system is affected as well, as shown in Figure 5. Note that the axis for \( b \) is shown reversed in order to depict the response surface. Moreover, the oil temperature has a large impact on the reaction time of the system as the maximum flow rates decrease with increasing viscosity at lower fluid temperatures. As can be seen in Figure 6, the factor \( \alpha \) greatly depends on the vertical \( M_z \) disc displacement. It results from the extended calculation of the braking torque as introduced above. Consequently, this finding significantly differs from the results found in previous works [5] and [4].

For the calculation of the friction torques, some assumptions are made. For instance, a uniform pressure is assumed over the pad-disc contact area. In reality, the pressure can be larger in the center of the contact-area because the pads can deform due to the clamping force applied and due to the buckle of the brake pads resulting from the heat generated during braking. Both influences may lead to a redistribution of the local friction forces to the center of the contact area, and hence to a reduction of the influence of the displacements. Since those effects could not be estimated in a satisfying manner, measurements are necessary to evaluate their scope.

![Figure 4 - Response surface of the normalized braking torque \( T_B \) over the clearance \( b \) and friction coefficient \( \mu \)](image1)

![Figure 5 - Response surface of the reaction time \( t_R \) over the clearance \( b \) and oil temperature \( T_{\text{oil}} \)](image2)

![Figure 6 - Response surface of the factor \( \alpha \) over horizontal \( M_x \) and vertical \( M_z \) disc displacements](image3)
3 Novel brake system

3.1 Development of a control concept

The concept of the closed-loop braking-torque control has been introduced in previous works [5] and [4]. Therefore, it is only briefly described here. The supporting force along the supporting rod is utilized as a feedback variable for the closed-loop control. The braking torque is controlled by a hydro-mechanical unit, which represents a prototype valve consisting of spool, sleeve and body. Those parts are moved against each other in order to keep an equilibrium between the set and the actual braking torque.

To realize a closed-loop control, the following sub-functions are necessary. The actual variable has to be fed back, the set variable and the actual variable have to be balanced, and the control variable has to be applied. Within this study, the feedback variable is the supporting force, the set variable is a set pressure and the control variable is the braking pressure, i.e. the pressure in the brake actuator. Since the supporting force can either be a tensile or a compressive force, depending on the direction of the vehicle motion, a rectification as a fourth sub-function is implemented.

Based on this, a mechanical rectification is implemented enabling compression of the spring at compressive and tensile forces. This can be achieved by varying the active areas for the force transmission, as illustrated in Figure 7. In the upper case, a compressive force acts on the structure. Therefore, the left joint acts on the main body and the right joint presses against the central rod to the left. In the lower case, a tensile force is acting. Therefore, the active interfaces change. Now the right joint acts on the main body and the left joint pulls on the central rod to the left. Thus, in this example, the spring is always subjected to compressive stress, independent of the direction of the acting forces.

![Figure 7 – Concept of a mechanical rectification by variation of the active area](image)

Since the analyzed brake system is of a spring-applied type, a lower braking pressure leads to higher braking torques, and thus to higher supporting forces. To realize a force balance on the valve spool, the supporting and the set pressure force are summed up and act against a spring. A disturbance leads to a spool displacement, which results in connection of the brake-actuator port with either the pressure-supply or the return-line port depending on the difference between the set pressure and the supporting force. The principle of the control unit is depicted in Figure 8.
3.2 Parametrization of the control unit

The characteristic between set pressure and normal force of the conventional brake is used for the parametrization of the control valve. For a set pressure $p_{set} = 0$ bar, the brake is completely closed. On the control unit, the spring has to keep the valve completely open to the tank against the maximum possible supporting force. Thus, the spring pre-tension for the maximal spool displacement $y_{max}$ has to be equal or greater than the maximum supporting force.

$$F_{Spring}(y = y_{max}) \geq F_{S,\text{max}}$$  \hfill (9)

Furthermore, the brake is barely closed for a set pressure $p_{set} = p_{th}$ acting on the valve area $A_V$. Therefore, no supporting force is acting and the spool is balanced in its mid-position.

$$F_{Spring}(y = 0) = A_V \cdot p_{th}$$  \hfill (10)

Finally, if the maximum set pressure $p_{set} = p_{max}$ is applied, the valve fully opens to the pressure supply to completely open the brake.

$$F_{Spring}(y = y_{max}) \leq A_V \cdot p_{max}$$  \hfill (11)

When assuming a constant spring stiffness, the necessary spool area $A_V$ can be obtained as follows

$$A_V \geq \frac{F_{S,\text{max}}}{2 \cdot p_{th} - p_{max}}.$$  \hfill (12)

The maximum supporting force $F_{S,\text{max}}$ can be found from a moment equilibrium on the caliper. Subsequently, either a maximum spool displacement $y_{max}$ or a spring stiffness can be chosen. Since standard springs are only available with certain stiffnesses, while the maximum displacement can be chosen rather freely, a suitable spring is selected and the displacement is set according to the spring characteristic. In order to achieve limited displacements of the caliper, a spring with large stiffness has to be found. For this purpose, Belleville springs are selected for this concept because they feature a compact design and a large stiffness.

To achieve a suitable volume flow signal gain for the valve, metering notches have to be machined on the metering edge of the spool. Their geometry and size strongly influence the valve performance. The geometry defines the volume flow signal gain depending on the spool displacement. Typical forms are triangular, rectangular and cylindrical notches. In case of a rectangular notch, the gain linearly rises with the displacement, while cylindrical and triangular notches feature progressive gain characteristics, which allow for a fine metering around the valve mid-position. [7] In this case, the spring significantly restricts spool displacements due to disturbance variables in the control loop, and thus limits the disturbance reaction. To achieve an acceptable disturbance reaction, the gain must not be too low for small displacements. Therefore, rectangular notches are chosen for this application.
Finally, the volume flow signal gain is optimized to achieve a robust control behavior and an acceptable system dynamics. For this purpose, a genetic algorithm is implemented in the software AMESim. The goal of the optimization is to minimize the integral square error (ITSE) of a step response.

3.3 Final design

Figure 9 shows the final design of the control unit. The valve replaces the supporting rod of the original system. On the spool, the supporting force, the set pressure and the force generated by the spring are in balance. Due to the mechanical rectification, the supporting force is always acting in the same direction on the spool. In this picture, it always acts on the right side.

In order to reduce leakage, the spool is sealed with O-rings. Note that components such as fittings, seals, joints and the spring are not shown.

3.4 Behavior of the closed-loop system

A parametric study is conducted also for the closed-loop system to assess its behavior by evaluating the normalized braking torque and the reaction time (see Figure 10 and 11). Again the braking torque was normalized with the mean braking torque of the study.

As can be seen, influences of disturbances on both responses of the closed-loop system changed compared to the open-loop system. The braking torque remains constant for a large range of the friction coefficient.
Based on the influences on the factor $\alpha$, the generated braking torque depends on the vertical displacement of the brake disc. Though, compared to the conventional open-loop system, the braking torque variation is significantly reduced (compare Figure 4). Nevertheless, the impact of the vertical disc displacement on the braking torque will be experimentally investigated. If necessary, compensation will be carried out.

The oil temperature still impacts the reaction time. There is an additional dependency on the friction coefficient since lower friction coefficients have to be compensated by a higher clamping force. A direct comparison of the reaction time of the conventional and the novel braking system is difficult since the main influences are now different. For a given working point, the reaction time of the closed-loop system is a bit lower.

4 Conclusion and outlook

In this paper, a novel prototype valve for a hydro-mechanical brake torque control was introduced. The behavior of the open-loop and the closed-loop system was analyzed with the help of a parametric study. Based on the results, the introduced control is able to compensate for the influence of a variable friction coefficient on the braking performance.

It was found that the vertical displacement of the brake disc has a major influence on the control loop. This effect was considered already in the design phase by extending the calculation of the braking torque.

Future work will include the manufacturing of the prototype valve. First, the prototype will be examined on a test rig. Then, a reference vehicle will be equipped with the novel brake system to investigate its brake performance under real operating conditions.

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6 References

Load Cycle Investigation of Axial Piston Units Integrated into a Forwarder Application

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Abstract: The consideration of the drivetrains behavior under real operating conditions is becoming increasingly important. While in the past decades it was good enough to have a rough knowledge about the operation conditions in field, a more accurate and detailed picture of the duty cycles is required today. Caused by specific and complex operation conditions the load deviation between different applications is significant. In many cases like e.g. lifetime calculation, a precise knowledge of field conditions leads to the more accurate calculated or simulated results. This paper covers the duty cycles investigation of a forwarder application in field including the measurement concept and implementation. Especially, the analysis of the load conditions, clustering algorithm and lifetime calculation is included. The load spectrum generation is consequently done based on comprehensive load cycle determination. The Aim of this work is to verify the calculation accuracy of the load spectrum by using the clustered and raw data. The outcome of the investigation gives an indicator for the load degree of the drivetrain. Here the developed method creates a new and fast possibility of data analysis and is applicable to most drivetrain applications. Furthermore, the collected data is used as reference for future investigation of the forwarder evolution.

Keywords: axial piston unit, pump, motor, load cycles, clustering algorithm, measurement system, field, drivetrain, load assessment.

1. Introduction

Comparison of the real operating conditions in field to the test lab evaluation of mobile machines leads to a clear recognition of a result deviation. While the test lab evaluation of axial piston units is done under standardized test conditions, the field use is very broad and the same type of units have to withstand many different conditions and load cycles [STO07, LAN02]. The standardized test conditions allow the evaluation of existing development progress of a product according to standards. But, on the other hand the suitability of a product to a specific application is possible as an approximation only. Therefore, different assessment approaches do exist. One approach is the additional cost- and time-consuming test lab evaluation according to field requirements. The other strategy can include simulation tools, but this would only yield theoretical results, whose quality is dependent on the used data and models [HAE05]. Both procedures require a clear picture of the operation conditions in field. Thus, a real load spectrum from the field leads to more accurate estimations of the expected system behavior. Many manufacturers of mobile machines have recognized that the real load spectra are important for product development [MAR03, HOL98]. However, in the past it was not easy to get such data for the component suppliers of the drive system. Nevertheless, an assessment of a drive component based on a specific application requires extensive knowledge of the application. Especially, in the context of a drivetrain simulating of a real operating behavior, the presence of measured load data from a typical application is essential.
To illustrate the approach of the filed data analysis, a forwarder application is analyzed in this paper. The load cycles analysis of a forwarder application is motivated by the needs of stress level investigations and load spectrum determination concerning the lifetime prediction. Thus, a detailed overview and analysis of the load cycles will improve a subsequent phase like product development, drivetrain system sizing or analysis of lifetime prediction. It should be mentioned that this paper is a part of the work around the method improvement of the lifetime calculation for axial piston units. Hence, the goal is to increase the accuracy of calculation methods in comparison to the currently used method according to the field experience.

This work covers the representative load cycle investigation of forwarder applications and the development of a method for just-in-time-analysis. Thus, an automatic tool is able to sort the data immediately according to requirements and to process these directly after the receiving from the field. Therefore, the just-in-time expression is used for this specific procedure in this paper. The use of a measurement equipment in the field is included in this paper as well. Besides the definition of a measurement system and the data transfer via GSM-device (Global System for Mobile Communication), the analysis of relevant load cycle values is taken into account. Thus, an algorithm investigates the incoming data according to the relevant factors and synchronizes the data with a load cycle library. Parameters like pressure, pump displacement and speed are collected here and combined/clustered to a load spectrum form. The load cycles acquisition is consequently followed by a characteristic analysis of the forwarder conditions. The focus of the identification method includes the investigation of significant profile-influencing factors like: maximum drive speed, load and operation phases.

The aim, verification of the calculation accuracy of the load spectrum in comparison to the accuracy by using the raw data, is addressed in this work. The outcome of the identification method gives a first picture of the propel load. Here, the developed just-in-time method creates a new and fast possibility of data analysis and is applicable to most drivetrain applications. The algorithm-performed reduction of raw data is an additional positive outcome of this work, which increases the data quality and leads to improvement of the lifetime prediction as well. Additionally, the offline clustering method is compared to a field connected system. Moreover, the collected and clustered data will be used as reference for future investigation of the drivetrain evolution.

2. Forwarder Application

A forwarder is a forestry vehicle that carries felled trees from the sawed position to a roadside landing area. This application type belongs to the most complex drivetrains in the off-road market, caused by hard transport and operation use in most challenging terrain. Typically, a forwarder picks up the logs from the ground and transports them to the stock. This application does reduce soil impacts like erosion. Forwarders are typically used together with harvesters.

A forwarder is designed for extremely demanding conditions. An illustrative example of a forwarder is shown in Figure 1. The characteristic of a forwarder includes powerful engine, load independent drivetrain and ability to transport heavy loads. Furthermore, the forwarder application has to be robust, terrain- and situation-adoptable. During a typical operation procedure, the vehicle can operate on extremely steep slopes. This often brings the drive system to the limit. Hence, the high system pressure, vibration and temperatures do have an additional impact to the drivetrain. Therefore, the highest safety and technical demands have to be fulfilled. While the drive axles have a very high degree

![Figure 1: Forwarder application.](image)
of freedom to the ground, the drivetrain ability of the load independency provides the needed level of safety standards. In case of high safety standards, special requirements are set on the maintenance management, which is connected to the reliability of the drivetrain components as well [FUS18]. Furthermore, the drivetrain system has to have high off-road grip and at the same time to avoid the destruction of the paved roads. Hence, weather conditions and effects play an important role. Therefore, the operation of the forwarder is typically seasonal in the most global fields of use. The must-have of the seasonal availability brings additional requirements on the calculated lifetime accuracy, like other seasonal application e.g. harvester or combine [SHE15].

3. **Experimental Setup & Data Management**

This chapter covers the generic measurement concept based on the forwarder application. The measurement concept includes the hardware implementation, software of the different transmission levels and data transfer to the database. A forwarder drivetrain includes a simple setup of two main components (pump and motor). Here, a generic hydraulic and sensorics diagram of a forwarder drivetrain is shown in Figure 2.

![Figure 2: Schematic of a generic experimental setup.](image)

The measurement equipment is flexible and adjustable to different vehicles due to fact that this concept is used for different application types. Furthermore, the design of the measurement equipment is built independent from the size and form of the system. The measurement system includes among sensors (pressure, speed, temperature, etc.) a microcontroller unit (signal processing unit) and an integrated wireless device (GSM-device). Different load-affected values are measured by sensors or transferred via CAN-BUS-Message. In this example, the command currents of pump and motor coils are transferred via CAN. The displacement value of the pump is obtained by an angle sensor or can be calculated by coil commands with sufficient accuracy. It is recommended to measure the displacement by an additional sensor, due to possible case of power supply losses, see position 4 in Figure 2. The system delta pressure can be calculated by measured pressure at port A and B. The detection of the shaft speed of both pump and motor is implemented by analog sensors, as well as the integration of two temperature sensors, see sensor positions 8 and 9 Figure 2.

All measured values are device-intern transferred via a second CAN, which is drivetrain-separated. The raw data are temporary stored in a ring buffer. As soon as the GSM-module has the connectivity to the GSM-signal, the data can be transmitted to the database. Additionally, it is possible to exchange the GSM-device by a data logger with a “big” memory. But, the disadvantage is that a data logger requires a manual procedure to restore the data from the measurement system to the database. The data handling is much faster, easier and resource-saving if the GSM transfer is in use.
Today many different data management concepts exist. Here, the distinction between “online” and “offline” methods are in focus. While an online method processes the data onboard of the drivetrain control unit, the offline method transfers the data “into the cloud”. The advantage of the online concept is the “real-time” analysis of the operation conditions. A typical field application for an online system could be a condition monitoring of field data for a real-time maintenance procedure [FUS18]. An online system is however very cost- and resource-intensive [KUN05]. Due to the hardware limitation of memory space, the data management has to reduce the data amount significantly [MAR94]. On the other hand, the disadvantage of the offline system is the high amount of the data transfer versus data resolution. The strength of the offline data handling includes the possibility of secondary data investigation. Thus, among the lifetime calculation e.g. the efficiency or economical aspects can be analyzed at later stage [BUX92, JAR05]. Already in the stage of raw data, the first analysis delivers a good picture of operation conditions. Hence, load spectrum can be visualized to verify the application conditions like pressure limitation. As well as the operating points (profiling investigation) are identified through trigger procedures (threshold values) and thereby consider the status/mode of a mobile machine. However, a deep analysis of the drivetrain behavior at this point requires additional activities. Hence, the data “quality” of continuous record must be improved. In this specific case the data quality was improved by time stamp alignment, threshold filtering of non-operation status, division in modes by specific conditions, etc. Generally speaking, the data have to be structured, filtered or specified by significant factors of interests before analysis [BER04]. Thus, analysis methods like: clustering and lifetime calculation or rain flow analysis are only few examples for subsequent steps [AMZ94].

The procedure of the lifetime calculation requires the analysis of duty cycles. Hence, the raw data are split in predefined sectors (Clustering). A simple clustering includes the “pressure over speed” overview. The third dimension is a count number of the occurrence in this sector. In a typical situation of lifetime calculation, the vehicle manufacturer delivers characteristic duty cycles, which include the application-specific operation behavior. In case of the forwarder application, it may be the worst-case scenarios of a work and/or a transport mode. If the ratio of drive profile scenarios is unknown, some assumptions have to be done. The assumptions are typically based on experiences in this specific field of use. In some cases, the duty cycles are measured as a shortcut of typical operation. Thus, the scenarios ratio is typically equally divided and is recorded at maximum load condition to get all limits of the application behavior. The equal ratio (50/50 > drive profile to work profile) is one factor of the accuracy losses of the currently used calculation method. Thus, the experience shows the deviation between vehicles of the same application and not only between different applications. The duty cycle deviation of same vehicle types is caused by the area of use [KRU92].

4. Forwarder Profiling

A forwarder application does not have standard recurrent cycles due to the arbitrary complex operation conditions of off-road operation. Only operation profiles like drive (transport) or work (creep & pick up) modes can be defined. Thus, the quasi-steady-state conditions are analyzed and lead to the profile separation. For the purpose of profile analysis, the time record of know operation conditions is split and timed. Followed by the analysis of the standard deviation and RPM-thresholds of the axial piston units, the drivetrain profiles are verified, see Figure 3 [DEI09]. The derivative values of the mode ratio were confirmed by the clustering matrix as well, see Figure 4. This paper includes a real data example of a forwarder, used in working mode of 88% and drive mode of 12% of the whole record duration.

Forwarder characteristic analysis shows a fixed operation point of 1400 RPM (pump) during the working procedure. In that case, the driver/operator set an individual fix point of the operation conditions, which is optimal for the working phase like pick up of the tree logs. On the other hand, the drive mode is clearly detectable by RPM rising of pump and motor. Based on the RPM, gear stage and the transmission, the maximum vehicle speed was calculated at 7,5
km/h. Thus, it is obvious that used the test vehicle is operating in the forest only, during the time record phase. The typical maximum vehicle speed for a forwarder can be up to 25 km/h.

5. Profile Ratio vs. Accuracy Loss

In this chapter, the evaluation of lifetime calculation regarding the accuracy loss caused by supposed profile ration is addressed. In this case, the expectation is a significant deviation by using of wrong profile ratio. A real data of a forwarder from the field build the calculation base for the evaluation. The calculation covers bearing example for demonstration purpose of the accuracy loss.

A review of state-of-the-art methods for lifetime calculation may be found in [BAU18]. The calculation example which is mentioned here, was proceeded on Palmgren & Miner method only. Already in 1924, Palmgren presented a simple method for the lifetime calculation of the mechanical components for the example of a bearing [PAL24]. With this first approach a simple calculation possibility was built, which allowed to make conclusions about the load on a bearing over its entire service life. For this purpose, the number of cycles in a certain load class $n_i$ was recorded in field. This value represents the actual load occurring, which is given by measurement. Afterwards, this load is checked in relation to the maximum number of duty cycles for the respective load class $N_i$ before a failure occurs. This value represents the load capacity of the component in relation to the respective load class. The relations are shown in Figure 4 (Calculation subpart). The resulting quotient:

$$D_i = \frac{n_i}{N_i} \tag{5.1}$$

is the partial damage of the rolling bearing for the considered load class. Thus, Palmgren assumes a linear behavior in damage with the number of cycles in the respective load class $N_i$. If the partial damage $D_i$ is summed up for the different load classes of the bearing, the theoretical life of the bearing can be calculated. If the sum exceeds 1, the lifetime of the bearing is exceeded. Different to Palmgren, the Miner’s curve does behave according to the Wöhler line. Thus, the curve runs with a slope of zero, which is equal to fatigue strength. All load classes below the fatigue strength do not have any effect on the lifetime duration, when dealing with iron/steel alloys. Consequently, these stresses do not play a role in the lifetime calculation, according to Miner below the fatigue strength [MIN45].

As mentioned above, it is important to have the field data of a considered application to be able to calculate the lifetime with high accuracy. Normally, it is difficult to get the right duty cycles before a development process due to the pre-study phase of vehicle or subcomponents. Therefore, some estimations regarding the duty cycles have to be done. In that case, it is helpful to have a rough picture of the application and the field of use. A library with load spectra of different applications and their profiles/modes can help to avoid mismatching of the development goals or support the product sizing.
The clustering and calculation tool does include an automatic algorithm. The algorithm downloads the data from the database. Specific load values have to be preselected by the user, based on the decision focus of affected load values. This first tool generation has a matrix resolution of 15x15 cells, see a reduced overview in Figure 4 (Clustering). Thus, a value like pressure or speed can be divided and clustered. The issue during the clustering is the decision of resolution and/or the calculation value of the cells (median, average, etc.). In the next process phase the matrix is reordered due to the calculation equation needs.

\[ L_{10,50/50} = \left( \sum_{i=1}^{n} \frac{c_i}{\Delta p_{ref}} \right)^{\Delta p_i} = 1191 \] \hspace{1cm} (5.2)

where \( L_{10,50/50} \) is the calculated lifetime based on assumed cluster with 50/50 ratio, \( c_i \) is a cluster cell, \( \Delta p_{ref} \) the reference pressure, \( \Delta p_i \) the effected pressure and \( n \) is the sum of the cluster cells. While the real situation is different, as seen in the results of the real ration from the field, see equation (5.3) below. The calculation of the real (clustered) value from the field shows a longer lifetime than assumed before the measurement. Here the lifetime is equal to 2547, which generally strengthens the experience gained from field lifetime in the past.

\[ L_{10,88/12} = \left( \sum_{i=1}^{n} \frac{c_i}{\Delta p_{ref}} \right)^{\Delta p_i} = 2547 \] \hspace{1cm} (5.3)

where \( L_{10,88/12} \) is the calculated lifetime based on cluster with real ratio of 88/12.
6. Clustering versus Raw Data

This chapter covers the verification of the accuracy loss of the lifetime calculation caused by discretization of the load spectrum. The comparison of the clustering versus raw data calculation is done here. The assumption of the accuracy loss is confirmed by using of field generated data of a forwarder. Due to simplification reasons, the calculation was done with one influencing factor (pressure) only.

The acquisition of field data by the measurement concept and the automatic proceeding by the calculation tool allows the simple and fast possibility of lifetime calculation or verification. The direct (field to desk) calculation method of field data (alias: just-in-time-Calculation) has clear benefits in case of condition monitoring. Also, the vehicle specific calculation accuracy increased by using of field data.

In case of calculation algorithm, two different aspects are analyzed. At first the resolution of the clustering matrix is investigated. Thus, the expectation is confirmed that an accuracy loss exists due to the matrix resolution of load clustering, see the encircled area in Figure 5. The cluster steps are the main issue. The cell specific load is inaccurately calculated with assumed value. Here, different approximation methods like load-average or -median can be used. If the load distribution of a cell is unknown only cell average can be used, which can increase the deviation to reality even more. While the raw data considers every single point of the time record, which avoids data loss. This is confirmed by the result of equation (6.2).

The forwarder example with the given data leads to following results: The calculation example is applied to a rolling bearing and pressure only. The results show that a 6.7%-higher lifetime is calculated based on the raw data:

\[
L_{10,\text{Cluster}} = \left( \sum_{i=1}^{n} \frac{c_i}{\sum_{i=1}^{n} c_i} \times \frac{\Delta p_{\text{ref}}}{\Delta p_i} \right)^k = 2547 \quad (6.1)
\]

\[
L_{10,\text{Raw\_Data}} = \left( \sum_{i=1}^{m} t_s \times \frac{\Delta p_{\text{ref}}}{\Delta p_i} \right)^k = 2717 \quad (6.2)
\]

where \(L_{10,\text{Cluster}}\) is the calculated lifetime based on the cluster, \(L_{10,\text{Raw\_Data}}\) is the lifetime based on the raw data, \(c_i\) is a cluster cell, \(\Delta p_{\text{ref}}\) the reference pressure, \(\Delta p_i\) the effected pressure, \(n\) is the sum of the cluster cells, \(m\) is the sampling sum and \(t_s\) is the sampling time of the raw data.

On the other hand, this concept can be used as a verification method of the lifetime exponent \(k\), of the Wöhler curve. This idea requires the long-term use of a measurement system in a big vehicle fleet [HOS14]. The claimed situation can be used in a positive way. Thus, the lifetime exponent can be extrapolated by the feedback from the field. If the load spectrum, the failure
mode and the failed component is known, the load spectrum can also be used for the backward-calculation to get the lifetime exponent:

\[ k = \frac{\ln \left( \frac{C}{P} \right)}{\ln L_D} = \frac{10}{3} \rightarrow \text{for roller bearings} \tag{6.3} \]

where, \( L_D \) is the lifetime until damage, \( C \) is the basic dynamic load rating according load spectrum (time record), \( P \) is the equivalent dynamic load and \( k \) is the specific lifetime exponent of rolling bearing [ISO281].

7. Conclusion & Outlook

The real/raw data from the field delivers a more complete picture of the duty cycles, than the clustered data. Thus, the clustering causes a certain loss of accuracy, which is analyzed and confirmed in this paper. The load spectrum is partly assumed in profile ratio and that leads to an additional calculation failure. Therefore, the lifetime calculation is done more accurate with time record data of a long period in field. Thus, a direct analysis of the raw field data leads to a more accurate result. The outcome of the identification method gives a first picture of the propel load. The algorithm-performed reduction of raw data is an additional positive outcome of this work, which increases the data quality and leads to improvement of the lifetime prediction as well. Moreover, the collected and clustered data will be used as reference for future investigation of a forwarder evolution. Finally, the load spectrum of the real field data and the just-in-time-calculation is the right direction and is one of the first factors to reach the high accuracy method of the lifetime calculation. This is confirmed by the mentioned example of the forwarder based on the comprehensive load determination.

In the next stage, the lifetime calculation approach of the method improvement will cover the separated investigation of subcomponents regarding the lifetime exponent and reliability. Thus, future wok will cover the analysis of the load chain of the axial piston unit as an assembly system. In case of the verification of the improved method a testing concept is required as well. At the later stage a reconstruction of the field duty cycles in the test lab will be under investigation. Thus, the data reconstruction over a period of time, with a large similarity of the original data record and also a nearly identical damage effect will be addressed as well.

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Improvement of Production Quality Related to Leakage at Hydrostatic Control Based on 1D Digital Twin

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Abstract:
One of key component of hydrostatic close circuit units (pump or motor) is hydrostatic control. The control unit influences the performance and behavior of whole hydrostatic unit. That is the reason to pay high attention on the qualitative aspects of whole production process, which includes assembly, machining and tooling impacts. Meeting of requirements in all aspect must be measured and assessed by several qualitative indicators over the processes. A key of them is the First Pass Yield (FPY) indicator, which means how many of the units come out of the production line successfully and lead to shipment divided by number of units going into the process over a specific time.

The aim of the paper is to show simulation deployment into the problem solving of leakage issue. High leakages caused low FPY indicator at production line tests, what was the primary impulse for finding out the root cause in relation to improve quality. Design parameters and their tolerances have direct relation with the issue, but question is which of them are essential with strong impact on the leakage. Fundamental think for agile solution of the problem was to have a digital twin, which was source of useful information about the physical twin. Selection and finding out the most influencing parameters from digital twin were based on sensitivity analysis and correlation with the test data and production processes. Final parameters have been found out from their combinations itself in relation to the machining processes. Outcome of the analysis leads to quality improvement of machining process and the knowledge was also applied to improve production process of similar components. This improvement of production process of hydrostatic controls resulted to increase FPY indicator close to 100% and reduce its variation into almost 0% after the machining improvement.

Keywords: hydrostatic control, sensitivity analysis, simulation, digital twin, leakage

1 Introduction
An effort of simulation engineers is to speed up development, cut costs, and reduce physical testing. On the other hand, simulations can help to reveal causes of the unexpected or unwanted behavior either components or complex systems [1]. Modeling and simulations might be valuable tools for cost reduction by optimization of existing design parameters in relation to production process what is described in this paper.

The more of new approaches are applied and fidelity of the model are achieved the more complex the model becomes or response of the model might be risky. Therefore, the risk should be mitigated by introducing Model-Based Design (MBD) gradually. It means to build model from simple to complex with respect to time and fidelity demand. Practically, this means that
any new design or tools and process changes need to be introduced incrementally [2]. From our point of view, the digital twin means to create 1D parametric model of the product, where parameters come from design. To build the model as digital twin of product is quite time demanding and challenging task with many sub-models, features, physical equations and other details linked and they must be verified. For close study this field see detailed equations which were implemented from Ivantysinova’s and Manring’s books [3], [4] in our case. Typically, once a subsystem of the model has been shown to work well, it might be incorporated within the rest of the model or system as shows [5]. If a simulation model matched all key trends of measured data in test laboratory with acceptable fidelity, then a model is assumed as validated digital twin. Validation process is very complex because there are many inputs which could influence test results and matching with simulation results. Verified models of components could help to solve specific design problems as they occurred in products. This approach was applied in servo controls for hydraulic pumps and motors like it is shown in Figures 1 and 2. If a virtual model of product as product twin is built, targeted benefits could be gain even without making full scale models of the product in real environment. Various extreme operating conditions could be tested and set the limit boundary conditions for reliable working of product during its lifetime. The better understanding how the product is working is fundamental benefit. The immediate advantage of MBD is using of simulations to test and validate designs for prototyping and testing. Later, there can be considered and adopted advanced tools and practices.

Product development and innovation processes are changing. Effort to speed up development, reduce costs, decrease time to market, increase efficiency and improve quality, the boundaries what are possible to change are constantly being pushed. To stay competitive the development process must promote and nurture creativity and innovation.

2 Description of the problem

The parametric model as the digital twin is a basis for agile root cause analysis if some comes from manufacturing. In the Figure 3, you can see workflow diagram of inquiry. The problem statement could arise in anyone stage of analysis process. An issue came from quality check and it initialized further investigation. We measure every servo control at the final check. Therefore, we had available plenty of data sets (correct and faulty units as well), but to simplify big data processing and speed up handling with that, we selected just representative measurements for correct characteristics near to the ideal. All correct servo controls had almost identic characteristics, just faulty products had significantly different characteristics. Low FPY at production test stands was indicator that quality requirements are not satisfied. That was the impulse to start root cause analysis. A reason why the final product did not pass final quality
control was quite high leakages against our internal requirement. Measurements were done after several work cycles to stabilize fluid flow and remove air bubbles from cavities. Leakage means a difference between supply flow and flow to servo. That means internal losing fluid flow to the oil tank. Challenge of the inquiry was how to reduce redundant leakage. There were available 2 ways how to treat high leakage. Design might be changed or tolerances might be constrained strictly, if it is relevant.

For the purpose to reduce redundant leakage, it is most important to find out the best parameters with positive impact on leakage decreasing. Input parameters for an optimization came from production drawings. Several closely connected among each other were merged in other to reduce number of parameters, which were a few tenths. The most studied output characteristics were pressure vs electric current, leakage vs electric current dependence and swash angle vs electric current dependences, but the leakage was the key for the analysis.

### 3 Solution

Input parameters of digital twin were defined based on dimensions, tolerances and other parameters in drawings with regard of function and behavior. Some coupled dimensions were mixed together into the one parameter to decrease number of input parameters. Basis schema how the servo control works is expressed by block diagram in Figure 4 and 1D parametric model was based on that and created in MATLAB/Simulink. The servo control is symmetric with A and B side.
where $I$ – means electric current,
$F$ – means force,
$F_f$ – means flow force,
$p$ – means pressure,
$p_s$ – means servo pressure,
$q$ – means fluid flow,
$q_L$ – means leakage,
$x$ – means stroke,
superscript ‘ – means feedback variable.

Meaning of other variables is evident from the Figure 4. Every block in the block diagram represents one hardware component giving feedback to other parts. Command current in range $0 – 1520$ mA, case pressure 1 bar, charge pressures 25 bar and design parameters were the inputs. Our attention was focused on leakage output from the servo control. The expressions for the evaluated output ($q_L$) would be enormous and there would be little profit in writing them out from object of this paper point of view. Hardware components with design parameters, which are included in main mask, were represented by system of equations deeper in sub-models. The model contains apart of all equations one valuable feature, which is calculation of flow forces at very small openings of pilot edges according to Wu et al. [6]. They used and verified empirical equation for flow area

$$A(x) = \frac{wx}{1 - e^{\frac{x}{d_0}}}$$

where $A(x)$ – means flow area,
$w$ – means rectangular orifice width,
$x$ – means orifice opening,
$d_0$ – means height of square type orifice at the null position.

The height of the orifice, $d_0$, can be expressed as

$$d_0 = \sqrt{(2r + c)^2 + 4r^2 - 2r^2}$$

where $r$ – means radius of corner break (chamfer) on the pilot edges,
$c$ – means clearance between spool and housing.

Sensitivity analysis pointed out the parameters of the model, which have significant impact on behavior of a servo control. Trend lines coming from sensitivity analysis shown how the key parameters shape characteristic lines. Here was applied a reverse procedure as it is usual for sensitivity analysis to match up simulation outcomes with test data. Into the analysis was selected 3 representative measurements from the test data set. Input test data were regularized to create one group of reference characteristic lines. However, we could not have neglected combinations of parameters, because synergy of some parameters significantly changes
characteristics of the servo control. In exploration, we used the reference lines for comparison against simulations in Figure 5. You might assume real edge looks like sharp edge and ideal shape of edge might be like smooth edge as they are sowed in Figure 6.

Simulations and measurement were done for oil Shell Tellus S2 M 46 at charge pressure 25 bar, case pressure 1 bar, temperature 50 °C and both solenoids were energized and deenergized in cycle.

An effort is to match up output characteristics from simulations with test results thanks to various combination of input parameters. An aim was to achieve a combination of key parameters to the best fitting of simulation trends with measurements. Specially, leakage-current curves should fit measurement data. Outcomes from the sensitivity analysis helped us to understand impact of each parameters on the characteristics. Number of incorporated parameters into final simulation case matching test data were eliminated as much as possible, but kept connection between them. Hence, we restricted combination of parameters as few as possible. Finally, here were 2 criteria – sensitivity analysis and correlation with measurements which we considered.

In next step, we explored what manufacturing processes determine values of key impact parameters. Additional challenge was to analyze and evaluate how much machining and tooling influence on the key parameters. Looking on manufacturing processes in detail opened space for improving final values of parameter and their tolerances by change of machining, tooling and other processing sub-step.

4 Results

Result of complex analysis including simulations, measurements and processing technologies were knowledge that key parameter for quality requirements and issue of high leakages was close connected with pilot edges as it shows Figure 7.
These edges are functional in meaning of right working of a servo control. When we selected one parameter, we were possible to focus on this parameter and explore it deeper. In detailed view, we could evaluate overlaps or underlaps, shape of corner breaks, some notches coming from machining as well and others. A design used to assume nice rounded shape of corner breaks on the pilot edges, but microscopic measurement might show different shape of edges. Real shape of edges or corner breaks depends on machining technologies and it differed from ideal, nice rounded edge considerably as the Figure 6 shows. That was place where all observed troubles were arising from.

Here are also several processes as milling, several-step deburring, brushing involved into the result value. We must take care of their exact fulfillment. At the end we could select just one process to raised up quality indicators rapidly. After the verification of the remedy it showed to be only one necessary thing to improve. Impact just one, but crucial remedy was quite huge on final quality increasing.

5 Conclusions

The simulation was based on upgrade of existing model and sub-systems in library. Testing and measurements are realized on existing test stands and hardware. Therefore, these simulation and testing background was not included in estimation of saved effort. The root cause analysis from pre-processing to post-processing was 6 weeks. The root-cause analysis based on simulations included the analysis of 18 parameters (with minimal, nominal and maximal values) and their combinations. Approximately 60 combinations of limit values of several parameters were analyzed and 4 combinations were selected as high influencing leakages at the end. Final solution is deduced from this result. One loop in prototyping and physical testing by trial and error to analyze one design variation might take comparable time as whole root-cause analysis done by simulation. Based on that we might estimate approximately 80 times shorten lead time to deliver the results. Cost saving might be assumed as sum of costs for prototyping and testing on production line and loss due to interrupted manufacturing because of testing on production test stand. Total cost saving by simulations are even higher because costs for manufacturing of hardware is not expressed, but it is substantial costs.

The intention is pointed out the facts that thanks to remedy, scraps were significant reduced and required quality indicator reach to almost at 100% FPY. Production was stable and without high variations of leakages after the remedy. The highest cost savings were in production line thanks to early remedy with huge impact on manufacturing. Analysis pointed out how important is to observe drawing requirements, take care about technologies processes and their relevance for ensuring required and guaranteed quality indicators as well. It all would not be achieved without deploying modeling and simulations in MBD. Afterwards, the work showed how enormous impact just one, but crucial operation can have on final quality.
6 References


Gear Pump with External Gearing for Low Speed with Utilizing the Pressure Field Distribution in the Pump

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Abstract: The purpose of this research is to reduce the minimum speed (revolution) of the gear pumps by 60% up to 200 rev per min. Development was realized under way of NGGP (New Generation of Gear Pumps) approved by the agency of TACR. In the first period, we focused on the determination of pressure field in the teeth area and finding of the optimal hydraulic balance for use in low speed regime. The pressure field was inspected in workspace of gear pump as well as on bush blocks balancing surfaces. In the second period, we focused on the elimination of the so called "dead" areas (where the pressure liquid is going back into the intake) in the pump internal workspace.

Keywords: Gear pump, low speed regime, hydraulic balance

1 Introduction

The gear pump works in fact as the labyrinth seal between the pressure side and the suction side. The flow efficiency therefore highly depends on the pump speed. The higher the speed, the higher the efficiency. The minimum speed of gear pumps ranges about 500 rpm. At high pressures it is still possible to achieve very good volume efficiency. At 250 bar, the volume efficiency of the pump is still around 90%. The loss of flow occurs in the axial direction - contact of gears with the bearing faces and in radial direction - the contact of tooth head with pump body. At the same time, there are losses in the dead space of the pump. These are the gaps that occur when the involute gearing engages.

2 First stage

As mentioned above, in the first stage of development, we focused on axial compensation where the greatest hydraulic losses occur. At low pressures, axial compensation is affected mostly by mechanical thrust, which is due to the pre-stress of shape rubber seal. At high pressures, the axial compensation is affected by so-called hydraulic thrust. The thrust is given by the area between the head circumference (outer diameter of the sleeves) and the Balancing seal. At the same time, this thrust also influences even the back pressure produced in the pump working area. This back pressure we can partly influence by the position and shape of introducing groove. The groove introduce the pressure from the pump discharge to the position we have chosen.

Selected variants that have been tested and on which the characteristics were measured are shown in Table 1 and can be seen in individual figures below.
<table>
<thead>
<tr>
<th>Sample identification</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>E1.1</td>
<td>Standard design of bearing faces for the reference</td>
</tr>
<tr>
<td>E1.2</td>
<td>Standard design with increased mechanical thrust via higher protective plate</td>
</tr>
<tr>
<td>E2</td>
<td>Segment bearing sleeves - lower bearing sleeve + bronze pressure plate</td>
</tr>
<tr>
<td>E7</td>
<td>Bearing sleeves with short introducing groove</td>
</tr>
<tr>
<td>E11</td>
<td>Bearing sleeves without introducing groove</td>
</tr>
<tr>
<td>E12</td>
<td>Bearing sleeves with segment seal dividing the balancing surface into several pressure fields.</td>
</tr>
</tbody>
</table>

Tab. 1 – tested samples

Figure 1 - E1 samples after the tests
Figure 2 - E7 samples after the tests
Figure 3 - E11 samples after the tests
Figure 4 - E12 samples after the tests
Fig. 5 a, b - Samples E2 after the tests; the assembly on the left, view of separate segments on the right - the base is reduced bearing sleeves with closed sealing on the face in contact with the bronze plate.

Characteristics measurements were performed on the pumps with the following configuration and parameters:

- geometric volume $V_g = 16$ ccm, body with threaded ports on inlet $G \frac{3}{4}$ and outlet $G \frac{1}{2}$
- versions of bearing sleeves structure designs see tab. 1
- outlet pressure: $p = 0, 50, 100, 150, 200$ and $250$ bar
- speed: $n = 200, 500$ and $1,000$ rpm
- temperature: $t = 45$ °C

The following diagram compares individual variants in tests at $200$, $500$ and $1,000$ rpm. The results show that, at lower pressures, surprisingly, it was the best standard implementation. The best results are achieved with E11 variants - without insertion groove and E12 with segment seal. Very poor results have been achieved by the variant with increased mechanical thrust.

The volumetric efficiency describes the loss of fluid in workspace at pump.

$$n = \frac{\eta_{\text{act}}}{\eta_{\text{th}}} = \frac{\frac{Q_{\text{act}}}{V_g n}}{\frac{Q_{\text{th}}}{1000}} = \frac{Q_{\text{act}} \cdot 1000}{V_g n}, \quad (1)$$

where $n$ [rpm] – volumetric efficiency,
$Q_{\text{th}}$ [l·min⁻¹] – theoretical flow rate,
$Q_{\text{act}}$ [l·min⁻¹] – actual flow rate,
$V_g$ [cm³] – pump displacement,
n [rpm] – rotation speed.
Figure 6 - Plotting of volumetric efficiency waveforms - 200 rpm

Figure 7 - Plotting of volumetric efficiency waveforms - 500 rpm
3 Measuring the pressure field in the pump

Part of this stage was also measurement of pressure field of gear pumps with the different positioned insertion groove. The theory claims that the pressure in the gear pump increases linearly around the body chamber. This, however, in practice brings a certain limitation following from the pump design itself. First of all, it can be deduced that due to the pump burying at suction side, the biggest pressure increase will be mainly here. For analysis, the sample was designed and manufactured with the possibility of sensing pressures at individual pump locations.

The resulting measurements (pressure sensing was carried out by means of pressure sensors - the actual wiring on the test stand, see Figure 17) was processed graphically - see the following figures (results from measurements at 200 bar pressure provided). The results illustrate the distribution of pressures in inter-tooth areas on perimeter of internal chamber for
individual design variants. From results it can be deduced that the samples show the lowest efficiency at the pressure applied furthest towards the suction area of the pump. From the results, it is also possible to read the refutation of theoretical knowledge which states that the pressure in the pump increases linearly around the gears circumference.

**Figure 10** - Location of measuring points inside the pump in the chamber (on the circumference of addendum circle)

**Figure 11** - Key to the graphical results of pressure field measurement

**Figure 12** - Graphic presentation of E1 sample pressure field at output pressure of 200 bar for 500 and 1,000 rpm

**Figure 13** - Graphic presentation of E2 sample pressure field at output pressure of 200 bar for 500 and 1,000 rpm

**Figure 14** - Graphic presentation of E7 sample pressure field at output pressure of 200 bar for 500 and 1,000 rpm

**Figure 15** - Graphic presentation of E11 sample pressure field at output pressure of 200 bar for 500 and 1,000 rpm
4 Second stage

Highest efficiency achieved at 200 rpm and pressure of 250 bar was only 75 %. Following the results of the first stage, it was decided to make gearing adjustments. It has been combined with the variant of bearing faces with the highest E12 volume efficiency with segment balancing.

When testing samples for low pulsations, it has been observed that the dual flank contact (DFC) gearing has very good efficiency at low speeds. Therefore, for tests the gearing was made not with the DFC, which is very demanding for precision of production and used production technology. The gearing have been calculated with minimal lateral play to avoid rapid wear of involutes on gearing.

The above-mentioned volume characteristics show the considerable increase in efficiency at all pressure levels. Volume efficiency on the new type is not less than 90 %. After checking and inspection of individual pump parts it was decided to pass the durability test (DT) in specification and parameters:

- geometric volume \( V_g = 16 \) ccm
- body with threaded ports on inlet G \( \frac{3}{4} \) and outlet G \( \frac{1}{2} \)
- gearing corrected to minimum lateral play - DFC design variant
- long pressure cycles "on/off" 180/20 sec

Figure 16 - Graphic presentation of E12 sample pressure field at output pressure of 200 bar for 500 and 1,000 rpm

Figure 17 - Connection of test sample on the test stand; pressure sensors connected to measuring points 1, 2 and 3

Figure 18 - Flow characteristics of test sample before DLZ starting
- output pressure $p_2 = 260$ bar
- speed 200 rpm
- oil temperature $40 \div 50$ °C

The pump has worked 10,260 long cycles of 180 s / 20 s at speed 200 rpm on the pressure of 260 bar at temperature about 45 °C with VG46 oil. During the first half of the test, the flow decreased, and it was further decreasing to the final decrease, in total - 10.43 %. Due to heavy duty operation at very low speeds, the decrease is acceptable and within the tolerance after the DT (max. up to 20 %). Overall, the pump wear responds to its intensive load and the pump is capable of further operation.

![Figure 19 - Change of flow rate during DLZ (measured before, in ¼, ½, ¾ and at DLZ end)](image)

5 Conclusion

During development, we had achieved very high efficiency at decreasing the minimum speed to 200 rpm. However, due to the specific construction and the given target, the pump was not tested at more than 1,000 rpm.

In course of development we have found two fundamental principles that can be implemented in subsequent development and application of some individual parts to actual pumps operation:

- DFC principle (double flank contact)
  - The double flank contact principle can be used not only in applications to reduce pump pulsations and noise, but also in applications requiring low operating speeds
- Segment balancing sealing
  - The segment seal principle shows the best axial thrust and, at low speed, it positively influences the flow efficiency

6 References

Optimization Method for Designing of Hydraulic Elements

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Abstract: Two methods are generally possible to design and optimize hydraulic components and devices. The classic method is the experimental method. In the hydraulic laboratories, various models of components and devices are examined to understand their basic properties, to verify proposed assumptions, or to alter derived theoretical equations to equations that approximate reality, etc. In some cases, which are very difficult to solve theoretically, or even yet unsolvable, you can only get the values you need using an experiment. However, not all phenomena can be described through models. Mathematical-Physical modeling is a method by which Mathematical models based on the application of physical laws and phenomena can achieve the necessary results. These mathematical models consist of the definition of equations describing the given processes, which must be solved by means of numerical methods. Fluent, CFX Computerized software is used to solve the problem. Simulation can be performed within these softwares, which allows to evaluate different variables in a short period of time, to change the design of the element to suit the application, etc. However, it is a prerequisite to check the retained results by the experimental method. The method of optimizing the parameters and shapes of products and equipment is already an integral part of the design process. This achieve product shape improvement without having to produce A number of Prototypes, you can create a variety of variants and perform simulations for different conditions. At present, the mathematical optimization method is based on the principle of adjunction, which is part of the ANSYS Fluent solution, which means saving time and finance while achieving qualitative improvement. The article focuses on the theoretical and practical possibilities of using this method in the field of hydraulic elements.

Keywords Air and vapor cavitation, experimental measurement, CFD,

1 Introduction

The general fluid mechanics were historically divided into two areas, hydrostatics and hydrodynamics. Practical applications often involve both areas at the same time, [3]. The solution of flow problems can, in principle, be realized by mathematical and physical approach. But, both mathematically and physically, this is a complex problem. The solution of individual tasks is mostly separated, i.e. either by mathematical or physical approach. However, it seems very advantageous to deal with both approaches simultaneously and to combine the
strong aspects of these approaches. The physical experiment provides basic information on the
flowing fluid, geometry of the area, definition of the boundary conditions and the verification
data. Verification data are possible at selected points in the area. The mathematical approach is
extremely advantageous especially in terms of visualization of the area with the flowing fluid
[4]. It allows inspection into the fluid together with the calculation of all important physical
quantities in the whole area of the flowing field. This approach allows you to use simpler
experimental measuring equipment even for complex tasks.

The mathematical model consists of the definition of transfer phenomena by means of the
conservation laws of mass, momentum or other quantities. Because flow is a one-dimensional,
planar, two-dimensional, axially symmetrical or generally three-dimensional and time-
dependent, it is described by a system of integral or differential equations defined in the space,
to be solved by numerical methods. Their use is conditioned by the need of extended knowledge
of the field of flow, turbulence, numerical methods, computer technology, etc.

Modern engineering science can create and solve mathematical models of physical systems.
Students often think skeptical about fluid mechanics, as the basic mathematical models seem
very complicated and difficult for applications. But for example, a simplified hydrostatic theory
can be used to determine the distribution of pressure in the area. It is true that the fluid dynamics
described by Navier - Stokes equations are complex and simple solution, except for minor
exercises, doesn’t exist.

The last period is characterized by the rapid development of numerical methods for flow
modelling, both laminar and turbulent. In this context, several quality software products are
created that can be used. Part of them is dedicated to the one-dimensional flow of liquids and
is mainly usable in the solutions of fluid systems, which consist of fluid elements and piping
system [4]. The mathematical model is based on the electro-hydraulic analogy (Flowmaster the
extension of Matlab simulink, ie. SimHydraulics, AMESIM).

Other software specialize in spatial flow and their use primarily falls into the area of
construction and currently, even in mathematical optimization of fluid elements (Ansys Fluent,
CFX, STAR CC +). The result is the distribution of pressure and the velocity or flow of the
entire solved area [5], [6], [7]. The boundary conditions that significantly affect the solution
must be considered.

Theory of optimization using gradient method

The idea of Adjoint method is appearing everywhere in modern and classical mathematics.
It dates back to the 18th century. It has only recently proved to be a strong means to extend
engineering analysis using CFD methods. Adjoint Solver provides specific information about
the fluid system, which is very difficult to obtain in another way, [1], [2]. It can be used to
calculate derivations of the engineering quantities with respect to all inputs into the system. An
example is

- Derivation of resistance with respect to the shape of the vehicle.
- Derivation of the total pressure gradient with respect to the shape of the flow path.

Adjoint Solver is a specialized tool that extends the analysis built into the standard
(conventional) solver and provides detailed information on the sensitivity of the fluid system.
In order to perform the simulation using the standard flow solvers of ANSYS, the user creates
the geometry with the computational mesh, determines the properties of materials and physical
models, configures the boundary conditions of different types. As soon as the standard solver
converges, it provides a detailed set of data describing the state of flow. If a change is made to
any of the data that define the problem, then the results of the calculation may change. The
changes depend on how sensitive the flow is to the edited parameter. It is a specialized tool that
extends the standard analysis, capable of providing a record of sensitivity of the system, that
can be used to optimize the design of a given element. In fact, the derivation of the solution
result according to this parameter is quantified by the sensitivity of the first order. Determination of derivatives is the essence of sensitivity analysis.

There are many optimization methods which suitability for each case is defined by the time demands of the calculation and the efficiency of handling many constructional variables. The gradient method is the best-known method that is capable of working with a large number of construction variables.

2.1 Methodology of solving Adjoint solver
As mentioned above, Adjoint Solver can be used only when a computational mesh is created, physical models, boundary conditions, etc. are set and data set, generated by a standard flow calculation, is available. The following resolution procedure is divided into several phases.

- Adjoint Solver Settings – it concerns with defining monitored variables, solution controls, and so on.
- Calculation of the sensitivity of the system against the specified variables - after completion of the calculations (if convergence occurs) there is a sensitivity data set which can be used to define the design change of the system.
- Morphing-Adjoint Solver - it allows you to precisely and easily determine which part of the geometry is to be adjusted. After you modify a shape, you do not have to create a new mesh, because the computational mesh automatically reshapes when geometry changes.
- Standard Calculation – analysis of the flow in the new geometry.
- Calculation Repetition - if the calculation converges, it is possible to repeat the whole procedure as long as the residuals of both calculation of the sensitivity of the system and the standard calculation converges or until the change in design and the value of the monitored variable is found sufficient.

On Fig. 1 the solution methodology of the Adjoint Solver module is displayed [8]

2.2 Restrictions on the use of adjoint solver
Adjoint Solver is a method that has a specific limitation and is based upon the following basis [1]:
- The flow state is defined as a permanently incompressible single-phase flow that is either laminar or turbulent and lies in an inertial reference system.
• The basic flow must be solved for such boundary conditions, so that the task converges well and quickly (i.e. not too severely turbulent, so that vortex path doesn’t appear due to wrapping of obstacles to ensure sufficient pressure in the area of interest, because then adjoint solver will converge well).
• For turbulent flow the assumption of frozen turbulence is used, in which the effect of changes in the state of turbulence is not considered in the calculation of sensitivity.
• In turbulent flow, standard wall functions are used on all walls.
• Adjoint Solver uses methods that are first order of precision in space by default. Second-order precision methods can be selected.
• Boundary conditions are only the following type: wall, input velocity, output pressure, symmetry, rotational and translational periodic conditions

It is important to note that these requirements are not a restriction for basic flow solver, but they are a limitation for Adjoint Solver. For hydraulic and pneumatic tasks (i.e., the flow in closed areas) is advisable to use parts of solver relating to optimization of pressure gradient, while resistance and pressure forces are evaluated for wrapping tasks. Also, the combination of monitored parameters is very illustrative.

Stability problems can occur when applying an adjoint solution to tasks with a very fine computational mesh, complex geometry, or possibly on tasks with a high Reynolds number. These instabilities may be based on the irregularity of small dimensions in the fluid field or, possibly because of severe shear stress and tend to be limited to small and isolated parts of the flow area. If they are neglected, their presence may disrupt the entire Adjoint calculation despite the fact that the occurrence can occur only in several cells of the computational mesh. To obtain a adjoint solution in such cases, it is necessary to apply a stabilization scheme, [6], [7].

3 Application of Adjoint Solver when optimizing geometry of an elbow

The task is to use the Adjoint Solver tool, which is a part of the ANSYS Fluent 18.2, to reduce the pressure loss in the elbow and to smooth the velocity profile, ie to minimize the mean variance of the total pressure. The flow is assumed to be stationary, isothermal, turbulent (k-ε standard model, standard wall function). The flowing medium is considered air with physical properties and boundary conditions defined as follows.

![Fig. 2 area geometry and calculation mesh](image)

<table>
<thead>
<tr>
<th>Area Dimensions:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input diameter $d_1$</td>
</tr>
<tr>
<td>Output diameter $d_2$</td>
</tr>
<tr>
<td>Input length $l_1$</td>
</tr>
<tr>
<td>Output length $l_2$</td>
</tr>
<tr>
<td>Elbow radius $R$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Air physical properties:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density $\rho$</td>
</tr>
<tr>
<td>Dynamic viscosity $\mu$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Boundary conditions:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input (Velocity inlet):</td>
</tr>
<tr>
<td>Input velocity $v$</td>
</tr>
<tr>
<td>intensity of turbulence $I$</td>
</tr>
<tr>
<td>hydraulic diameter $d_h$</td>
</tr>
<tr>
<td>Output (Pressure outlet):</td>
</tr>
<tr>
<td>static pressure $p$</td>
</tr>
<tr>
<td>intensity of turbulence $I$</td>
</tr>
<tr>
<td>hydraulic diameter $d_h$</td>
</tr>
</tbody>
</table>
The first step is to calculate the fluid field, while the task must converge as stationary, see fig. 3.

The pressure loss at steady air flow was evaluated as a difference of the mean value of the static pressure on the inlet and outlet $\Delta p = 32.5 \text{ Pa}$. With regard to the minimizing this pressure loss, optimization will take place.

By setting the Adjoint Solver parameters, appropriate methods of solution and, where appropriate, stabilization methods shall be obtained in a convergent solution. The sensitivity map is the prediction of geometry optimization and it shows appropriate places where it has sense to make changes to the geometry due to the minimization of goal function, which is defined as:

$$f = |p_{\text{inlet}} - p_{\text{outlet}}| + \text{var} \sqrt{p_{\text{total}}}$$

see Fig. 4. From this follows the definition of the area of deformation, see fig. 4.

From this follows the definition of the area of deformation, see Fig. 5.
Already at this stage you can determine the expected change of goal function. This value is estimated and only refined after the optimization by calculating the full fluid field. If this value is sufficient for the user, the geometry is changed, and a preview of the modified geometry is displayed. Deformation can be repeated, while monitoring the convergence of Adjoint Solver, the value of the expected pressure gradient and the reality of the new geometry, see Fig. 6.

The effect of the shape change on the pressure gradient and velocity distribution is checked by calculation of the basic fluid field using the original Navier-Stokes equations and the results of pressure are displayed on Fig. 7.

In the table 1 there is the evaluation of changes of monitored pressure in relation to the default state. The results after the two iteration loops are listed in the table.

1. Table 1 -Overview of the changes in pressure gradient in table with percentages

<table>
<thead>
<tr>
<th>Modification</th>
<th>0</th>
<th>1</th>
<th>2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Δp [Pa]</td>
<td>28,3</td>
<td>24,5</td>
<td>21,7</td>
</tr>
<tr>
<td>Variance [%]</td>
<td>0</td>
<td>13,43</td>
<td>23,32</td>
</tr>
</tbody>
</table>

In the last Fig. 8 the contours of velocity magnitude for three modifications are shown.
Fig. 7 Distribution of pressure in axial cross-section [Pa]

- Modification 0
  - original geometry with high negative pressure

- Modification 1

- Modification 2

Fig. 8 Distribution of velocity magnitude in axial cross-section [m/s]

- Modification 0
  - original geometry

- Modification 1

- Modification 2
6 Conclusions

Optimization of the geometry during fluid flow was so far solved by the method of trial and error, when, based on experience, a number of variants of the given geometry were created and subsequently tested. Optimization is currently a new trend in design of hydraulic elements. It uses numerical methods for spatial modelling of flow and numerical optimization methods. The paper focuses on a brief explanation of the principle and procedure of using the optimization method. It should be noted that the optimization should be carried out carefully so that the optimized element should have "reasonable shape", required properties and should be manufacturable. The consequent problem is to export of optimized geometry to the common CAD format.

The condition of a stationary flow in a given geometry may appear to be a significant limitation of usability. It is then on the user's experience if he can solve the optimization only on selected parts of the element where this condition can be accepted, and after optimizing them join these newly modified parts together. Often a stationary solution predicts cavitation on the walls of the hydraulic elements. Even here, optimization can be an effective means.

The most common use of optimization is in the automotive industry in the applications of the flowing through various piping systems.

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7 References

Intelligent Parameter Identification of Hydraulic Systems Based on Salp Swarm Approach

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Abstract: In this research, a new intelligent method is developed for identification of hydraulic systems. An evolutionary algorithm is developed to identify the system. New Salp Swarm Algorithm is chosen for this purpose. The new developed algorithm is an adaptive version of the basic one. It is developed in a way that can update its parameters using previous steps of the optimization. To show the proficiency of the developed method, it is employed to identify the hydraulic system in two different conditions, in presence of the noise and without noise. The results of the simulation are presented.

Keywords: system identification, hydraulic systems, evolutionary algorithm, salp swarm algorithm, online identification

1 Introduction

To control practical systems, it is needed to model the system properly. To have more accurate results, it is necessary to collect the proper parameters of the system. Despite the model needs these parameters, some of them are not available in many practical systems. Different methods has been introduced to predict these parameters. System identification methods is one of the interesting topics for the researchers in recent years. Different methods have been developed for system estimation and identification [1], [2]. For instance, an iterative algorithm has been developed to identify the nonlinear systems [3]. This method is a orthogonal least squares forward regression based approach. In another research, two well-known methods, least mean pth power (LMP) and normalized LMP, has been employed to design new proper approach for system identification [4]. In the paper, the proposed method has been designed in an environment with alpha α noise. In [5], a dual Kalman filter has been used to estimate the parameters of the system. The approach has combined the Kalman filter with a probability density function truncation method. Hydraulic system is one of important practical systems, which need to model and identify properly. Many researches have been focused on these systems [6].

On the other hand, the different approaches of the artificial intelligence, such as fuzzy logic [7], neural network [8], and evolutionary algorithms [9], [10], have a wide usage in different engineering subjects. For instance, the combination of the type-2 neuro-fuzzy structure and sliding mode controller has resulted to introduce a new self learning disturbance observer [11]. The proposed approach has been designed in a way that can learn the uncertainties, which can help it to estimate more accurately. It has been used to estimate time varying disturbances. In [12], dynamically driven recurrent networks have been employed to create a new observer for nonlinear systems. The proposed method has been used to estimate the states of two different batteries. Metaheuristic algorithms are one of the artificial intelligence methods, which have a good performance on solving engineering problems. For instance, Particle Swarm Optimization has been combined with H1 loop shaping structured controller to control a robotic arm. The controller has been designed in a cost-effective manner. The evolutionary algorithm has been
used to develop new methods for estimating and parameter identification of different nonlinear systems. For example, an improved gravitational search algorithm has been employed for parameter identification of nonlinear system [13]. These intelligent methods have different applications on hydraulic systems. For example, a multi-objective genetic algorithm has been employed to design new approach for parameter estimation [14]. The proposed method has been used for modelling a seven degree of freedom hydraulic manipulator. Salp swarm algorithm is known as a strong optimization algorithm [15]. Despite the algorithm has acceptable performance on engineering problems [16], it has an important disadvantage. This disadvantage is that the algorithm does not consider the fitness value of the candidate solution to update it. It can be considered as a research subject to develop an algorithm with better performance.

In this paper, a novel intelligent method is presented to identify the parameters of nonlinear systems. The proposed method is based on an evolutionary computation. The salp swarm algorithm is chosen for the metaheuristic computation. The algorithm is developed in adaptive manner, in a way that can adapt itself during the optimization procedure. The adaptive version is updated itself regarding the fitness function of the candidate solutions. The proposed method is applied on a hydraulic position servo system. Different scenarios are defined to test the presented method. The results of the simulation are presented.

The rest of this paper is arranged in three main sections. Section II describes the principles of the new presented method. The description of the hydraulic position servo system and the simulation results are given in section III. Finally, the conclusion of the paper is presented in section IV.

2 Methods and materials

This section describes methods that will be used to design the optimal parameter identification. First, the proposed identification scheme will be expressed and then, the new developed evolutionary algorithm will be presented.

2.1 Parameter identification

The parameter identification methods are applied to estimate the known parameters of different practical systems. The general model of the nonlinear system can be supposed as following:

$$\dot{x} = f(x, u, \theta)$$  \hspace{1cm} (1)

where the states of the system, its input and the unknown parameters are given by $x$, $u$, and $\theta$, respectively. The procedure of the parameter identification is designed in a way that can estimate and identify the unknown parameters of the system. It is tried that the result of the identified system is similar to the real system. The identified system can be modelled as following:

$$\dot{\hat{x}} = f(\hat{x}, u, \hat{\theta})$$  \hspace{1cm} (2)

Different criteria has been used to evaluate the performance of the identified system. One of the most important and well known criteria is Mean Square Error (MSE), which is used in this paper. The criteria can be formulated as following:

$$MSE = \frac{1}{m} \sum_{l=1}^{m} (x(l) - \hat{x}(l))^2$$  \hspace{1cm} (3)

where $m$ is the number of the samples. Regarding to abovementioned definitions, the overall procedure of the identifying of the system can be summarized as following steps:

- Running real system
- Collecting data with measurement noise
- Producing the estimated output using the model of the system
- Evaluating the performance of the estimated system
• Optimizing the parameters of the model during optimization procedure regarding evaluation criteria

The schematic of the proposed procedure is given in Figure 1.

![Figure 1 – The schematic of the intelligent parameter identification](image)

### 2.2 Salp Swarm Algorithm (SSA)

In this section, a new adaptive version of the salp swarm algorithm is presented. The basic algorithm has been introduced recently [17]. It is based on the behaviour of a swarm. The algorithm has been developed regarding the behaviour of salp swarm. The procedure of the basic algorithm can be summarized as following steps:

**Step 1: Initialization:** Initialize the position of the candidate salps using following equation:

\[
X_i^0 = B_u + \text{rand} \times (B_u - B_l); \quad i = 1; \ldots; N
\]

where \(X_i^0\) shows initial value of the \(i^{th}\) candidate solution, which is defined as salp in this algorithm. The parameters \(B_u\) and \(B_l\) present the upper and lower bound of the search space. The rand is a random number in range (0, 1).

**Step 2: Updating of leaders:** The position of the salp with best fitness value is updating as follows:

\[
\begin{cases}
    X_1 = \left\{ \begin{array}{ll}
    \text{Food} + \alpha_1 ((B_u - B_l) \times \alpha_2 + B_l) & \alpha_3 \geq 0 \\
    \text{Food} - \alpha_1 ((B_u - B_l) \times \alpha_2 + B_l) & \alpha_3 < 0
    \end{array} \right. \\
    \alpha_1 = 2 \times \exp \left( -\left(\frac{4+k}{\text{Max Iteration}}\right)^2 \right)
\end{cases}
\]

where the position of the first salp is shown by \(X_1\). The parameters \(\alpha_1\), and \(\alpha_2\) are random numbers in the interval of [0; 1]: The current iteration and the maximum value of iterations are given by \(k\) and \(\text{Max Iteration}\), respectively.

**Step 3: Updating of follower salps:** The position of these followers are updated regarding the Newton’s law of motion. This law can be shown as following:

\[
X_i = \frac{1}{2}at^2 + v_0 t
\]

where \(t\) and \(v_0\) presents time and initial speed, respectively. Regarding some simplifications, which are given in [17], the updating equation of the follower salps can be shown as

\[
X_i = \frac{1}{2} (X_i + X_{i-1}), \quad i \geq 2
\]

where \(X_i\) is the position of the \(i^{th}\) salp.

**Step 4: Choosing the best solution:** The salps are sorted regarding their performance, and best one is chosen.

**Step 5: Stop criteria:** If the stop criteria is satisfied, stop the algorithm and return the best solution, if it is not, go to step 2.

#### 2.2.1 Adaptive Salp Swarm Algorithm (ASSA):

Despite the acceptable results of the basic algorithm, it has a main disadvantage. It can be expressed that to update the salps, the algorithm does not consider their fitness value. By considering this fact, the algorithm can be changed in a way that has a better performance. One of the improvements that can be considered, is
introduced in this paper. In order to update the position of the salps with considering their previous fitness, the following update equation are suggested to replace in the basic algorithm:

\[
X_1 = \begin{cases} 
Food + \alpha_1 \left( (B_u - B_l) \alpha_{\text{adaptive}} + B_l \right) & \alpha_3 \geq 0 \\
Food - \alpha_1 \left( (B_u - B_l) \alpha_{\text{adaptive}} + B_l \right) & \alpha_3 < 0 
\end{cases}
\]  

(8)

where \( \alpha_{\text{adaptive}} \) is the new updating factor, which can be expressed as following:

\[
\alpha_{\text{adaptive}} = 1 - \exp \left( -\frac{\text{fit}_{\text{best}}}{|\text{fit}_{\text{best}} - \text{fit}_{\text{worst}}|} \right)
\]  

(9)

where \( \text{fit}_{\text{best}} \), and \( \text{fit}_{\text{worst}} \) present the best and worst value of salps’ fitness. In order to considering the value of the fitness of the salps, the updating equation of the follower salps are changed as following:

\[
X_i = \frac{1}{2} \left( \frac{\text{fit}_i}{\text{fit}_{i-1}} X_i + \frac{\text{fit}_{i-1}}{\text{fit}_i} X_{i-1} \right), \quad i \geq 2
\]  

(10)

An other innovation in the new algorithm can be mentioned as using the mutation function to avoid stacking in sub optimal solutions. The flowchart of the procedure of the adaptive version of the algorithm is given in Figure 2.

Figure 2 – The flowchart of the adaptive salp swarm algorithm

3 Simulation and discussion

In order to show the proficiency of the proposed intelligent method, it is applied to parameter identification of a hydraulic system. Different scenarios are defined two test the approach. First scenario is based on the parameter identification with measurement noise and the other one is defined in presence of measurement noise. In this section, first, the hydraulic system is presented and then the results of the simulation will be given.

3.1 Hydraulic position servo system (HPS)

Hydraulic position servo is a well-known hydraulic system which has been used as application for different approaches [17], [18]. In this study, it is tried to identify the parameters of the system. The schematic of a HPS system is presented in Figure 3. The state space of this system can be expressed as following [19]:

\[
\begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \frac{1}{m_1} (b_1 x_1 - b_1 (x_2 - x_4) - k(x_1 - x_3) + A_1 x_3 - A_2 x_6) \\
\dot{x}_3 &= x_4 \\
\dot{x}_4 &= \frac{1}{m_2} (-b_2 (x_4 - x_2) - k(x_3 - x_1)) \\
\dot{x}_5 &= \frac{\beta_e}{V_1} (Q_1 - A_1 x_2) \\
\dot{x}_6 &= \frac{\beta_e}{V_2} (Q_2 - A_2 x_2)
\end{align*}
\]  

(11)
where
\[
Q_1 = \begin{cases} 
    c_v x_v \sqrt{P_s - x_5}, & x_v \geq 0 \\
    c_v x_v x_5, & x_v < 0 
\end{cases}
\]
\[
Q_2 = \begin{cases} 
    c_v x_v \sqrt{P_s - x_6}, & x_v < 0 \\
    c_v x_v x_6, & x_v \geq 0 
\end{cases}
\]
\[(12)\]

In equations (11) and (12), the state variables of the HPS system are considered as \( X = [x_1, x_2, x_3, x_4, x_5, x_6]^T = [x_v, \dot{x}_v, x_1, \dot{x}_1, p_1, p_2]^T \). The definition of the parameters and their values, which are used in above-mentioned model, are presented in Table 1.

![Figure 3 – The schematic of a HPS system [19]](image)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Definition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A_1 )</td>
<td>The area of the piston in chamber one</td>
<td>( 5024 \times 10^{-6} )</td>
</tr>
<tr>
<td>( A_2 )</td>
<td>The area of the piston in chamber two</td>
<td>( 1908 \times 10^{-6} )</td>
</tr>
<tr>
<td>( b_1, b_2 )</td>
<td>The coefficient of friction</td>
<td>3000</td>
</tr>
<tr>
<td>( \beta_e )</td>
<td>Effective bulk modulus</td>
<td>( 14 \times 10^9 )</td>
</tr>
<tr>
<td>( v_0 )</td>
<td>Dead volume</td>
<td>( 5 \times 10^{-4} )</td>
</tr>
</tbody>
</table>

### 3.2 Simulation results & discussion

In this section the results of numerical simulations are presented. To show the proficiency of the presented method, different scenarios are defined. These can be separated into three configurations: parameter identification without noise, parameter identification in presence of noise, and online identification. As a matter of fact the simulations need a fitness function, which should be optimized by determining the parameters. For this research, the continuous version of MSE criteria has been chosen and it will be tried to find the parameters in a way that minimize the function. In all of the simulations the stop criteria is chosen as reaching maximum iteration. The simulation results of different scenarios will be given in the following.

#### 3.2.1 Parameter identification without noise:

In this simulation, it is supposed that the system is working normally. Any noise will not influence the system, and its parameters, which should be identified, are constant. Regarding these assumptions, the system formulation can be considered equation (1). By applying the proposed method to identify the parameters of the hydraulic system, results can be shown as Figure 4. As it can be seen in the figure, the developed algorithm identified the parameters of the system properly. Regarding to the results, it is obvious that the algorithm can be applied on a practical system for identification of the some of the parameters of the system. The designed method shows its power to detect and identify different parameters simultaneously.
3.2.2 Parameter identification in presence of noise: This part simulate the system in presence of the noise. Regarding this fact, the system model which has been shown in equation (1), can be rewritten as

\[ \dot{x} = f(x, u, \theta) + \omega \]  

where \( \omega \) is a white noise. For the simulation, the covariance of the white noise is set as \( 10^{-5} \). By applying this assumptions on the system, the result of the identification procedure is shown in Figure 5. The figure shows that the presented algorithm has found the parameters with a minimum difference with the real values. The figures which show error of the identification of each parameter, the error is acceptable and it is good for an optimization problem. This simulation is done in presence of the noise which mimics the real world condition. By considering this facts, it can be drown that the algorithm can be a good choice for the practical system. In our other simulations, more powerful noises affected on the system and the proposed method finds the parameters properly.
4 Conclusion
This paper has presented a new intelligent identification method for nonlinear systems. The proposed method has used a new developed evolutionary algorithm. The salp swarm algorithm has been chosen for this purpose. The algorithm has been changed into an adaptive version. This version has updated the feasible candidates regarding their fitness function. The proposed method has been applied on a hydraulic position servo system. Different scenarios have been defined for the identification procedure. The results of the simulation have been presented. For the future work, it has been planned that the proposed method applied on the practical systems to identify their unknown parameters.

5 References


Development of Excavator HILS Simulator for Evaluating Leveling Motion

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Abstract: This paper introduces excavator HILS simulator for evaluating controllability of leveling motion. A real-time simulation model of the excavator is developed for middle class open-center system to simulate not only single motion but also leveling motion. The hydraulic and dynamic parts are modeled using AMESim. And, they are reduced to the extent that it satisfies real-time capability and simulation accuracy. Then, the model is verified with measured data of real machine. To build an operator environment, Remote Control Unit is used. In addition, Unity based excavator virtual reality is applied to enhance sense of reality. Finally, Human-In the-Loop-Simulation is established interfacing real-time simulation model and operator environment through VeriStand and PXI controller. The developed excavator HILS simulator is validated by operator’s intuitive tests.

Keywords: Excavator, HILS (Human-In the-Loop-Simulation), Hydraulic System

1 Introduction

The controllability of the hydraulic excavator is an important performance index that affects productivity as well as fuel consumption. And that is affected by the dynamic characteristics of the hydraulic system. An open center system, such as a negative control system, is one of the common ways of constructing an excavator hydraulic system. The MCV (Main Control Valve) has a number of spool valves as shown Figure 1 to control each hydraulic actuator. The controllability is affected by design parameters such as spool notch, spool open area. So it takes a lot of effort to optimize the performance with tuning activities. Recently, various researches using a commercial software have been conducted to predict physical behavior of excavator hydraulic system from the viewpoint of design or control [1][2]. However, model accuracy is important considering the controllability in the simulation. Meanwhile, operator models are typically generated by tuning control models to imitate specific trajectories. If the system parameters are changed, or the operating conditions are varied, the model often has to be retuned to match the new operating profile [3]. On the contrary, human operator can adapt to changes in the machine. In these reason, Excavator HILS is developed to evaluate leveling motion. The HILS consists of a real-time excavator model, an operator environment featuring Virtual Reality and an interface environment [4][5].

2 Real-Time Simulation for Excavator

The HIL method requires real-time simulation model for interfacing operator environment. The model can be used to predict the physical behavior of the real machine. In this study, real-time model of the excavator is developed for middle class open-center system to simulate not only single motion but also leveling motion. The excavator model is mainly composed
with hydraulics and dynamics. The dynamics of the attachment is modeled using 2D Planar Mechanical Library in the AMESim. Then, the parameter definition is set with drawing. The hydraulic components such as pump, MCV, valve and actuator are simply and functionally modeled by means of Hydraulic library in the AMESim. And design parameters are entered in table or function form based on experimental data and drawing. An orifice and chamber are singly or equivalently used considering the pressure loss. To calculate the flow sharing due to internal confluence or regeneration, the maximum flow coefficients are determined through experimental data. This study, a predefined time step with respect to working cycle is 0.1ms. The numerical instability of the excavator model is reduced by experience-based tuning activities using the linear analysis. But the model reduction is minimized to ensure model accuracy. Also, model decomposition is performed base on model’s frequency to disperse the computing load. The completed model for fixed step integrator is verified with measured data during leveling motion as shown in the Figure 2. Figure 3 shows that overall simulation behavior match well with testing data.

Figure 1 MCV and Spool valve

Figure 2 Schematic of Leveling Operation

(A) Hydraulic Pump Pressure

(B) Hydraulic Cylinder Stroke

Figure 3 Model Verification of Leveling Motion
3 Human-In the-Loop-Simulation

Figure 4 show the system configuration for HILS simulator of Excavator. The system consists real-time simulation and operator environment. In the former, real-time excavator model is embedded to the Target PC through NI (National Instruments) VeriStand. The Target PC has Phar-Lap as real-time OS and is a quad-core PXI computer that handles the real-time model computing and signal conditioning. Also, since this system supports multi-core computation, the computing load can be reduced by using the decomposed models of excavator. In the operator environment, the RCU (Remote Control Units) are constructed to manipulate hydraulic actuators in the virtual excavator. In addition, the excavator VR (Virtual Reality) is created using Unity engine and is applied to enhance the sense of reality when a human operator has intuitive manipulation in the virtual excavator.

![Figure 4 Schematic diagram of Excavator HILS Simulator](image)

4 HIL Evaluation

To validate effectiveness of the developed excavator HILS simulator, HIL test as shown in Fig 6 is implemented according to changing design parameter. In generally, leveling operation of excavator is divided into “Boom-Up & Arm-In” and “Boom-Down & Arm-Out”. The test scenario is as follow. The human operator performs the leveling operation by manipulating RCU in virtual excavator. At this time, the bucket trajectory is manipulated so as to have
possible consistency. When leveling motion is completed a certain number of times, the leveling operation is performed by changing design parameter in real-time. The test results are shown after Figure 7. The design parameter of spool is changed from "Origin" to "COA", it means that the flow rate from the pump to the arm cylinder is reduced during “Arm-In” operation. Figure 8 shows that leveling trajectory was similarly operated for each case. In case of "COA" in Figure 9, when the leveling operation (Boom-Up & Arm-In) starts from about 35 seconds, pump flow is relatively low due to the late opening of Arm1-In-PC. In addition, this reduction of pump flow affects arm speed as shown in Figure 10 during “Arm-In” operation. Consequently, human operator can feel intuitively that relative speed difference result from change of design parameters in the HILS simulator.

![Figure 7 Changing of Spool Open Area (Design Parameter)](image1)

![Figure 8 Bucket Tip Trajectory](image2)

![Figure 9 Pump Pressure & Flow](image3)

![Figure 10 Arm Cylinder Velocity](image4)

5 Conclusions

In this study, excavator HILS simulator is proposed to evaluate controllability of leveling motion. A real-time simulation model of the excavator is developed for middle class open-center system to simulate not only single motion but also leveling motion. The hydraulic and dynamic parts are modeled using AMESim. And, they are reduced to the extent that it
satisfies real-time capability and simulation accuracy. Then, the model is verified with measured data of real machine. To build an operator environment, Remote Control Unit is used. In addition, Unity based excavator virtual reality is applied to enhance sense of reality. Finally, Human-In-the-Loop-Simulation is established interfacing real-time simulation model and operator environment through VeriStand and PXI controller. As a result of the validation, developed HILS simulator is effective to evaluate leveling motion.

6 References

LNU-Fuzzy Network as a Mathematical Adaptive Model of a Hydraulic System

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Abstract: Model adaptive controllers such as Model Predictive Control or Model Reference Adaptive Control need a precise mathematical model of the controlled system adaptable in real-time. Systems consisting of a hydraulic 4-way proportional valve and a linear motor have non-linear behaviour such as hysteresis of a linear hydraulic motor and valve, death zone of a valve spool, time delay of a data transfer and control unit, dependence on coils temperature and oil temperature and nonlinear flow characteristics. This paper introduces modified Neuro-Fuzzy network as a mathematical adaptive model of a hydraulic system with above mentioned properties. The paper presents the basic architecture of Neuro-Fuzzy network which consists of artificial neural units a fuzzy layer and introduces modifications focused on identification. The basic real-time learning method such as Normalized Gradient Descent is introduced specially for the designed Neuro-Fuzzy Network. Identification and real-time learning abilities of the model were tested on the hydraulic stand.

Keywords: Neuro-Fuzzy network, 4-way proportional valve, adaptive model

1 Introduction

Systems consisting of a hydraulic 4-way proportional valve and a linear hydraulic motor are usually controlled by PID or their modifications. Designed controller has to fulfil requirements for the control criteria and also has to be designed with robust control for changing behaviour of the controlled system. Using classical control strategies can result in following:

- Each controller for each machine has to be set separately.
- The controller of machine has to be reconfigured during the machine lifetime.
- There are requirements for linearity of the systems components.

All above mentioned points increase the price of the machine or the price for their service. That’s the motivation for designing nonlinear adaptive controllers for hydraulic systems. Nonlinear adaptive controllers such as Model Predictive Control (MPC) [1] or Model Reference Adaptive Control (MRAC) [2],[4] are able to change their parameters during process according to changing systems behaviour. They are also able to partly calculate with nonlinear systems. Using nonlinear adaptive controllers is not meant to promise high precision of controlling but it promises low price of machines.

Nonlinear adaptive controllers need a precise mathematical model of the controlled system adaptable in real-time. This article presents the basic architecture of Neuro-Fuzzy network which consists of artificial neural units [5],[6] and fuzzy layer, and introduces modifications focused on identification of hydraulic systems. The basic real-time learning method such as Normalized Gradient Descent [7] is introduced specially for the designed Neuro-Fuzzy Network. Identification and real-time learning abilities of the model were tested on the hydraulic stand.

2 Properties of 4-way proportional valve

One of the most important property of the valve is their flow characteristic, dependence of flow through the valve on the spool position or current to the coil. We expect that spool is much faster than the hydraulic motor so it is possible to neglect dynamic of the spool. The
flow through the valve isn't directly proportional to the current in the coil for following reasons:

- Position of the spool depends on force-current characteristics of the coil and the temperature of the coil.
- The channel opening is not directly proportional to the position of the spool.
- The flow depends on Bernoulli's equation.
- The Figure 1 presents measured flow-characteristics of the valve.

There is obvious offset depending on minimum opening of the channels, hysteresis depending on the spool and the magnet friction and nonlinear characteristics.

Figure 1- Current-Flow Characteristic of a Valve Hydraforce SP08-47C- Spool-Type, 4-Way, 3-Position, Closed Center

3 LNU-Fuzzy Model

The main advantage of Artificial Neural Network is their universality of use and their ability to learn. But they are not as suitable for using in real-time process including in simple control units because they can be too large and their learning can be too computationally demanding. Fuzzy sets represent intuitive approach for when you one has mainly only theoretical or verbal knowledge. For example in our case: "The slope of the Current-Flow characteristic depends on actual current and its derivation." According that it is possible to build a space of fuzzy values. By combining the simplest neural model Linear Neural Units (LNUs) and output fuzzy layer a mathematical model was built with following advantages:

Even though the model is nonlinear it is linear according to LNUs weights, so it is possible to teach LNUs with simple optimization algorithms that search only for local minimums such as Normalized Gradient Descent (NGD).

The size of the model is smaller than neural networks with hidden layers

Building the model can be done intuitively according to technical knowledge of the system and it is also possible to use machine learning.

Linear Neural Unit (LNU) is the simplest HOUN model [8]. The formula without activation function is following:

\[
y_{i(k)} = \sum_{j=1}^{q} w_{ij(k)} \cdot x_{i(k)}
\]  

(1)

Where \( y_{i} \) is the neural output, \( w \) is a vector of neural weights. \( x \) is an input vector into LNU and for systems with 1 degree of freedom can be follows:

\[
x_{(k)} = \left[ x_{i(k)}, \ldots, x_{n(k)} \right]^T = \left[ y_{(k-ny)}, \ldots, y_{(k-1)}, \Delta u_{(k-\tau-nu+1)}, \ldots, \Delta u_{(k-\tau)} \right]^T
\]  

(2)
Where \( y \) is the vector of recent \( n_y \) samples of piston position, \( \Delta u(k) = u(k) - u(k-1) \), \( u \) is the vector of recent \( n_u \) samples of control input and \( \tau \) is the input delay of the system. The reason why \( \Delta u \) is used here instead of \( u \) is in hysteresis. Hysteresis moves the absolute value of the input signal, but \( \Delta u \) is independent of that.

The slope of the Current-Flow characteristic depends on actual input, hence the fuzzy variable was chosen as the delayed input \( u(k-\tau) \). The space of the fuzzy variable is shown in Figure 2 where each of fuzzy set \( \theta_i \) belongs to one of the LNU. The final built Neuro-Fuzzy model is in Figure 3.

Fuzzy membership \( \alpha_i \) to a given fuzzy set \( \theta_i \) is calculated as follows:

\[
\alpha_i = e^{-2n^2(\Delta u(k-1) + \tau)^2}
\]

(3)

Where \( n \) is the number of fuzzy sets and \( S_i \) is the center position of the fuzzy set \( \theta_i \). The final output of the model is calculated as a center of maxim as follows:

\[
y(k) = \frac{\sum_{i=1}^{n} y_i(k) \cdot \alpha_i(k)}{\sum_{i=1}^{n} \alpha_i(k)}
\]

(4)

LNU-Fuzzy model given be (1),(2) and (4) can be rewritten to the matrix form as follows:

\[
y(k) = a(k) \cdot W(k) \cdot x(k)
\]

(5)

Where the neural weights \( w_{ji} \) of the matrix \( W \) have to learn in real-time. For that Normalized Gradient Descent algorithm was chosen given by (6).

\[
W(k) = W(k-1) - \mu_{n}(k-1) \cdot \frac{\partial Q(k-1)}{\partial W(k-1)}
\]

(6)

\[
Q(k-1) = \frac{1}{2} e_{ref(k-1)}^2 = \frac{1}{2} (y_{ref(k-1)} - y(k-1))^2
\]

(7)

Where \( Q \) is the cost function where \( e_{ref} \) is the reference error between real output of the system \( y_{ref} \) and the output from the model \( y \). Putting (5) and (7) into (6) we get the following formula for weights adaption.

\[
W(k) = W(k-1) + \mu_{n}(k-1) \cdot e_{ref(k-1)} \cdot a_{k-1}^T \cdot x_{k-1}^T
\]

(8)

According [3] and (7) is the normalized learning rate follows for this case:

\[
\mu_{n}(k-1) = \frac{\mu}{\varepsilon + a_{(k-1)} \cdot a_{(k-1)}^T \cdot x_{(k-1)} \cdot x_{(k-1)}^T}
\]

(9)
Where $\mu$ is the learning rate and $\epsilon$ is dumping small value.

4 Identification of the Hydraulic Stand

The experimental adaptive identification was tested on hydraulic stand shown in Figure 4 with hydraulic schema Figure 5. The lower hydraulic motor controlled by 4-way proportional valve in Figure 1 with the input signal $u$. The output signal was the position of the lower piston $y$. The upper hydraulic motor was used as a constant load controlled by using pressure reducing valve. This arrangement of two hydraulic motors against each other simulating a hydraulic press. Important settings of the hydraulic stand and LNU-Fuzzy model are in Table 1.

![Figure 4 Hydraulic Stand](image)

![Figure 5 Simplified Hydraulic Scheme](image)

Table 1. Hydraulic Stent and LNU-Fuzzy Model Settings

<table>
<thead>
<tr>
<th>Learning rate $\mu$</th>
<th>Fuzzy sets $n = 20$</th>
<th>Max. force $F_{v1} = 1100[N]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu = 0.01$</td>
<td>$n = 20$</td>
<td>$F_{v1} = 1100[N]$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pressure 1 $p1 = 50[\text{bar}]$</th>
<th>Max. force $F_{v2} = 400[N]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p1 = 50[\text{bar}]$</td>
<td>$F_{v2} = 400[N]$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pressure 2 $p2 = 20[\text{bar}]$</th>
<th>Max. speed $v = 150[\text{mm/s}]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p2 = 20[\text{bar}]$</td>
<td>$v = 150[\text{mm/s}]$</td>
</tr>
</tbody>
</table>

For identification 2 data sets were measured. Output data was compared with output from the model given as a prediction from the 0.2s data. The first data are from controlling without load (without hydraulic motor V2) and the LNU-Fuzzy model was pre-trained for it in Figure 6. The second data was with the load (with hydraulic motor V2) and was tested if and how quickly the model is able to adapt to these changes. Figure 7 shows data with load and the model output trained on the data without load. The position of the contact of the pistons and higher inaccuracy of the model is visible.
Figure 6 Trained Model without a Load
- line reference data, -- line predicted output

Figure 7 Trained Model with a Load
- line reference data, -- line predicted output

Figure 8 shows model adapting to new data with load. Comparing Figure 7 and Figure 8 shows the ability to adapt to new data or new behaviour with load.

Figure 8 Adaptive Model with a Load
- line reference data, -- line predicted output

5 Conclusion

This article presented one of the options for identification of a hydraulic system consisting of a 4-way electromagnetic proportional valve and a linear hydraulic motor. Using Linear Neural Units and fuzzy sets a nonlinear model was built. The main reasoning for that is using adaptive controllers such as MPC or MRAC.

The article briefly introduces properties of a 4-way proportional valve which should be respected in the model. The article explained the basics of neural units and fuzzy sets and LNU-Fuzzy model was created. A specialized augmented Normalized Gradient Descent was used for LNU-Fuzzy model learning.

In the last section the LNU-Fuzzy model was used for identification and their retraining after changing the load on the stand simulating a machine press.

Using LNU-Fuzzy model as a model of hydraulic system brings following advantages:
- Model is adaptable during process and during lifetime of the machine and it's not necessary to retune it.
- Model is linear according to neural weights so it's able to learn by using simple local searching algorithms such as mentioned NGD.
- Model is not as big as universal neural networks with hidden layer or as a High Order Neural Units. Therefore it is more suitable for use in simple controllers.
Disadvantages of this type of modelling must be mentioned, and are following:

- The parameters of this type of the model don’t have any real meaning such as dumping, stiffness or mass. If the model doesn't work or stops working flawlessly it is hard to tell the reasons why or where the problem could be. Another problem can be brought by real-time learning.
- For using in practice it should be ensured that the model can't be retrained to worse results.

9 References


MOBILE HYDRAULICS
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Abstract: Development and research for an Open Center System (OC-System), which is used typically for excavators, has been conducted from the perspective of hydraulic efficiency. However, total system efficiency including the internal combustion engine (ICE) has not been considered thoroughly. On the other hand, a Constant Pressure System (CP-System) enabling the engine to be driven optimally is developed but is not accepted in the industry due to the complexity of the required components. Thus in this research, a hybrid system combining an OC-System with a CP-System is proposed enhancing the total system efficiency. The new system consists of an open center valve, an accumulator and a minimum of required components for the CP-System. This architecture results in a simple configuration. Moreover, this system is designed based on three basic principles according to energy efficiency. First, energy can be recuperated by using the accumulator. Second, the flow rate to the actuators is provided directly from the pumps if a mismatch between the required pressure level and the actually provided pressure level from the accumulator occurs. Third, the ICE can be operated in a high efficiency region. In order to confirm the effect of these basic principles, experiments are conducted with a test rig based on a 7t excavator with the OC-System. For the new system switching valves and the accumulator are added to the test rig and moreover, an electric motor is installed to drive pumps instead of ICE. The system efficiency is compared by measuring torque and speed of the shaft of the pumps and calculating fuel consumption using this measurement data and an efficiency map of the ICE. The test results lead to an estimated reduction in fuel consumption of 16 % compared to the conventional OC-System.

Keywords: Hydraulic Hybrid, Open Center, Constant Pressure, System Efficiency

1 Introduction
During recent years, in an attempt to improve efficiency of hydraulic excavators, a number of new hydraulic systems have been proposed in the world [1][2][3][4][5]. One of the most common valve controlled architectures is an OC-System [6]. Advantages of this system are simple configuration and high efficiency when a pump provides flow rate to only one actuator because of low throttling losses between the pump and the actuator. However, when the single pump supplies fluid to multiple actuators simultaneously throttling losses are unavoidable due to the mismatch between the pump’s pressure level, low pressure level actuators and high pressure level actuators. Moreover, in this system, the ICE is not driven optimally resulting in poor total energy efficiency.

To improve the total efficiency ifas at RWTH Aachen University proposed a CP-System, called STEAM [7][8], see Figure 1. This system consists of a large number of switching valves, two accumulators used for driving actuators and a pump for charging the accumulators. The most important feature is that the rotational speed of the ICE is fixed in a high efficiency
Moreover, by using switching valves, installed at the piston and the rod side of a cylinder and connected to the accumulators, this system can create different cylinder forces which contribute to the reduction of throttling losses and recuperate actuator energy. There are, however, some disadvantages. Since a high number of switching valves is needed, acceptance of the system in the industry is difficult.

For improving this problem, the authors propose a new hybrid system combining an OC-System and a CP-System [9]. This paper begins to introduce basic principles of the new system, and then experimental results are shown in comparison to a conventional system. In this research the OC-System is used as a reference system.

2 New hybrid architecture

In this research, a levelling cycle is used to design the new hybrid system. Figure 2 shows an outline of this cycle. This cycle consists of two motions. During the roll-in motion, the arm is pulled to the machine and the boom is lifted slightly. The next motion is roll-out and return to the initial position. These figures show the strokes of each actuator.

Figure 3 shows the hydraulic circuit of the new hybrid system. The new system consists of open center valves used as a basic hydraulic system, the accumulator and a minimum of required components which are two valves with proportional solenoids and a pressure sensor measuring pressure level of the accumulator. Moreover in order to charge the accumulator with a pump, electrical actuation applies to a proportional valve 5 in the open center valves. All valves with the proportional solenoids are controlled by a controller based on joystick
signals and value of the pressure sensor. Proportional directional valves from 1 to 4 in the open center valves are operated with hydraulic actuation depending joystick signals. By using the open center valves, this architecture results in a simple configuration.

![Figure 3 – Hydraulic circuit of new hybrid system](image)

This system is designed based on three basic principles from the perspective of energy efficiency. First, boom energy can be recuperated using the accumulator and stored energy can be provided to actuators. Thus, the pump’s power for this system can be reduced by using recuperated energy. Second, the flow rate to the actuators with quite lower or higher pressure level than the accumulator pressure level is provided directly from the pumps. For example, if the actuator with a low pressure level is powered by the accumulator, a large differential pressure between the accumulator and the actuator occurs. Moreover, if the accumulator pressure level is set to the highest actuator pressure, large throttling losses will occur between the accumulator and other actuators which have a lower pressure level. In the levelling cycle during roll-in, the arm cylinder operates in the low pressure region and during roll-out the highest pressure region. Thus the flow rate of the arm is provided by the pump directly, and the accumulator pressure level set to the lower pressure level compared to the highest pressure level of the actuator. Third, the ICE can be operated in the high efficiency region like the STEAM-System. For explanation, a simple relative efficiency map of the ICE is shown in Figure 4. Generally the high efficiency region extensively appears at lower rotation speeds than are used in today’s conventional excavators. Moreover, the ICE’s friction depending on rotation speeds can be reduced at lower rotation speeds than higher rotation speeds. Therefore, the ICE is set to a low rotation speed. The reduction of the ICE’s power resulting from altering the high rotation speed into the low rotation speed is compensated by the accumulators.
Table 1 shows which actuator is powered by the pumps or the accumulator and is decided based on three basic principles which were explained. In levelling roll-in motion, flow rate of the pump 1 and the pump 2 goes to an arm bottom side through the valve 3 and the valve 4 due to the low pressure level of the arm cylinder. For the boom, flow rate is provided by the accumulator with valve 7, and a boom rod side connects to a tank through valve 1 and valve 2 in the open center valves. Therefore an additional valve for connecting the boom rod side to the tank is not necessary. According the levelling roll-out motion, the pump 2 sends flow rate to the arm rod side with the valve 4, and the pump 1 charges the accumulator. For that, the valve 5 should be closed and the valve 6 should be opened. The pressure level of the accumulator is basically higher than pressure level of the boom bottom side because of providing flow rate to the boom from the accumulator. This means that it is impossible to recuperate the boom energy with the accumulator during a boom down motion. In order to resolve this problem, the bottom side and the rod side of the boom are connected with the valve 1 in the open center valves. This results in approx. double of the pressure of the bottom side for the cylinder of the boom. Namely, the pressure level of the boom becomes higher than the pressure level of the accumulator during the boom down motion. Thus, the accumulator recuperates boom potential energy.

<table>
<thead>
<tr>
<th>Motions</th>
<th>Pump1</th>
<th>Pump2</th>
<th>Accumulator</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll-in</td>
<td>- Arm</td>
<td>- Arm</td>
<td>- Boom</td>
</tr>
<tr>
<td>Roll-out</td>
<td>- Accumulator charge</td>
<td>- Arm</td>
<td>- Boom recuperation</td>
</tr>
</tbody>
</table>

### 3 Test rig for experimental evaluation of system efficiency

In order to confirm the system efficiency of the new hybrid system, experiments are conducted with a test rig based on the 7t excavator. In this research, only a levelling cycle is used. A digging cycle with soil is not conducted. The test rig consists of a front-end attachment of a 7t excavator, its main control valves, an accumulator (32 L), and additional valves for the CP-System. An electric motor (55 kW) is installed to drive pumps instead of an ICE. The system efficiency is compared by measuring torque and speed of the shaft of the pumps and calculating fuel consumption with this measurement data and the efficiency map of the ICE. Moreover, the actuator’s energy is calculated by measurement data of cylinder strokes and pressure levels in order to compare the system efficiency. In figure 5, the setup of the test rig is shown.
Figure 6 indicates a hydraulic circuit of the test rig. To measure value needed for calculation of the system efficiency, a torque sensor and a speed sensor are installed on the shaft of the pumps, and also pressure sensors for the pumps and the cylinders, flow rate sensors for the pumps and stroke sensors for the cylinders are added. The test rig can be operated with two modes. One is the standard mode, where only pumps and open center valves are used to drive actuators, and another mode is the hybrid mode in which the accumulator and the additional valves are used. Therefore, by using this test rig, the system efficiency of the standard and the hybrid mode can be compared. To use two modes, actuation of valves in the open center valves is changed to electrical actuation. Moreover, since the open center valves are used for the standard mode, valve 1 and valve 2 can not be used for the hybrid mode. Therefore valve 8 and valve 9 are added for the hybrid mode in the test rig.

Pump speed of the hybrid mode is reduced by 30% against the standard mode based on the basic principle. The accumulator is set to about 40% of maximum pressure, corresponding to the pressure level of the boom to provide flow rate to the boom with low throttling losses.

In figure 7 hybrid mode operations are shown in the levelling roll-in motion. Both pumps provide flow rate to the arm, and the accumulator is used to supply flow rate to the boom. In the standard mode, the pumps and the open center valves are only used. Namely in the levelling roll-in motion for the standard mode, the pump 1 sends flow rate to the boom bottom side through the valve 1, and the pump 2 provides flow rate to the arm bottom side through the valve 4. The valve 2 and the valve 3 are not used in this motion for the standard mode.
In figure 8 hybrid mode operations in the levelling roll-in motion are shown. In this motion, the flow rate for the arm cylinder is provided by pumps. The boom energy can be recuperated by the accumulator, and at the same time, the bottom side and the rod side of the boom are connected with the valve 8 to increase the pressure level of the boom bottom side. During levelling roll-out motion, the accumulator can be charged with the pump. In the levelling roll-out motion for the standard mode, the pump 1 sends flow rate to the boom rod side through the valve 1, and the pump 2 provides flow rate to the arm rod side through the valve 4.

Figure 7 – Levelling roll-in motion

Figure 8 – Levelling roll-out motion

4 Test results

The experiments were conducted with three levelling cycles. In figure 9, experimental results of three strokes are shown. The upper figure is the boom and the lower figure indicates the arm. The black line is standard mode and the red dash line shows hybrid mode. Using these measurement data, the system efficiency can be estimated.
In figure 10, a sankey diagram is shown in order to compare the system efficiency of the OC-System with the new hybrid system. The diagram indicates how much energy of diesel fuel was used for driving actuators and how much dissipated as heat losses in the machine. According to the OC-System, 66.0% of diesel fuel is dissipated as ICE losses, and also 9.5% are auxiliary and pump losses. Moreover, 15.9% are throttling losses in the valves and hoses. The OC-System cannot recuperate actuator energy, and therefore recoverable energy is also dissipated as losses. Thus, only 6.7% of diesel fuel energy is used for driving actuators. On the other hand, in the new system, 7.9% of diesel fuel energy could be used to power cylinders. The reasons for the system efficiency improvement are the efficient operation of the ICE, the reduction of auxiliary and pump losses due to the ICE’s low rotation speed and recuperation of boom energy.

Figure 10 – Sankey diagram for system efficiency
The fuel consumptions of both systems are shown in Figure 11. The result was calculated based on the ICE’S efficiency map and the pump’s shaft power which was measured by the test rig. The new system consumes 16 % less fuel than the OC-System during the levelling cycle.

Figure 11 – Comparison of fuel consumption for each system

5 Conclusions
A new system, which combines advantages of the OC-System and the CP-System, has been proposed. In particular, the new system is designed based on three basic principles, which are recuperating energy by an accumulator, providing flow rate from the pump directly and high efficient operation of ICE. In order to estimate the system efficiency and compare the fuel consumptions for the levelling cycle, the test rig was built based on a 7t excavator. Experiments with the test rig, show that the efficiency of the new system improves from 6.7 % to 7.9 % compared to the OC-System. Moreover, the experimental results show that the new system consumes 16 % less fuel than the OC-System for one sample levelling cycle. In the next phase, based on this test results a validated simulation model will be developed, and also other duty cycles will be considered with this model.

6 References


Energy Balancing for Zero Emission Excavator

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Abstract: Standard production of miniexcavator uses combustion engine. Innovative strategy for mini excavators is oriented primarily on the system with maximum use of hydraulic functions, mechanical functions and minimum of electronic components.

The main focus of this strategy is the highest efficiency of components: pumps and motors. Today‘s miniexcavator with zero emission is a complex mechatronic system with a mechanical, hydraulic, electric drive system and electronic systems. Effective synthesis of those systems needs new strategy for the control of energy flow.

The capacity of an electric battery pack is critical for the functionality of this complex system. Energy capacity of this battery pack is limited by dimension, weight and electric properties of the cells. For a better functionality, performance and efficiency of this complex system we need a new strategy with a primary focus on an energy balancing of every function within this machine.

Currently, we have enough components for the hydraulic, electric and electronic systems for building miniexcavators with zero emission. Synthesis of the machine construction requires compromise with a reducing of functionality of machine and duration of the continuous work. By adapting a new strategy with the active energy balancing we can build a construction machine that is fulfilling all of current requirements from the customer.

This article describes basic principles of the energy balancing systems for mini excavator.

Keywords: Miniexcavator, Zero emission, Hydraulic System, Electric drive, Electronic system

1 Introduction

Standard production of miniexcavator uses combustion engine. The main focus of this strategy is the highest efficiency of components: pumps and motors. Innovative strategy for mini excavators is oriented primarily on the system with maximum use of hydraulic functions, mechanical functions and minimum of electronic components. Most often, diesel engines are used for construction machines. Energy of diesel is 35,86 MJ / dm3. The efficiency of diesel engines is steadily rising. The hydraulic system is used with 1 circuit with pilot hydraulic control. The control for this system is limited to the power control against the blockage of the internal combustion engine.
2 Design

Today's miniexcavator with zero emission is a complex mechatronic system with a mechanical, hydraulic, electric drive system and electronic systems. Effective synthesis of those systems needs new strategy for the control of energy flow.

The capacity of an electric battery pack is critical for the functionality of this complex system. Energy capacity of this battery pack is limited by dimension, weight and electric properties of the cells. For a better functionality, performance and efficiency of this complex system we need a new strategy with a primary focus on an energy balancing of every function within this machine. Currently, we have enough components for the hydraulic, electric and electronic systems for building miniexcavators with zero emission. Synthesis of the machine construction requires compromise with a reducing of functionality of machine and duration of the continuous work.
By adapting a new strategy with the active energy balancing we can build a construction machine that is fulfilling all of current requirements from the customer. For effective solution, we have tools for simulations of mechanical and hydraulics subsystems:

Figure 3 Simulation model

3 Solution

The solution for this miniexcavator is complex mechatronics system based on a powerful controller communicating with human machine controlled devices, communicating with a 3-phase electric motor powered by the safe voltage of a modern 48-volt Li-MNC battery and ensuring effective control of the hydrostatic subsystem of the machine.

Energy density of cells is 200 Wh/kg.

The priorities of this system are the security and protection against overloading any of this system components and the efficient use of available energy stored in the battery pack.

Due to the multiple transformation of energy from the electric energy / battery pack / to mechanical energy / rotational speed of the electric motor / followed by hydrostatic energy / pressure and oil flow / and consequently mechanical energy / force and piston speed / acting simultaneously on the boom arm and bucket, the balancing and redistribution of energy is the most important for the efficient use of the machine and the most important condition for the decision to deploy the machine into practice.

The electric motor has been specially designed for our application with respect to the required speed and torque range for hydraulic pump while ensuring high efficiency of energy transformation.

The hydraulic pump allows you to limit the maximum working pressure with electric signal to the actual working conditions.

Electro Hydraulic distributor with flow sharing enables simultaneous operation of individual motors according to the control unit commands. The joystick and pedals are connected to the control system via the CAN BUS network. Batteries, electric motors and display machines
use their own CAN BUS connection with a control system. The power control function fulfills the control system based on the current information of the individual subsystems of the machine, supplemented by the maximum pressure on the active hydraulic motors. Our machine prototype is currently able to work in full deployment for more than 7 hours.

![Prototype of battery powered miniexcavator on tests.](image)

**Figure 4**: Prototype of battery powered miniexcavator on tests.

### 4 Conclusion

The development and use of battery electric building machines has reached the stage of effective implementation in construction practice. Extremely fast-growing area of battery drive elements enable the use of these systems not only in the field of transport equipment but also in construction machinery. A factor behind the growth of this segment is the need for the development of machine control systems including electric drive systems, hydraulic systems, mechanical systems and electronic systems. Efficient balancing of energy flows cannot be solved only by optimizing one subsystem. The complexity of the system requires the work of a strong team. Only the consistency of these subsystems enables efficient balancing of energy flows for the efficient use of limited power sources - battery packs.

### 5 References


Abstract: This contribution deals with the comparison of the energy consumption of the hydraulic control system of the telescopic excavator UDS 114. There is described the hydraulic system of the excavator, which consists of three independent hydraulic circuits. The main sources of pressure fluid in these circuits are two SPV 23 control piston pumps and one U 80L constant-flow gear pump. In order to compare the individual losses in the hydraulic circuits for controlling the working mechanisms, the hydraulic oil flow rates through the OTC H50 meter were measured. Firstly, for the machine before a repair and subsequently after the repair. For each hydraulic circuit, the hydraulic oil flow rates between the pump and the distributor and then between the distributor and the appliance were measured at first. For the subsequent calculations, it was firstly necessary to measure all the necessary geometric dimensions of the given hydraulic circuit, such as the inner diameters of the hoses, the pipes and the lengths of the lines. From the previous flow measurements in individual hydraulic circuits, the values for the losses calculation in a direct line and local resistances were obtained. All calculated and obtained values are given and described in the tables, where are always denoted individual hydraulic elements, number of pieces, calculated values of pressure losses and power losses. The overall power losses of the hydraulic circuit is then determined from particular calculations. From these calculated power losses of individual hydraulic circuits, the efficiency of individual hydraulic circuits in the state of the machine before and after repair was evaluated. The values of the efficiency of individual hydraulic circuits were then compared in the table. The resulting values in the table clearly show that the hydraulic circuits after the repair have a significantly reduced power loss and rapidly increasing their overall efficiency.

Keywords: loss, hydraulic, flow rate, pressure

1 Introduction
At present, the development of construction machinery has shifted rapidly and it is at a very high level. Higher demands are demanded on their performance, maintainability, operational costs, versatility, low fuel consumption, and so on. Satisfying these ever-increasing requirements would be unfeasible without the using of a hydraulic circuit. The hydraulic circuit can be divided into particular elements that has an individual influence on the energy losses in the flowing hydraulic oil. Because these elements are in series (consecutively) in the hydraulic circuit, we can summarize their loss influences and thus obtain the entire loss of a hydraulic circuit. If it is necessary to determine the total loss energy, it is necessary to determine it at the output of the hydraulic circuit (for the appliance). The main losses
occurring in the hydraulic circuit include pressure losses. The pressure losses are divided into three types:

- Losses, that are produced in lines, straight pipe friction losses,
- Local losses,
- Losses, that are produced in individual hydraulic elements of the circuit.

2 Material and Methods

2.1 Description of the Telescopic Excavator UDS 114

Universal finishing machine UDS 114 is the earthmoving machine mounted on the undercarriage of Tatra 815. The UDS 114 excavator is mounted on the Tatra 815 chassis by a lower frame with four pull-out stabilizing supports. The lower frame is a rotatable connected with the upper assembly. On the rotating frame the upper assembly is mounted a drive unit of the finishing machine - the four-stroke, liquid-cooled, six-cylinder engine Zetor 8701.102. This engine drives three hydraulic pumps through the drive box. One is the gear pump U 80L (Jihostroj Velešín) and two are piston control pumps SPV 23 (ZTS Dubnica nad Váhom). These allow the excavator to achieve five basic motions of the working tool:

- Lifting and lowering the external arm,
- Extending and inserting the inner telescopic arm,
- Rotation of the working tool (using a rotating head),
- Opening and closing the working tool,
- Rotating the upper assembly of the excavator UDS 114.

2.2 Hydraulic System of the UDS 114 Telescopic Excavator

The hydraulic system of the UDS 114 excavator consists of three main independent hydraulic circuits. The source of the pressure fluid are two SPV 23 control piston hydraulic pumps and one the U 80L gear pump of the constant flow rate. The first one the SPV 23 hydraulic pump delivers the hydraulic fluid through the RS 32 distributor to the linear hydraulic motor of the inner telescopic arm. The second one the SPV 23 hydraulic pump supplies the hydraulic fluid through the RS 32 (Hydracol) distributor to the linear hydraulic motors of the boom lift. The U 80L gear hydraulic pump delivers the hydraulic fluid through the RS 25 (Hydracol) distributor to:

![Telescopic excavator UDS 114](image)

1 – the basic external arm, 2 – the inner telescopic arm, 3 – the rotating head, 4 – the depth five-teeth shovel of the 0.63 m³ volume, 5 – the positional arm, 6 – the lower frame, 7 – the Tatra 815 undercarriage.

2.2 Hydraulic System of the UDS 114 Telescopic Excavator

The hydraulic system of the UDS 114 excavator consists of three main independent hydraulic circuits. The source of the pressure fluid are two SPV 23 control piston hydraulic pumps and one the U 80L gear pump of the constant flow rate. The first one the SPV 23 hydraulic pump delivers the hydraulic fluid through the RS 32 distributor to the linear hydraulic motor of the inner telescopic arm. The second one the SPV 23 hydraulic pump supplies the hydraulic fluid through the RS 32 (Hydracol) distributor to the linear hydraulic motors of the boom lift. The U 80L gear hydraulic pump delivers the hydraulic fluid through the RS 25 (Hydracol) distributor to:
The rotating hydraulic motor of the upper assembly turn HMB 630U (ZTS Brno),
The linear hydraulic motor of the working tool,
Hydraulic circuits of the boom lift and the extension of the telescopic boom (acceleration of motion),
Gear hydraulic motors of the rotating head UMN 80 (Jihostroj Velešín).

The overall schematic of the hydraulic system of the UDS 114 exceeds editing capabilities of this paper. The overall schematic is available on demand from the first author of the paper. As an example, here is a scheme of the hydraulic telescopic boom control system.

Fig. 2 – Hydraulic scheme of the control circuit of the telescopic boom extension

1 – the engine Zetor 8701.102, 2 – the control piston hydraulic pump SPV 23, 3 – the distributor RS 32, 4 – the linear hydraulic motor of the inner telescopic arm (110/70/4150), 5 – the relief valve PV 32, 6 – safety valves PV 20, 7 – the hydraulic fluid cooler, 8 – the hydraulic fluid reservoir, 9 – the lower hydraulic fluid reservoir.

2.3 Methods of Measurement
For each hydraulic circuit, measurements were performed firstly between the hydraulic pump and the distributor and then between the distributor and the appliance. As an appliance, in this case, it is considered a linear or rotating hydraulic motor. The measuring device OTC H50 (Owatonna tool company, Minnesota, USA) was used to measure the pressure (0 - 40 MPa), the flow rate (0 - 200 dm³·min⁻¹) and the hydraulic fluid temperature (0 - 120 °C) in a given hydraulic circuit.

After connecting the combined measuring device to the measured hydraulic circuit, the measurements were carried out as follows:
• Adjust the speed of the internal combustion engine to 1800 min\(^{-1}\).
• Set up the distributor for the appropriate movement of the hydraulic motor into the working position.
• Load the hydraulic circuit by means of the measuring device throttle valve to the predetermined pressure.
• Record values of the hydraulic fluid pressure, flow rate and temperature including the engine speed of the internal combustion engine.

For the testing of the hydraulic circuits between the hydraulic pump and the distributor, the final value of the preset pressure was 17 MPa, and for testing of the hydraulic circuit between the distributor and the appliance was the final value of pressure 16 MPa. Because, it was assumed that when the load is measured and simulated at the end of the hydraulic circuit at the input to the appliance at a maximum pressure of 16 MPa, the hydraulic fluid pressure will rise at the hydraulic pump output.

![Combined measuring device OTC H50 for measuring pressure, flow rate and temperature](image)

**Fig. 3 - Combined measuring device OTC H50 for measuring pressure, flow rate and temperature**

### 2.3.1 Testing of hydraulic circuit between pump and distributor

During the testing of the hydraulic circuit of the lifting the external arm and the extending of the inner telescopic arm, the inlet hose was connected to the discharge of the hydraulic pump SPV 23 and the outlet hose from the measuring device was connected to the SPV 23 pump intake to keep the circuit closed. In this testing, it is necessary to set the control lever in the cabin after starting the engine to the working position so that the tested pump has the maximum oil supply.

When testing the control hydraulic circuit of the upper assembly rotating, the working tool rotating, opening and closing the source of pressure fluid is the gear pump U 80L. The inlet hose of the measuring device is connected to the circuit behind the U 80L hydraulic pump and the outlet hose of the measuring device is connected to the supply of the RS 25 distributor.

### 2.3.2 Testing of hydraulic circuit between distributor and appliance

When testing the hydraulic circuit between the distributor and the appliance, the measuring device was connected to the hydraulic circuit instead of the appliance (instead of linear or rotating hydraulic motors). The supply hose to the combined measuring device was connected to the supply duct into the appliance and the outlet hose from the measuring device was connected to the return duct from the appliance.
3 Results and Discussion

In the calculations, it was primarily necessary to measure all the required geometric dimensions of the given hydraulic circuit (hoses and pipes inner diameters, length of the lines). From the schemes of the individual hydraulic circuits for controlling the working movements of the UDS 114 excavator, the individual elements arranged in series were determined. These elements were written to the appropriate table that is always relevant to a particular hydraulic circuit. The name of the element in the hydraulic circuit, the number of pieces, the calculated value of the pressure loss $p_z$ and finally the loss power $P_z$ are always given in each table. The pressure losses must be determined for all elements in the hydraulic circuit except the pump. These pressure losses in the individual elements of the hydraulic circuit after multiplying by a given flow rate in a given section of the hydraulic circuit mean the power loss of the element in the hydraulic circuit. The power loss will be calculated for the hydraulic pump. Summing these individual loss powers, we obtain the total loss power of the hydraulic circuit. It was necessary to calculate the pressure losses in the direct line for hydraulic hoses and hydraulic pipes according to the measured hydraulic fluid flow rates for machine states before and after the repair. Further, it was also necessary to calculate the pressure losses in the local resistances of the hydraulic circuits. For example, fittings (the conjunction between hose and steel pipe or the distributor throat), further the 90° elbow or the safety valve. Namely, for the state of the machine before and after repair. For subsequent calculations of the efficiency of individual hydraulic circuits, it was necessary to determine the values of the pressure efficiency, the geometric volume and the power consumption of pumps. Further, pressure losses for the RS 25 and RS 32 distributors that are reported by their manufacturers. From the already known values, theoretical flow rates, total efficiencies, and power losses of hydraulic pumps were calculated. The individual calculated or determined pressure losses of the elements used in the hydraulic circuits were multiplied by measured pump flow rates, for the determining the loss powers. In addition, the pressure losses of the elements behind the distributor were multiplied by the actual measured flow rate at the end of the hydraulic circuit. By this procedure, the total loss power was determined in the prepared tables for each hydraulic circuit.

Specified values:

**Gear pump U 80L:**
- The pressure efficiency $\eta_p = 0.94$
- The geometric volume $V_g = 0.08 \, \text{dm}^3$
- The pump power input $P = 39 \, \text{kW}$

**Piston control pump SPV 23:**
- The pressure efficiency $\eta_p = 0.94$
- The geometric volume $V_g = 0.089 \, \text{dm}^3$
- The pump power input $P = 45 \, \text{kW}$
Used hydraulic oil: ISO VG 46

- The density at 15 °C \( \rho = 866 \text{ kg} \cdot \text{m}^{-3} \)
- The kinematic viscosity at 40 °C \( \nu = 45.92 \text{ mm}^2 \cdot \text{s}^{-1} \)

Tab.1: Table of measured values in state before repair for the hydraulic circuit of the upper assembly rotating between the U 80L gear pump and the RS 25 distributor

<table>
<thead>
<tr>
<th>p [MPa]</th>
<th>Q [dm(^3) \cdot min(^{-1})]</th>
<th>t [°C]</th>
<th>n [min(^{-1})]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>80</td>
<td>25</td>
<td>1800</td>
</tr>
<tr>
<td>5</td>
<td>75</td>
<td>25</td>
<td>1800</td>
</tr>
<tr>
<td>10</td>
<td>60</td>
<td>30</td>
<td>1720</td>
</tr>
<tr>
<td>17</td>
<td>46</td>
<td>35</td>
<td>1660</td>
</tr>
</tbody>
</table>

where 
- \( p \) [MPa] - means the set pressure on the measuring device
- \( Q \) [dm\(^3\) \cdot min\(^{-1}\)] - means the reading of the hydraulic oil flow rate from the measuring device
- \( t \) [°C] - means the reading of the oil temperature from the measuring device
- \( n \) [min\(^{-1}\)] - means the measured speed of the Zetor engine

The theoretical flow rate of the pump SPV 23

\[
Q_t = V_g \cdot n = 0.089 \cdot \frac{1720}{60} = 2.55 \text{ dm}^3 \cdot \text{s}^{-1}
\]

The overall efficiency of the pump SPV 23

\[
\eta_{CHG} = \eta_1 \cdot \eta_p = \frac{2.3}{2.55} \cdot 0.94 = 84.74\%
\]

The power loss of the pump SPV 23

\[
P_{ZHG} = (1 - \eta_{CHG}) \cdot P = (1 - 0.8474) \cdot 45000 = 6867 \text{ W}
\]

Tab.2: Calculation of the total loss power and overall efficiencies of the boom lift hydraulic control circuit - after repair

<table>
<thead>
<tr>
<th>Element in the circuit</th>
<th>number of pieces</th>
<th>( P_z ) [kPa]</th>
<th>( P_z ) [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>piston pump SPV 23</td>
<td>1</td>
<td></td>
<td>6867</td>
</tr>
<tr>
<td>90° elbows Ø24mm</td>
<td>2</td>
<td>29.54</td>
<td>67.94</td>
</tr>
<tr>
<td>pipe Ø24mm/500mm</td>
<td>1</td>
<td>10.43</td>
<td>23.99</td>
</tr>
<tr>
<td>hose Ø25mm/500mm</td>
<td>1</td>
<td>8.59</td>
<td>19.76</td>
</tr>
<tr>
<td>fitting Ø24</td>
<td>1</td>
<td>1.16</td>
<td>2.67</td>
</tr>
<tr>
<td>distributor RS 32</td>
<td>1</td>
<td>700</td>
<td>1610</td>
</tr>
<tr>
<td>fittings Ø24</td>
<td>3</td>
<td>3.12</td>
<td>6.77</td>
</tr>
<tr>
<td>pipe Ø24mm/1750mm</td>
<td>1</td>
<td>32.97</td>
<td>71.54</td>
</tr>
<tr>
<td>hose Ø25mm/1250mm</td>
<td>1</td>
<td>19.4</td>
<td>42.10</td>
</tr>
<tr>
<td>pipe Ø24mm/1200mm</td>
<td>1</td>
<td>22.61</td>
<td>49.06</td>
</tr>
<tr>
<td>90° elbows Ø24</td>
<td>4</td>
<td>52.6</td>
<td>114.14</td>
</tr>
</tbody>
</table>

Total loss power \( P_{ZC} \) [W]

\( P_{ZHG} = 6867 \text{ W} \)
\( \eta_{CV} = 80.28\% \)
\( \eta_{CN} = 79.95\% \)

\( Q_1 = 2.3 \text{ dm}^3 \cdot \text{s}^{-1} \)
\( Q_2 = 2.17 \text{ dm}^3 \cdot \text{s}^{-1} \)
where \( p_z \) [kPa] - means the calculated loss pressure for the hydraulic circuit elements
\( P_z \) [W] - means the calculated power loss for the hydraulic circuit elements
\( P_{ZC} \) [W] - means the total calculated power loss of the hydraulic circuit

Unfortunately, the energy losses caused by the hydraulic motors are not taken into account in this paper, because during the measurement, the combined measuring device was connected at the end of the hydraulic circuit instead of the hydraulic motor. This combined measuring device loaded the hydraulic circuit to the required pressure. For each hydraulic circuit, the same table was assembled as the above Tab. 2, where the total power loss of the hydraulic circuit was evaluated.

The total power losses of the hydraulic circuits in the state of the machine before and after the repair were quantified there. Due to the limited length of the paper, the only one chosen table is presented here.

Then it was possible to calculate the overall efficiency of the individual hydraulic circuits as determined by the calculations and the overall efficiency of the hydraulic circuits determined by the measurement.

Tab.3: Comparison of loss powers and efficiencies of individual hydraulic circuits before and after repair

<table>
<thead>
<tr>
<th>Hydraulic circuit</th>
<th>before repair</th>
<th>after repair</th>
<th>Saving power loss [W]</th>
<th>Average increase in efficiency of the hydraulic circuit by [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>calculated</td>
<td>measured</td>
<td>calculated</td>
<td>measured</td>
</tr>
<tr>
<td></td>
<td>( P_{ZC} ) [W]</td>
<td>( \eta_{CV} ) [%]</td>
<td>( P_{ZC} ) [W]</td>
<td>( \eta_{CV} ) [%]</td>
</tr>
<tr>
<td>rotating the upper assembly</td>
<td>25 359.04</td>
<td>34.98</td>
<td>8 518.11</td>
<td>78.16</td>
</tr>
<tr>
<td>lifting the external arm</td>
<td>19 088.46</td>
<td>57.58</td>
<td>8 874.98</td>
<td>80.28</td>
</tr>
<tr>
<td>opening and closing the working tool</td>
<td>25 804.16</td>
<td>33.84</td>
<td>9 789.53</td>
<td>74.90</td>
</tr>
<tr>
<td>rotation of the working tool</td>
<td>26 170.95</td>
<td>32.89</td>
<td>8 305.21</td>
<td>78.70</td>
</tr>
<tr>
<td>extending the inner telescopic arm</td>
<td>18 988.98</td>
<td>57.80</td>
<td>9 752.85</td>
<td>78.33</td>
</tr>
</tbody>
</table>

where \( P_{ZC} \) [W] - means the overall calculated power loss of the hydraulic circuit
\( \eta_{CV} \) [%] - means the overall calculated power loss of the hydraulic circuit as determined by the calculations
\( \eta_{CN} \) [%] - means the overall calculated power loss of the hydraulic circuit determined by the measurement instead of the appliance

4 Conclusion

The UDS 114 telescopic excavator had approximately twelve thousand operating hours before repairing. A visible leakage of all hydraulic cylinders and visible damages of hydraulic hoses were observed on the excavator. The reduced working performance of the UDS 114 telescopic excavator has been found by the measuring of hydraulic oil flow rates in individual hydraulic circuits. Therefore, the owner of the machine has decided to carry out a more extensive repair. A total repair of the engine Zetor 8701.102 was performed on the UDS 114
telescopic excavator. Further, the U 80L gear pump was replaced by a new one. Two axial piston pumps SPV 23 have been repaired. All linear hydraulic motors have been repaired and re-sealed. A repair of the rotating hydraulic motor of the upper assembly turn HMB 630U was performed. The repair of two gear hydraulic motors of the rotating head UMN 80 was realized. Three safety valves DPV 25 and two PV 20 were repaired. Within the repair, the RS 25 and RS 32 hydraulic distributors were re-sealed. It was necessary to replace seventeen damaged hydraulic hoses and four damaged hydraulic pipes. Furthermore, eight hydraulic oil intake filters were replaced and the hydraulic oil ISO VG 46 was completely changed.

When examining the pressure losses in the straight line and the local resistances of the individual elements of the hydraulic circuits, it is obvious that the pressure losses are higher at the excavator after repair. This is due to a higher flow rate of the hydraulic oil in a given hydraulic circuit. For hydraulic circuits where the UL 80L gear pump is a source of pressure hydraulic fluid, the fluid flow rate increased by 0.935 dm³·s⁻¹ on average, after repair. However, in the case of hydraulic circuits where the axial piston pump SPV 23 is the source of the hydraulic pressure fluid, an increase of the flow rate by 0.665 dm³·s⁻¹ on average has been observed. When examining the total loss power $P_{ZC}$ of hydraulic circuits, where source of the hydraulic fluid is the U 80L gear pump, we find that the machine after repair has an average of 2.75 times decrease of the loss power $P_{ZC}$ comparing to the machine before repair. For hydraulic circuits where the source of hydraulic fluid is the axial piston pump SPV 23, the reduction in total loss of the machine after repair is the twice on average. For hydraulic circuits where the source of hydraulic fluid is the U 80L gear pump, the efficiency is increased of more than 40 %. However, in the case of hydraulic circuits where the source of hydraulic fluid is the axial piston pump SPV 23, the efficiency has been increased by more than 20 %.

5 References

HYDRAULIC SYSTEMS, MOBILE HYDRAULICS
The Modern Formulation of Hydraulic Fluids for the High Pressure Hydraulic Systems

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Abstract: The target of this contribution is to inform about the new modern formulation of hydraulic fluids developed for high pressure industrial hydraulic systems and mobile equipment. These new types of hydraulic fluids with a very high viscosity index are remarkable for their excellent technical parameters on the one hand and on the other hand for the energy efficiency and durability over time. This type of hydraulic fluids are formulated on the petroleum basis and DYNAVIS special technology with a very efficient additivation effect. The formulation of fluid is characterized by a minimal internal friction, enhanced oxidation stability and reliability compared with conventional HM or HV hydraulic oils. As it was said these new formulation of hydraulic oils with a minimal internal friction have high and very shear stable viscosity index which provide a higher lubricating film thickness, ensuring a better wear protection during all the fluid service life. They also have reinforced anti-foam properties and very good release behaviour limiting the air content of the fluid, maintaining a very low fluid compressibility and reducing cavitations issues. The article demonstrates concrete examples of hydraulic circuits in the injection presses and mobile public works machine, where due to the modern formulation of hydraulic oil with a minimal internal friction and a very high viscosity index was reached energy savings up to 5 %.

Keywords: hydraulic, fluid, energy, saving, dynavis
High Pressure Hydraulics in Diesel Engine Fuel System

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Abstract: The use of high-pressure fuel injection with ultra-fast actuators provides an opportunity for flexible molding of injection parameters, not only for the correct size of the fuel batch, but also in terms of very precise timing. These are good prerequisites for optimizing the combustion process and improving of engine performance parameters.

Keywords: fuel distribution, injection system, injector

1 Introduction

The diesel engine converts the chemical energy that is contained in diesel fuel first into heat, then into mechanical energy. For this process, it needs to provide both the flow source and the precise fuel distribution mechanism. Fuel delivery and distribution - that is, injection and rapid disintegration of fuel into small particles (droplets) - is subject to a number of parameters. The process is dependent on operating conditions, including load and angular speeds, but must be subject to stringent emission standards. It is not enough to ensure that the fuel mixture is ignited and burned up early. The process must be controlled during operation to achieve the required performance parameters with high efficiency and minimize negative environmental impacts as much as possible.

2 Fuel distribution

The flow source must overcome many resistors, including hydraulic resistors in the system, but also the pressure in the combustion chamber of the engine cylinder. However, this is not the main reason for which the fuel system is designed as high-pressure. Why, then, when the fluid mechanisms are high in load and therefore high pressure due to the problems not only of the strength but also bring other negatives - eg higher noise level of the mechanism, higher demands on sealing leaks etc.? For ignition and cultivation, accurate dosing with both time (position) and quantity is required. The dose volume of a few cubic millimeters, with a mass flow of a few milligrams, is necessary for transport to the combustion chamber at the right time and in a relatively short time interval. Increasing injection pressures is a response to higher demands - with an emphasis on efficiency and production of pollutants. The high pressure in the injection system will favorably influence both the demanding chronology of the delivery of injection - injection and the kinetics of the preparation of the mixture. Of great importance for the initiation and the actual course of combustion is the conversion of the continuous flow of fuel into the doses and the subsequent atomization - the fuel flow falls into a stream of droplets of very small diameter - in the order of micrometers. It will be appreciated that the total droplet area will be important for the rate of evaporation of the fuel, which increases with the number of elements and thus accelerates the chemical processes in the preparation of the flammable mixture. The so-called ignition delay is shortened and the dose burns faster. These are the prerequisites for the good performance of high-speed combustion engines that can achieve higher engine speeds in the past, while improving other parameters (especially reducing emissions of pollutants and noise levels).
3 Fuel delivery systems
The fuel flow is always provided by a hydrogenerator, yet high-pressure devices have a number of design variants. The basic arrangement is based on an external flow source - a high pressure generator - an injection pump. The pump is connected to a high-pressure line with the injection unit. Development has led to integration in one unit, the generator part of the associated injection unit was generally driven directly by the cam mechanism. The required injection control was provided for high-speed diesel engines via electromagnetic or piezoelectric actuators. But even this arrangement was not optimal. A somewhat different way of supplying fuel is accumulator systems with pressure accumulators (schematically shown in Figure 1). Such an arrangement is often referred to as Common Rail (CR). CR is a pressure accumulator that is continually refilled to provide high-pressure fuel availability. From there, the fuel is then distributed to the injectors to provide the required fuel dose setting. The injectors themselves are relatively complex and highly accurate electrically controlled valves. The advantage of such an arrangement of the hydraulic circuit with the CR is, in particular, during operation a constant supply of high pressure fuel as a source for operational injection control.

![Figure 1 – Hydraulic scheme of Common Rail system.](image)
from the fuel tank through the low- and high-pressure pump and the fuel tank through the hydraulic line to the injectors.

4 Injection control
Achieving high efficiency but also controlling emissions, in particular noise, vibration and pollutant emissions, is subject to strict timing of individual parts of the fuel batch. It consists of a sequence of parameters dependent on injections. The first group consists of a very small pilot dose - a pre-injection (or several pre-injections) that is initiated before the end of the piston compression stroke, ie several angular degrees before top dead center (TDC). The main injection represents a substantial fuel supply of several angular degrees behind TDC. Another additional injection (or late injection - usually two post-injection) of a small amount of fuel affects the reduction of pollutants. Injection units have been of great importance in the past. Their quality influenced both performance parameters and operational reliability. The current trend of strict limitations on consumption and other negative phenomena poses extraordinary pressure on the design and operation of the fuel system. Injection units are now relatively
complex electrically controlled elements and play a dominant role in high-pressure CR systems. It has to meet the main fuel distribution requirements. These are in particular controlled dosing sequences, which must comply with current operating parameters. That is, accurately determining the dose size, start and duration, with precision to the angular degrees of crankshaft rotation. On the timeline, the interval is very short (in fractions of milliseconds). The speed of such actuators is therefore extraordinary and the accuracy requirements are high. The response of actuators to the control signal is not entirely proportional, so signal modulation is the subject of development - the following figure (Figure 2) shows an example of a control signal modification. The characteristic in the figure shows the difference in timing and shortening the reaction time - the green characteristic shows a new (shorter) time interval for the series of injections. From the difference in the size of the areas under the curves it is possible to consider differences in the fuel supply - the adjusted signal corresponds to a higher fuel supply, even in a shorter time interval.

![Figure 2](image)

**Figure 2** – The timing of the control signal.

The main representatives of current injectors are electromagnetically operated injection units (solenoids) and piezo injectors (Figure 3). The latest in-market injectors include an electromagnetic injector with a so-called multiplier. The advantage of piezo injectors is that the individual injections are more accurate and faster due to the almost instantaneous reaction of the piezoelectric element to the change in the supply voltage. They work under difficult conditions. In addition to quasi-static loads (they work at pressures above 200 MPa), they have to face the dynamic phenomena that complicated and fast injection mode is.

![Figure 3](image)

**Figure 3** – Injection unit with piezo-actuator.

The stroke of the needle is controlled by a piezoelectric element. In the right part of the figure, the initialization process is indicated. The almost instantaneous reaction of the piezoelectric element to the change in the supply voltage guarantees fast and very accurate timing of each injection.
The demanding operating mode of the combustion engine, on the one hand, and the high demands on impeccable functionality, force designers to continuously upgrade the fuel system and its components. An example of such a modification is shown in Figure 4. The original arrangement in which the pressure force from the high-pressure circuit was used to activate the piezo-actuator was replaced by another solution. The power source for the initiation is derived from the low-pressure circuit (low-pressure - waste hydraulic branch). The load of the control mechanism can be significantly lower, which brings many advantages. These include a lower wear rate, but also a faster response.

![Figure 4 - Piezo-valve innovation.](image)

The figure on the left represents the original version (the control force is derived from the pressure in the high-pressure branch), on the right is a different design with connection to the low-pressure - waste hydraulic branch (with a pressure of approximately 1MPa).

### 5 Conclusion

Although strong pressure on the development of so-called electromobility could be a precursor to the end of a drive train with a traditional combustion engine, it is precisely thanks to the development and application of progressive systems that the required performance parameters are achieved, with significant reductions in negative environmental impacts. The diesel engine fuel system hydraulic circuit is still a major challenge for designers. By exploiting the benefits of a combination of mechanics, hydraulics and electronics, the parameters of the internal combustion engine can continue to improve.

The paper shows the trend of the development of diesel engine injection systems and points to some concrete examples (only selected examples with respect to the protection of property rights).

### 6 References


Experimental Testing of Power Assisted Steering

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Abstract: The paper is focused on the description and creating of the simulation model of the power assisted steering and testing of the functionality and efficiency on the laboratory test rig. The mathematical model is describing the structure of the system and is created from the mathematical models of the subsystems, which compute the dynamic behaviour of the steering wheel, pinion, rotating control valve and of the hydraulic cylinder. The simulation model was realized in the simulation programme MATLAB-Simulink and was used for the simulation of the testing experiment. The measurements were done on the laboratory test rig, which is controlled by the computer and operated by the swivel pneumatic drive instead of the driver hand. It allows better repeatability of the experiment. The experiments were focused on the evaluation of the efficiency of the hydraulic power assisted steering system.

Keywords: Power assisted steering; hydraulic drive; mathematical model; simulation;

1 Introduction
The paper deals with the creating of the mathematical model, simulation and experimental testing of the hydraulic power assisted steering for a passenger car. After the analysis of the structure of the hydraulic power assisted steering system the mathematical models of the subsystems are described. Finally, the whole simulation model built in the simulation programme MATLAB-Simulink which consists of the mechanical and hydraulic subsystem is introduced and achieved simulation results are presented. The experimental work, simulations, experiments and the measurements on the test rig were focused on the evaluation of the efficiency of the power-assisted system for different driving situations characterized by different load torque. To have the opportunity to steer the steering wheel repeatedly in the same quality, the steering wheel is operated by the pneumatic swivel drive controlled by the proportional valve and digital controller. The presented results were obtained using the computer-controlled experiment and allow to demonstrate the principle and effectiveness of the hydraulic power assisted steering system.

2 Mathematical Model of Hydraulic Power Assisted Steering
The hydraulic power assisted steering is a system designed to lower the driver’s steering effort. The system can be divided in two main subsystems - the hydraulic servo system, which is connected in parallel with the mechanical steering system. The driver is operating using the steering wheel the mechanical subsystem. The steering wheel is connected with the steering rod terminated with the torsion bar. The second end of the torsion bar is connected with the pinion. The main components of the hydraulic drive are the control valve, the hydraulic cylinder and the fixed displacement pump directly driven by the engine supplying the system with compressed oil. The control valve is an open center valve and typically rotating construction.
The function of the valve can be described using the Wheatstone bridge with hydraulic resistances, Fig. 1 [2].

![Fig. 1. Structure of the hydraulic power assisted steering system](image)

The driver that produces the torque operates the steering wheel [2,3,5,6]. Due to the stiffness of the torsion bar, the difference between the angle position of the steering wheel and the angle position of the pinion is the variable, which defines the valve displacement. The flows to the hydraulic cylinder chambers depend on the valve displacement and pressure drop on each control edge. Finally, the force produced by the hydraulic cylinder is added to the force produced by the driver on the steering wheel and transmitted using the steering rod, torsion bar and pinion on the piston rod. The sum of these torques, or forces is acting on the wheels.

The mathematical model of the power assisted steering system copies the structure of the system and is created from the equations modelling the mechanical subsystem and equations describing the behavior of the hydraulic subsystem [4].

![Fig. 2. Structure of the power assisted steering system – mechanical subsystem](image)

The mechanical subsystem is described using two motion equations. The first described the steering wheel characterized by the mass with inertia \( J_v \) doing the rotational motion.

\[
J_v \ddot{\phi}_v + B_v \dot{\phi}_v + K_T (\phi_v - \phi_p) + M_{f,v} = M_v
\]  

(1)

The connection between the steering wheel and the pinion is realized by the torsion bar modeled by the torsion spring with the torsion stiffness \( K_T \), Fig. 2. \( J_t \) is the inertia of the steering wheel and steering rod, \( B_v \) is the viscous friction coefficient of the steering wheel, \( M_v \) is the torque
produced by the driver’s action on the steering wheel, $M_{fr}$ is the friction torque of the steering wheel. If the friction is very low, the friction torque can be neglected, and the motion equation becomes the form

$$Jv \ddot{\varphi}_v + B_v \dot{\varphi}_v + K_T(\varphi_v - \varphi_p) = M_v$$

(2)

The stiffness of the torsion bar can be calculated using the formulas

$$K_T = \frac{G J}{l}, \quad G = \frac{E}{2(1+\mu)}, \quad J = \frac{\pi d^4}{32}$$

(3)

where $G$ is the shear modulus, $J$ is the quadratic torque of the cross-section area of the torsion bar, $l$ is the length of the torsion bar, $E$ is the Young’s modulus in pulling, $\mu$ is the Poisson number of the steel, $d$ is the diameter of the torsion bar.

The second motion equation is describing the behaviour of the pinion and has the form

$$J_p \ddot{\varphi}_p + B_p \dot{\varphi}_v + K_T(\varphi_v - \varphi_p) - p_p S \cdot r_p + p_L S \cdot r_p + M_z = 0$$

(4)

where $J_p$ is the inertia of the pinion and rack, $B_p$ is the damping coefficient of the pinion, $M_z$ is the external load torque. The friction torque of the pinion is neglected.

The hydraulic circuit of the power steering assisted system is shown in Fig. 1. The flows $Q_{L1}$, $Q_{L3}$, $Q_{P1}$, $Q_{P3}$ through the hydraulic resistances in the Wheatstone bridge are given by the formulas

$$Q_{L1} = \alpha \cdot S_L(\varphi) \sqrt{\frac{2}{\rho}} \sqrt{|p_S - p_L|} \cdot \text{sgn}(p_S - p_L)$$

(5)

$$Q_{L3} = \alpha \cdot S_P(\varphi) \sqrt{\frac{2}{\rho}} \sqrt{|p_L - p_T|} \cdot \text{sgn}(p_L - p_T)$$

(6)

$$Q_{P1} = \alpha \cdot S_P(\varphi) \sqrt{\frac{2}{\rho}} \sqrt{|p_S - p_P|} \cdot \text{sgn}(p_S - p_P)$$

(7)

$$Q_{P3} = \alpha \cdot S_L(\varphi) \sqrt{\frac{2}{\rho}} \sqrt{|p_P - p_T|} \cdot \text{sgn}(p_P - p_T)$$

(8)

$\alpha$ is the losses coefficient, $\rho$ is the oil density, $p_s$ is the system pressure, $p_T$ is the tank pressure and $p_P$ and $p_L$ are the pressures in the cylinder chambers. $S_L$, $S_P$ are the cross-section areas of the openings of the control valve, which depend on the difference between the angular position of the steering wheel and pinion $S_p(\varphi)$ and $S_L(\varphi)$ and are given by the functions
Two equations describing the hydraulic capacity in the left and right chamber of the hydraulic cylinder allows to compute the pressures $p_L$ and $p_P$ as follows

\[
\dot{p}_L = \frac{K}{S(L/2-x)} (Q_{L1} - Q_{L3} + S\dot{x} + Q_{SP})
\]  
(11)

\[
\dot{p}_P = \frac{K}{S(L/2-x)} (Q_{R1} - Q_{R3} - S\dot{x} - Q_{SP})
\]  
(12)

Where $K$ is the bulk modulus, $S$ is the cross-section area of the piston, $L$ is the stroke of the hydraulic cylinder, $x$ is the actual position of the piston, $Q_{SP}$ is the leakage flow.

The introduced mathematical model of the power assisted steering system represented by the equations (1) ÷ (12) was programmed in MATLAB-Simulink. Different steering situations were simulated using the created simulation model. The simulation model of the whole system is shown in Fig. 3, [2].

![Fig. 3. The simulation model of the power assisted steering system.](image)

3 Test Rig and Experimental Testing of the Power Assisted Steering System

The test rig of the power assisted steering system shown in Fig. 4 was built in the laboratory [2].
The steering wheel is operated by the swivel pneumatic drive. To achieve better repeatability of the experiment the steering wheel is operated by the swivel pneumatic drive instead of the driver hand. The angular position of the shaft of the pneumatic drive is controlled using the proportional pneumatic valve and the digital control system SPC200. The piston rod of the hydraulic cylinder is connected using the force transducer with the mechanism producing the friction force, which models the forces acting on the front wheel in the car during various driving manoeuvres. It is possible to tune the magnitude of the produced force using the screw. Important variables are measured and the values transmitted in the computer for the next evaluation.

The experimental testing was focused on the evaluation of the efficiency of the power assisted steering system by driving the car. The driver’s manoeuvres in the moose test as a good comparable and repeatable driver’s action has been chosen, Fig. 5. The test simulates the situation, when an animal – for example a moose, or a child is rushing out onto the road and the driver is trying to avoid it [1,2,5] and can be described as follows. The driver is driving the car in the right line and changes the line to the left line and back. Typically, the test is repeated with increased speed.

The course of the piston position of the power assisted steering system during the moose test at the speed of 60 km/h is shown in Fig. 6, [1]. The course of the angular position of the steering wheel has the similar shape due to the fixed connection between the piston rod and steering wheel. For that reason, the similar profile was defined for the desired value of the position angle of the pneumatic drive, which operates the steering wheel of the test rig instead of the driver.
The course of the steering wheel position during the moose test is shown in Fig. 7.

The moose test was repeated on the test rig for different magnitude of the friction force. Using the measured variables, the torque produced by the pneumatic drive, the torque produced by the hydraulic cylinder and acting on the pinion and finally the load torque on the pinion were evaluated. The results of the measurements on the test rig are summarized in Fig. 8 and Fig. 9, [2].

Fig. 8. Torque of the pneumatic drive (upper plot), torque on the pinion from the hydraulic cylinder (middle plot) and load torque (down plot) for different load (different colours), experimentally on the test rig obtained values (left) and values obtained using the simulation experiments (right).
Fig. 9. Comparison of the torques obtained experimentally on the test rig (black line) and using simulation (red line).

The contribution to the comfort of the driver when steering the car can be demonstrated using the bar graph in Fig. 10, where the driver’s torque is depicted by blue colour, the green bar shows the contribution of the power assisted steering system, and the orange colour is used for the load torque. The results are shown for five different magnitudes of the load torque. The upper graph summarized the experimental results, and the simulation results are shown in the lower graph.

Fig. 10 Comparison of the activity of the power assisted steering system for different loads. Experimental data – upper graph, simulation results – lower graph. Orange – maximum load torque, green power assisted steering system contribution, blue maximum driver’s torque.

The simulation and also the measurement results demonstrate the increasing contribution of the hydraulic cylinder in dependence on the increased load torque. The driver’s torque is practically
the same. This is the effect of the hydraulic power assisted steering system. Higher load will result in increased contribution of the power assisted steering system, while the driver’s effort stays practically constant.

4 Conclusions
The paper presents the experiments realized with the simulation model in MATLAB – Simulink and experimental test rig of the power steering system for a passenger car. The presented test rig was realized in the laboratory and allows demonstrating the function of the power assisted steering system, to measure the important variables determining the state of the system and evaluating the efficiency of the system. To achieve repeatedly the same results of the experiments, the steering wheel is operated by the pneumatic drive controlled by the proportional valve and digital control system.

The experimental testing was done for the driver’s maneuver in the moose test. The driver’s action was simulated, and also experimentally realized on the test rig. The simulation and the experimental results are shown in plots, and the contribution of the power assisted steering system was evaluated for five various loads. The outputs of the experimental work evaluate and quantify the contribution of the power assisted steering system to the comfort of the driver by driving the car.

The results are summarized in the bar graphs separately for the experiments on the test rig and for the simulation results. It allows also the evaluation of the good quality of the simulation model. The evaluation of the effect of the power assisted steering system on the driver’s activity can be used for the comparison of the different steering systems used in cars. The test rig is a part of the laboratory of mechatronic systems and allows students and researchers experimenting with the real system to study the principles of the power assisted steering, to make measurements and design some improvements.

5 Acknowledgment
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Tensile Resistance of the Combine Harvester with Tracked Concept of Chassis

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Abstract: This article is a continue of tensile resistance measuring of combine harvesters. After detection of tensile resistance for combine harvester with wheeled chassis there was measured tensile resistance also for combine harvester with tracked concept of chassis. For consequently comparison there was used the similar machine (due to same weight, same drivetrain mechanism and transmission, etc.) in this case it was the combine harvester John Deere S685i with track units instead front wheels. The same like at wheeled concept of chassis also here was determine of tensile resistance did based on drawbar pull. For reading of tension force there was used specially developed measurement tool - a pull dynamometer. It is constructed as a towing drawbar which was connected between rear hitch of combine harvester and three-point hitch of pulling tractor. It means that measuring set moved in reverse mode. Measuring was running on the same path as in case of wheeled combine harvester: asphalt surface, maximum slope 0.2°, length of path 120 meters. Speed of towing simulated usual working speed during harvest. There were measured three variants of speed it were 4, 6 and 8 km.h⁻¹ and each variant was repeated three times. After collecting of data for each speed variant consequently there were calculated different tensile resistance which we can observe. Tensile resistance correspond with load of drivetrain, load of engine and it directly affects fuel consumption. Measured data are also valuable for consequently comparison between wheeled and tracked chassis concept of combine harvesters.

Keywords: combine harvester, tensile resistance, tracks, travel gear

1 Introduction

For agricultural machinery are put more and more demands on performance [1]. Raper [2] reported that efficient mechanization in agriculture is a major factor underlying high productivity. Larger machinery is often related with timeliness, higher work rates, and lower labour requirements. The drawback of it is that larger machinery usually means increased machinery weight which increases the danger of soil compaction. Soil compaction affects the physical, chemical, and biological properties of soils and is one of the main causes of agricultural soil degradation [3].

Manufacturers of agricultural machines are trying to solve this problem by using wide low-pressure tires with low pressure on the ground. Or second way is installing the track units with belts on the machines.
The rubber tracks has an effects on tractive force, rolling resistance, torque, tractive coefficient, and tractive efficiency under different soft terrains [4]. Changes in these parameters can be easily observed in the change of tensile force. When characteristic of tensile force is influenced by ground. Bauer [5] discloses the characteristics of tensile for the various ground surface in Figure 1. The tracks are also reflected in the change of tensile force at different speeds. Tracks assembly and weight balance shows Figure 2.

Current knowledge of draught force could be a useful tool in many ways. The results can be used in routine practice to compare the energy performance of travel gear of self-propelled machines, verification of technical changes on machines and verification of agronomical measures [7].

For comparison is impotant to know that hydraulic components in hydrostatic system of travel gear at both machines are the same. Measuring of performance parameters at hydraulic systems (pressure, flow, etc.) is very beneficable for their compare and it shows energic
consumption of both type of chassis. Theoretical losses of hydraulic system was published in the past. Total power which is lost in hydrostatic system of combine harvester travel gear is 16.95 kW [8]. It means that at constant flow (constant speed) pressure changes depending on the load. Load is given many parameters and one of them is tensile resistance which will be discussed below.

2 Materials and methods

Field measurements took place in Nové Strašecí in Central Bohemia. In affiliated workshop of agricultural company Školní statek Lány, ČZU. The measurements were taken in 5th of March 2016. Combine harvester was dragged on the asphalt surface. During the measurement surface was wet and the ambient temperature around 4 °C. The measured path was 120 meters long with an average incline of 0.1 degrees.

To measure of the tensile draught force was used the combine harvester John Deere S685i without header. The weight of the machine was 16400kg and type of tires and their pressures are in Table 1. Combine harvester ravel gear was decommissioned by using disconnecting axle shafts due to mechanical resistance of gearbox and differential.

Table 1. Parameters of tracks and tires

<table>
<thead>
<tr>
<th></th>
<th>Front tracks</th>
<th>Rear tires</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maker</td>
<td>Camso</td>
<td>Goodyear</td>
</tr>
<tr>
<td>Dimensions</td>
<td>700 mm width</td>
<td>620/75 R26</td>
</tr>
<tr>
<td>Pressure</td>
<td>-</td>
<td>0.30 MPa</td>
</tr>
</tbody>
</table>
Figure 3 - Measuring set. From left: pulling tractor John Deere 7930, measuring instrument, pulled combine harvester John Deere S685i without header

For actual measurement was used measuring instrument of draught force developed in collaboration of Czech University of Life Sciences and BEDNAR FMT ltd. As a pulling tractor means served John Deere 7930. Combine harvester was dragged back in order to facilitate connect the measuring equipment (Figure 3).

Basic part of measurement apparatus was strain gauge load cell S-38 with measuring range up to 200 kN. The load cell was necessary to place into a steel cage so that the forces were applied only in tension or compression. Bending of the load cell may cause its destruction. The load cell was calibrated on a stationary workplace. Calibration was carried out on tensile testing machine ZDM 50t. The data from load cell were sensed every 2 s into the laptop which was situated in the cabin of the tractor. Measuring equipment was complemented by hinges for mounting between a pair of machines (Figure 4).

Figure 4 - Measuring equipment between combine harvester and tractor
The measurements were made for alternative speeds 4, 6 and 8 km•h⁻¹. These speeds simulates normal range of operating speeds, which combine harvester moves on the land at work. For each speed were always carried two repeats.

### 3 Results and discussion

Calibration results and calibration curve can be seen in Figure 4. Linear dependence of measuring apparatus output frequency on tensile force was proved. Resulting linear dependence was used as calibration equation for draught force calculation [9].

![Graph showing calibration curve](image)

Figure 4 - Dependence of measuring apparatus output frequency on tensile force. Load cell calibration curve.

The graph in Figure 5 shows that the tensile force values for the individual travel speed have similar values. When measuring at higher speeds (6 and 8 km•h⁻¹) is a problem of high variance of values. This is due to impacts due to the inertia during the measurement. Sensor these values recorded and these are after processing, appear as outliers and extremes.

![Box plot showing tensile force](image)

Figure 5 - Dependence of tensile force on pulling speed
This fact affects subsequent statistical evaluation with using of Fisher LSD test. Results of Fisher LSD test are given in Table 2. Fisher's test confirms the assumption that among the values of the tensile resistance are not statistically significant differences. Nevertheless in the average values of tensile resistance is visible trend of gradual increase in tensile resistance.

Table 2. Results of Fisher LSD test

<table>
<thead>
<tr>
<th>Speed [km·h⁻¹]</th>
<th>Tensile force [kN]</th>
<th>Average</th>
<th>Fisher LSD test</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>5.82</td>
<td>****</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>6.57</td>
<td>****</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>6.56</td>
<td>****</td>
<td></td>
</tr>
</tbody>
</table>

4 Conclusion

Performing of tensile test of the combine harvester has been found that at operating speeds 4, 6 and 8 km.h⁻¹ was not found significant difference in tensile resistance. The average value of tensile resistance at operating speed 4 km.h⁻¹ was 5.82 kN. At speeds 6 and 8 km.h⁻¹ were the average values of tensile resistance higher and almost the same (6.57 and 6.56 kN).

Higher tensile resistance needs higher torque of hydromotor. It means that increase pressure in high-pressure circuit of hydrostatic system of combine harvester travel gear. It is reason for higher energetic consumption of hydrostatic system and consequently higher fuel consumption.

In comparison with measuring of combine harvester with wheeled chassis these values are higher. But comparison of two different concept of combine harvester will be published in next article where will be stated exact values and reason of difference.

9 References


[7] Service training material of Deere & Company


HYDRAULIC AND PNEUMATIC SYSTEMS
Abstract: The performance of a mechanical filter is an implicit function of many variables pertinent to fluid condition, flow variables, filter element condition and operational parameters. This paper presents the details and results of a parametric study that examined the effect of oil temperature, contamination level and flow rate on the performance of a hydraulic filter through laboratory experiments. A 5 μm rated commercial filter with glass fiber made element that had an effective surface area of 0.154 m² through 57 pleats was used to filter VG32 hydraulic oil. The filtrate was supplied with the contaminant of ISO medium test dust at four gravimetric levels - 2, 5, 8 and 10 mg/L of oil. The tests were conducted at the flow rates of 40 and 120 L/min for different oil viscosities, corresponding to the temperatures of 30, 40, 50 and 60° C. As the temperature increases, the oil viscosity decreases due to weakened cohesive forces, which leads to increased filtration rates and hence more time to build the pressure, upstream of the element. On the other hand, the pressure on upstream of filter bed builds up at higher rate when the filtrate has higher level of contamination loading. An extensive investigation on the effect of flow variables and oil condition parameters on the pressure drop across the element would therefore give a better knowledge about filter element lifetime.

Keywords: Hydraulic filtration, temperature, flow rate, gravimetric level, pressure drop

1 Introduction

Solid contamination in the lubrication oil causes premature wear of mechanical components and adversely affects their life. Maintaining a quality oil film over the solid surfaces that are under potential contact is crucial to realize the anticipated longevity of the moving parts. In addition to reduced effect of friction, an effective lubrication can decrease the temperature of the contacting surfaces, thanks to heat transfer capability. Furthermore, a quality lubrication ensures a seal between the liner wall and the segments, and keeps the internal surfaces cleaner. However, the hydraulic oil is subject to inevitable contamination, because of not only the friction between the engine components, but also the uncertain and unmatrixed environmental and operational conditions. This is usual with marine propulsion units, where the solid contamination in the hydraulic oil contains soot, sand, and metal particles. This explains the need for filtration.

Pressure drop across the porous media is the prime motivator of the oil flow through the filter. While it is usual to display an approximately linear relationship between the pressure drop across the element and clean oil flow rate, the presence of particles in the oil has an appreciable influence on the pressure difference as well as other filtration indicators (Bémer and Callé 2000;
The flow pattern, where the particles of different shapes and sizes fall in a wide range of motion regimes, and their deposition on fibrous media in the filtration applications, constitutes a multiscale physical system. The effect of particulate loading on the pressure drop across the filter element is therefore exclusive and case dependent.

Flow rate, along with throughput and degree of particle removal, is a parameter of interest in studying the filtration. The flow rate of liquids is inversely moderated by the viscosity, which in turn has an inverse relationship with temperature. Although Darcy’s law gives a theoretical explanation about the effect of flow rate on the pressure drop across the porous media, this relationship exceptionally depends on the application. For example, see Lage et al. 1997; Gisinger et al. 2015; Saleh et al. 2016. The objective of this study is to characterize the oil pressure drop, $\Delta P$, across the filter element as a function of temperature, flow rate and contaminant gravimetric level, using laboratory experiments. A test bench with a series of sensors and instruments was developed to make the empirical observations on the variations of $\Delta P$ for different filtrate properties. The trends of $\Delta P$ obtained from the experiments will be useful to develop the correlations between oil condition parameters and flow variables.

2 Experimental setup

Figure 1 shows the laboratory test bench, developed for the parametric investigation on the hydraulic oil filtration, and Figure 2 presents the respective hydraulic network with usual notations and symbols. This network mainly consists of two circuits; one for injecting the solid contamination of medium test dust (ISO12103-1-A3) at predetermined rates, and the other for passing the contaminated oil through the test filter. A 5 μm rated commercial filter with glass fiber media having an effective surface area of 0.154 m² through 57 pleats was used as a standard test filter. ISO VG 32 oil, which has a specific gravity of 0.86 and kinematic viscosity of 32 cSt at 40° C, and 5.4 cSt at 100° C, was used as the working fluid. The positive displacement pumps were used to deliver the specific amounts of test dust into the oil tank, and contaminated oil to the filter. The filter element was subject to a constant circulation of oil at a deterministic temperature and flow rate. The contaminant was introduced continuously until the
pressure drop across the element reached a net value of 5 bar. The sensors and instruments used on the test bench included:

- Two pressure sensors, installed on the upstream and downstream of the filter assembly
- A pressure differential sensor, to measure the pressure drop across the filter element
- Two particle counters, connected at the upstream and downstream of the filter assembly, to ensure the right amount of particle injection and unfiltered amount of contaminant, respectively
- A flowmeter, to ensure the predefined flow rate through the filter
- Temperature sensors, to measure the oil temperature in the tanks

Cleaning mode was activated after each test run in the injection and test loops to avoid the influence of the residual solids in the oil and circuit lines of previous run on the present. Care was taken to avoid leaks in the circuit and operational obstructions, and to control the flow variables related to oil condition.

![Schematic diagram of hydraulic circuit (top), and explanations of symbols (bottom)](image)

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>C&lt;sub&gt;1&lt;/sub&gt;, C&lt;sub&gt;2&lt;/sub&gt;</td>
<td>Coolers</td>
</tr>
<tr>
<td>F&lt;sub&gt;1&lt;/sub&gt;, F&lt;sub&gt;2&lt;/sub&gt;</td>
<td>Cleaning and testing filters</td>
</tr>
<tr>
<td>H&lt;sub&gt;1&lt;/sub&gt;, H&lt;sub&gt;2&lt;/sub&gt;, H&lt;sub&gt;5&lt;/sub&gt;</td>
<td>Three-way ball valves</td>
</tr>
<tr>
<td>H&lt;sub&gt;3&lt;/sub&gt;, H&lt;sub&gt;4&lt;/sub&gt;</td>
<td>Two-way ball valves</td>
</tr>
<tr>
<td>H&lt;sub&gt;6&lt;/sub&gt;, H&lt;sub&gt;7&lt;/sub&gt;</td>
<td>Flow regulators</td>
</tr>
<tr>
<td>M&lt;sub&gt;1&lt;/sub&gt;</td>
<td>Gear motor (SEW EURODRIVE R32DT71D4, 0.55 kW)</td>
</tr>
<tr>
<td>M&lt;sub&gt;2&lt;/sub&gt;</td>
<td>AC MOTOR (ABB M2AA90L, 2.5 kW)</td>
</tr>
<tr>
<td>M&lt;sub&gt;3&lt;/sub&gt;</td>
<td>Gear motor (Magnetek Speed+, 3VZ132S4, 5.5 kW)</td>
</tr>
<tr>
<td>N&lt;sub&gt;1&lt;/sub&gt;, N&lt;sub&gt;2&lt;/sub&gt;</td>
<td>Pressure indicators</td>
</tr>
<tr>
<td>P&lt;sub&gt;1&lt;/sub&gt;</td>
<td>Allweiler A6 Pump (ADP 153 B21 P01, Q = 0.4 – 0.7 L/min)</td>
</tr>
<tr>
<td>P&lt;sub&gt;2&lt;/sub&gt;</td>
<td>Allweiler A6 Pump (NB 25-200/ϕ185, Q = 30 L/min)</td>
</tr>
<tr>
<td>P&lt;sub&gt;3&lt;/sub&gt;</td>
<td>Pump (Aktiebolaget D4F052K1)</td>
</tr>
<tr>
<td>T&lt;sub&gt;1&lt;/sub&gt;, T&lt;sub&gt;2&lt;/sub&gt;</td>
<td>Temperature sensors in tanks t&lt;sub&gt;1&lt;/sub&gt; and t&lt;sub&gt;2&lt;/sub&gt;, respectively.</td>
</tr>
<tr>
<td>t&lt;sub&gt;1&lt;/sub&gt;, t&lt;sub&gt;2&lt;/sub&gt;, t&lt;sub&gt;3&lt;/sub&gt;</td>
<td>Tanks for particle-supply, particle-oil mixing and overflow, respectively.</td>
</tr>
<tr>
<td>V&lt;sub&gt;1&lt;/sub&gt;, V&lt;sub&gt;2&lt;/sub&gt;</td>
<td>Check valves</td>
</tr>
<tr>
<td>VM&lt;sub&gt;1&lt;/sub&gt;, VM&lt;sub&gt;2&lt;/sub&gt;</td>
<td>Flow meters (SCFT-300-32-07 of Parker and VLA-ZF03 of Kytola with 10-300 L/min and 0.1-1 L/min range, respectively)</td>
</tr>
</tbody>
</table>
3 Results

3.1 Effect of oil temperature
Viscosity is one of the critical properties of oils that determine their suitability for lubrication. Temperature has a direct influence on the oil viscosity, and hence the particle transport and filtration. In the limiting case, the lubricant may form wax at low temperatures, and undergo a rapid oxidation at high temperatures. Effect of temperature on the viscosity varies from oil to oil; the lesser the variation, the more effective to lubricate. In the present study, Cannon-Fenske routine viscometer, which is a modified Ostwald type, was used to assess the effect of temperature on the selected hydraulic oil, VG 32. Figure 3 shows the viscosity-temperature relationship. The viscosity varies between 51 and 16 cSt for the temperature range of [30° C, 60° C], with a mid-point viscosity i.e. at 40° C of 32 cSt. The filter performance was contrasted within this temperature range for two different flow rates, and corresponding pressure drop characteristics are depicted in Figure 4. Inverse relationship between the fluid viscosity and filtration rate is observed from slower rise of pressure drop across the element at higher temperature. However, this effect of temperature on the pressure rise before the element is not linear. At low flow rates, say 40 L/min, the difference between the ΔP curves at 30° C and 60° C is diverging as the element gets loaded with particles, whereas this trend is converging for the flow rate of 120 L/min. As seen from Figure 4, oil flow rate has a significant effect on the element lifetime for a given gravimetric level of solid contamination, which led us to extend the experiments to the investigation on the effect of filtrate flow rates on the pressure drop across the element during filtration.

![Figure 3 – Viscosity change of ISO VG 32 oil with temperature](image1)

![Figure 4 – Effect of oil temperature on ΔP across the filter element for 10 mg/L contamination load](image2)

3.2 Effect of oil flow rate
For a given solids’ gravimetric level, the particles accumulate faster at higher flow rates. As a result, the pressure builds up in lesser time on the upstream side of the element, compared to the cases of low flow rates. Several statistical models are available for pressure drop across the porous media for homogeneous and heterogeneous particle accumulation (Xiao et al. 2012; Qin and Pletcher 2015). However, the complexity of practical hydraulic filtration systems with inherent uncertainties in the design and operating conditions explains the reason for paucity of similar research. In this study, the filtration was tested at two flowrates for different particle loading; 40 and 120 L/min. Corresponding pressure drop curves are plotted in Figure 5. Three major observations were made here. First, three times higher flow rate caused the pressure to rise before the element approximately three times faster. This relationship between the flow rate and pressure drop held true in all cases of contaminant loading. Second, the transition of ΔP characteristics from low-slope regime to high-slope was more gradual at higher flow rates than at lower flow rates. This explains that the characteristic time of particle accumulation is of the comparable order of time scales of flow through porous media. Finally, the contaminant
loading in the oil has an appreciable effect on the filter element saturation. A linear increase in the gravimetric level of the solids does not lead to a similar trend for element clogging.

3.3 Effect of contaminant gravimetric level
To investigate the filtration trend of the element as a function of contamination loading, multiple gravimetric levels of test dust in the oil were used in separate tests for a given oil flow rate and temperature. Pressure difference between upstream and downstream of the element against four gravimetric levels of contaminant for a flow rate of 120 L/min is plotted in Figure 6. Although an identical trend of pressure drop is evident in all cases, there is a noticeable difference in the filter element lifetime for different particulate loading. The flow rate decreases with increasing concentration of solid particles due to increased inertial forces, which results in faster clogging of the element. Higher gravimetric levels of contaminant cause the effective flow area of the porous media to decrease, and hence a quicker pressure rise on the upstream side of the element for a given flow rate. As seen from the curves, to reach the net pressure drop of 5 bar, 2 mg/L solid concentration in the oil at 60º C took nearly 49 min, which was 4.5 times of the case of 10 mg/L. Higher flow rates with higher temperature and larger particulate loading have insignificant effect on the pressure drop across the cleaner element, which is reported through the overlapping of ΔP curves at 50 and 60º C during the initial phase of filtration shows for 10 mg/L gravimetric level.
4 Conclusions

The process of filtration is associated with several complex physical phenomena, which demonstrate the interdependency of numerous fluid and solid properties. As a part of continued research on the analysis of filter element’s lifetime, the present study attempted to develop the data sets that reveal the effect of different flow variables on the performance of filter element. The changes in oil temperature and therefore the viscosity affect the flow rate as well as the particles’ motion. The time for the element to clog proportionally varies with the oil temperature. Increasing the oil flow rate speeds up the solids accumulation on the element. In relation to the degree of contaminant gravimetric level, the evolution of pressure drop curves is faster at high particle loading. Solid contaminants experience higher drag force as the oil viscosity increases. Further studies are under way to develop the correlations between these variables to formulate the ΔP trends to predict the element’s residual lifetime.

5 Acknowledgements

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6 References

Machine Tool Vibration Reduction Using Hydrostatic Guideway

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Abstract: Utility properties of the machine tools, such as machining accuracy, surface quality, and productivity are affected by many factors that also include guideway properties. Guideways enable machine tool parts to move. It is generally understood that hydrostatic guideways exhibit better damping properties than linear guideways with rolling elements. However, quantitative expressions of better damping appear in the literature very sporadically. Therefore, this paper presents a model of hydrostatic guideway damping, that include squeeze damping of the thin layer of oil. Furthermore, hydrostatic and linear guideways damping properties are compared on a model example of a large machine tool. Results indicate that hydrostatic guideway reduces the forced oscillation amplitude of the first eigenfrequency 15 times in case of the modeled machine tool.

Keywords: hydrostatic guideway, squeeze damping, vibrations, machine tool

1 Introduction
Utility properties of machine tools (MT), such as machining accuracy, surface quality, and productivity are also affected by damping of Machine tool structure [1]. Damping can be increased by manufacturing structural parts from cast iron or composite material [2]. Damping improvement is also achieved by various part fillings, e.g., aluminum foam and glass balls [3]. Another source of damping are guideways that movably connect machine tool parts. Linear guideways (guideways containing rolling elements) exhibit lower damping in comparison with hydrostatic (HS) guideways [4]. This article assesses improvement of dynamic properties of large vertical milling machine equipped with hydrostatic guideways. Forced oscillations amplitude of the ram tool center point is studied. A process of milling induces dynamic forces that lead to machine tool structure vibrations. Damping dissipates the energy of vibrations and reduces vibrations amplitude. The higher is damping the smaller is vibration amplitude. This paper assesses whether hydrostatic guideways significantly reduce vibration amplitude. The paper also proposes a methodology to compare different kinds of guideways concerning damping.

2 Model description
This chapter proposes a methodology to compare HS and linear guideways concerning damping. Furthermore, the chapter describes the damping model of HS guideways and FE model of studied machine tool ram.
2.1 Approach to guideway comparison

The operating principle of linear and HS guideways is rather different. Linear guideways make use of several rolling elements that recirculate in a guideway carriage to enable linear movement of machine parts. Rolling elements are small balls or rolls made of steel or ceramics. Rolling elements connect two sliding parts and are permanently in contact. Thus, vibrations are easily transferred thru linear guideways [5]. Rolling elements are elastic bodies with similar stiffness but the very low capability of damping.

The HS guideway comprises a rail (prism) and HS pocket. The pocket is shown in Figure 1 consist of a cavity and a land. The cavity is supplied with externally pressurized oil that flows out of the cavity thru the narrow gap between the land and the rail. The pressure of oil over the pocket area provide load carrying capacity. HS pocket and the rail are permanently separated by a thin layer of oil. Sliding parts are not in contact and energy of vibrations is dissipated in the thin layer of oil. HS pocket and the opposing surface of the rail are referred to as an HS cell, and the narrow gap is also referred to as a throttling gap.

Next paragraph discusses significant design parameter that enables us to compare two guideway types. Operating life of linear guideways depends highly on guideway type, load, preloads, environment and lubrication and can vary largely. On the contrary, the operating life of HS guideways is almost not limited because the surfaces of rail and pocket are not in mechanical contact. Therefore, service life is not a suitable parameter for guideways comparison. Installation dimensions are not the convenient parameter since one carriage of linear guideway can carry both radial and lateral forces while one HS pocket can carry only radial force in one direction. So, the design requirement is very different and not suitable for comparing. Operation of HS guideways requires energy whereas linear guideways are passive components. The friction of HS guideways is approaching to zero at low speeds. On the other hand friction coefficient of linear guideways equal approximately 0,01. Therefore, comparing guideways concerning energy is not suitable. Load carrying capacity appears to be sufficient parameter even though, load carrying capacity of linear guideways depends on service life. Stiffness is the beneficial parameter for evaluation of mathematical model results. Two guideways with the same stiffness have equal eigenfrequencies. Then resonance oscillation amplitudes can be compared and damping evaluated. Thus, it is beneficial to compare two guideways with equal stiffness and load carrying capacity and reasonable operating life.

2.2 Damping model of HS guideways

Damping of thin lands can be described by equation (1) [7], where dimensions of HS pocket are $a = 81 \text{mm}$, $b = 81 \text{mm}$, $l = 16,3 \text{mm}$ and pump pressure equals $p_p = 50 \text{bar}$. Dimensions are clear from Figure 2. Computed damping of one HS pocket is $5,6 \cdot 10^5 \text{Ns}^{-1}$.

$$b_{HS} = \frac{\eta A_l^2}{h^3} = \frac{\eta d l^3}{h^3},$$  \hspace{1cm} (1)
where \( \eta \) - means dynamic viscosity \\
\( A_L \) - denotes the area of land \\
\( l \) - is the width of land \\
\( h \) - means the thickness of oil layer 

To support radial loads in both directions, two HS pockets are required, and thus damping is also double.

### 2.3 Model of machine tool ram

The machine tool ram is three meters long with square cross-section 300 × 300 mm with the wall thickness of 30 mm (Figure 3). HS pockets or carriages are located at cross-slide in the distance of 800 mm. The tool is located at the lower end of the ram and, its vibrations in the direction of the \( Y \)-axis are examined. An excitation force is applied at the tool center point in the direction of \( Y \)-axis. The ram is mode modeled beams in 2D space and describes bending and axial displacement. Carriages and HS pockets are replaced by springs and dampers. A ball screw for positioning of ram is also replaced by the spring and the damper \((k_3, b_3)\).

![Figure 3 – Model of machine tool ram](image)

For analysis, the linear guideway is designed for machine toll ram with the service life of five years in five-day two-shift operation. The suitable linear guideway is designated BMA 30 with ball elements and preload V3 supplied by Schneeberger. Load-deformation graph of one carriage is shown in Figure 4. Derived linearized stiffness equals 640 N/\( \mu \)m. Damping of the linear guideway is very small and therefore the damping of the machine tool with linear guideways is assumed to be structural damping 1 % [8]. The ram is made of steel. In the analysis, the damping is modeled as Rayleigh damping.

![Load-deformation graph](image)
The HS guideway is designed with equal stiffness and load carrying capacity as linear guideway. Thus two hydrostatic pockets stiffness equal $640 \, N/\mu m$. Load-carrying capacity, pocket pressure, stiffness, oil flow and required power are shown in Figure 5.

The designed thickness of oil film layer equals $50 \, \mu m$. Regulation of the oil film thickness is performed by capillary regulator.

3 Calculated results
Calculated transfer curve is shown in Figure 6. The curve values are divided by a value of static compliance $9.5 \cdot 10^{-8} \, m/N$. Therefore, all values greater than zero indicate that dynamic deformation is greater than static deformation and vice versa.
The harmonic force is applied in the horizontal direction at the tool center point and deflection of tool center point is also calculated in horizontal direction. The deflection amplification of the first eigenfrequency is greater in the case of the linear guideway. It is assumed that phase is not essential for machining accuracy and surface quality and therefore it is not plotted.

An amplitude of tool center point forced oscillations is depicted in Figure 7. Driving force equals 1000 \(N\). The amplitude of the first resonant frequency for MT with linear guideway is 4417 \(\mu m\) whereas the amplitude of MT with HS guideway equals 282 \(\mu m\). The amplitude of MT with linear guideway is 15 times higher.

Calculated vibration amplitude of the first eigenfrequency is written in the Table 1.

<table>
<thead>
<tr>
<th>Amplitude ([\mu m])</th>
<th>HS guideway</th>
<th>Linear guideway</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amplitude ([\mu m])</td>
<td>282</td>
<td>4417</td>
<td>15.6 (\times)</td>
</tr>
</tbody>
</table>

4 Conclusion

This paper compared hydrostatic and linear guideways concerning dynamic properties on the example of the large machine tool vibrations. The paper assessed the impact of higher damping of hydrostatic guideways on forced oscillation amplitude of the tool center point. The amplitude of tool center point was calculated by the FEM model of the deformable ram and stiffness and damping model of guideways. Results indicate that hydrostatic guideway reduced the forced oscillation amplitude of the first eigenfrequency 15 times.

For future work, calculated transfer functions can be used for estimating limit chip thickness. Then in general for assessing whether it is beneficial to use hydrostatic guideway instead of the linear guideway. It is also planned to experimentally verify the dynamic model of hydrostatic guideways experimentally.
Acknowledgment
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References
The Use of Differential Gearing in Gear Pumps for to Reduce Pulsations in Hydraulic Systems

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Abstract: Under the programme NGGP (New Generation of Gear Pumps) obtained from the agency of TAČR, there were evolved gear pumps with reduced pressure pulsations. The target is to reduce pressure pulsations into half, compared with standard gear pumps with external gearing. Differential gearing is divided into three or more parts. This way splitte d gear wheels partly eliminate pulsations of separated toothed parts. Pressure peaks of each individual toothed part interfere with peaks from the other parts what decreases the pulsation. This system requires the accuracy of angle offset between gear parts. The reduction of pressure pulsation brings pump noise reduction too. These pumps can be used in systems that require less pulsation as special devices or mobile power steering systems

Keywords: Gear pump, Pressure pulsation, Differential gearing

1 Introduction

Each gear pump in the hydraulic circuit generates pressure pulsations that are determined by its design. The fluid in the gear pump is guided in inter-tooth gaps which are separated by individual teeth. Due to this break of liquid pass, the flow pulsation and therefore pressure fluctuations occur. Subsequent fluctuation in flow causes engagement of driven and drive wheel. The contact of involute grooving occurs here as well as the gradual uneven reduction of inter-tooth area. This results in further fluctuation in flow and consequently in pressure fluctuations in system. In most applications it is not necessary to solve the pump pulsations size. However, there are also systems where it is necessary to reduce the pulsation to a minimum. Here, the vibration dampers are still used. As an example of such application it can be the pump of power steering for trucks, buses or agricultural and construction machinery. Here, the vibrations can be transmitted to the vehicle's steering wheel

2 Possibilities of pulsations reduction from the pump

It follows from above mentioned that the number, the arrangement and the shape of the teeth have the major effect on pulsations size. However, with the higher number of teeth, the unit geometric volume decreases and therefore the width of the gear ring increases for the same geometric volume of the pump. Along with the width of gear ring, the effect of pulsations from gearing starts to increase. This suggests that the narrow gear ring with the large number of teeth would be ideal. However, this is not technically possible. There are several ways to reduce the pulsation of the gear pumps. One possibility is to adjust the teeth shape. The shape can be modified at first by way of teeth narrowing. In this case, the finished involute remains from the one side of the tooth. The second side is cut so that the narrowed teeth can not be in contact. The second variant of teeth modification is the
possibility of special gearing with so-called infinite dump. Here the involute gearing is completely abandoned.

Another possibility is apparent increase of teeth number. This means that the gearing is divided into two parts by standard way. This principle, so-called duo-pumps, is known for over fifty years and it is widely used by gear pumps manufacturers. However, there is no significant reduction in pulsations. Another way used is the dual flank contact (DFC). Here is the contact on both tooth involutes. This divides the inter-tooth area into two parts and there is a decrease in pulsations.

3 Differential gearing

Differential gearing follows the duo-pump type. However, the gear ring is divided into three or more parts here. The development was mainly devoted to so-called triple pump. The pump follows the T3 pumps series, which falls into the size group II of gear pumps. Here is the biggest demand for pumps with reduced pulsation. Standard T3 series gearing were used, where the gear ring has 12 teeth. The angle between the teeth of the gear set is therefore 30 °. The division to three parts thus simulates the gear pump with 36 teeth gearing. Teeth offset is always exactly about 1/3 of the angle between teeth. The angular rotation of second gear set against the first one is 10 °, of third one then 20 °. The first gear set (numbered from the shaft connection) is milled directly onto the wheel. Other gear sets are then pushed on wheels as floating wheels. Connecting the floating wheel with the drive wheel is via the key. The groove in the slide wheel is always exactly under the tooth head. The groove for spring on the drive wheel is then always rotated by the appropriate angle. On the driven wheel, the slide wheels are then only pushed, not secured by the spring. The individual parts of gearing are then separated by balancing plates to avoid sealing the fluid in the individual parts of gear set. These pumps are then identified with commercial name T3T.

4 Prototype testing and measurement of characteristics

For the first functional prototype, the geometric size of 16ccm was chosen, which is typical for II. Group of gear pumps. After performing the functional tests, comparative testing with
the standard pump was carried out in the field of pulsation testing. The test parameters were as follows:

- geometric volume $V_g = 16$ ccm
- body with threaded ports on inlet $G \frac{3}{4}$ and outlet $G \frac{1}{2}$
- design variants: triple and T3 (standard)
- output pressure: $p = 240 - 250$ bar
- speed: $n = 500$ and $1500$ rpm
- temperature: $t = 45^\circ C$

The measurements were made by piezoelectric pressure sensors with the scanning frequency of 1000 Hz. From the results it can be observed that pulsations of standard T3 series pumps are in the range of 6 bar. This in itself is creditable result that proves the excellent setting of T3 gearing series. The usual values of similar pumps can reach up to 10 bar. In the triple-pump version, the pulsations range up to 3 bar.

For the long-term tests, the most important tests were selected. In both cases the lifetime of sleeves and bearing faces and at the same time the lifetime of balancing plates were tested. Parameters of durability tests were as follows test:

- target: 10,000 cycles
- long pressure cycles "on/off" 180/20 sec
- output pressure $p_2 = 260$ bar
- speed $1500$ rpm
- oil temperature $45 \div 50^\circ C$ (ISO VG46 oil)

The pump has run in total of 10,035 cycles without significant wear. And despite of slight decrease in volume efficiency, the pump was capable of further operation.

- target: 10,000 cycles
- constant pressure load
- output pressure $p_2 = 240$ bar
- speed 3,200 rpm
- oil temperature $70 \div 75^\circ C$ (ISO VG46 oil)
- target: 300 hours
The pump was running for a total of 306 hours. After the long-term test, no significant wear of parts was found. The pump was assembled and used for further short-term measurements.

5 Restrictions arising from the differential gearing construction
Due to its design, some parameters had to be limited compared to the standard T3 pump series. The maximum continuous pressure of pumps and maximum temperatures were limited. This is mainly due to division of work space into three smaller separate workspaces. It increased the friction areas in the pump. This leads to limitation when there is a significant oil heating at high pressures and high revolutions. Then it can lead to pump seizure. At the same time, the mechanical and volume efficiency are reduced due to the increasing of friction surfaces in the pump. Each of efficiencies is lower by about 2-3 % points. Thus, the volume efficiency is ranging about 90-92 %, the mechanical one is then about 85 % depending on operating modes.

6 Impact of reduced pulsations to gear pump noise
The noise of gear pumps is mainly formed by above-mentioned pump pulsations. The decrease of pulsations is therefore also related to noise reduction. The advantage of T3T pump compared to other types of low pulsation pumps is subjective perception of high-speed noise. The pump noise does not shift to higher frequencies here. High speeds are then without "screaming"

In noise comparison with standard T3 pump it was made the significant reduction in noise. Pumps of 16ccm geometric size were compared.

7 Conclusion
Thanks to differential gearing the pulsations have dramatically been reduced. And to less than half size. At the pressure of 250 bar, the pulsations size of 6 bar for the T3 standard pump was reduced to less than 3 bar for T3T differential gearing pumps. At the same time noise was reduced, where the equivalent noise pressure level was lower by approximately of 3-7 dB depending on pump operating parameters.

These results led to rapid introduction of differential gearing into commercial sales. Today, the pump is used in hydraulic control circuits of agricultural machines. Now, the most utilization is in aggregates with low noise requirements.
Fig. 5 - graphical presentation of T3 and T3T pump noise measurements

8 References

Measuring of Rotary Air Motors Characteristics

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Abstract: In the article the construction of an experimental device for measuring the characteristics of small rotary air motors is described. Further a measurement methodology and measured data processing is explained. At the end of the article using of the measured characteristics for mathematical modeling is presented.

Keywords: air motor, measurement, mathematical model

1 Introduction
Rotary pneumatic motors can be divided into several categories by motor design. In this case we can divide the rotary pneumatic motors in following basic categories: vane, axial and radial piston, gear and turbine design. The objects of our research were mainly small radial piston motors with power up to 500 W and nominal speed up to 1000 min⁻¹. An example of design of these motors is shown in Figure 1. These motors are used in many branches of industry (food industry, chemical and pharmaceutical industry, paper and textile production etc.) where they are used for mixing, lifting, as a conveyor drive and many other applications.

![Figure 1 – Radial piston air motor [1]](image1)

![Figure 2 – Torque and power characteristics of motor [1]](image2)

During the rotary air motors characteristics measuring it is necessary to determine the following parameters: an inlet pressure and eventually an outlet pressure, a motor torque, a rotation speed and an air consumption. The processing the measured data is described below. The result of measurements is the rotary motor characteristic, which is mainly a torque and power dependence on a rotation speed, Figure 2. This characteristic may be complemented by
2 Test equipment, measurement methodology and results

During the measurement of rotary motors it is necessary to change the load of the output shaft continuously. This can be achieved by some types of brakes as an electrodynamometer, hydrostatic or hydrodynamic brake and also friction brake. For the motors with power up to 500 W the friction brake is quite sufficient. Therefore bicycle break has been chosen. Specifically type ZEE BR-M640 from Shimano in combination with disc SM-RT 66 with a diameter of 160 mm was chosen. According to the producer the brake was tested for rotation speed up to 1350 min⁻¹ and guaranteed braking power is 800 W.

To measure a torque the sensor T22 / 50 from HBM with a measuring range of +/− 50 Nm was chosen. Furthermore, the pressure sensor PR 15 with a measuring range of -1 to 6 bar, the speed sensor DS 03 and the recording device M5050 were chosen, all made by company Hydrotechnik. The flow measurement was realized by the sensor SD 6000 from IFM electronic with measuring range 4-1250 dm³·min⁻¹ (ANR). A frame to which were mounted all the necessary parts was also designed and built. The device is shown in Figure 3, a circuit diagram is in Figure 4.

1 - flowmeter, 2 - pressure sensor, 3 - recording device, 4 - reduction valve, 5 - torque sensor, 6 - speed sensor, 7 - FRL unit, 8, 9 - ball valve, 10 - pneumatic rotary motor, 11 - silencer, 12 - disc brake

The basic requirement for the measurement is to keep a constant pressure at the inlet of the motor. Therefore, the pressure regulation was realized by proportional pressure valve VPPE-3-1-1 / 8-10-010-E1 made by Festo. By selecting the control voltage the measurements were then carried out at a pressure level of 3, 4, 5 and 6 bar.

At each pressure level the torque was increased using the friction brake and after each change a record of all measured variables (pressure - \( p \), torque - \( M \), speed - \( n \), flow rate - \( Q_N \)) was made for 3 seconds with sampling period of 0.1 second. From the measured values the mean values were calculated. These were directly plotted into the graph, or were used to calculate the power \( P \), the overall efficiency \( \eta \) and specific consumption \( \bar{m} \). An equation for the calculation of individual parameters is shown in equations 1 to 3.

The motor mechanical power on the output shaft can be calculated from the torque and speed.
The motor overall efficiency is the ratio of mechanical output power to pneumatic input power. Due to the volume flow rate dependence on the pressure, it is necessary to recalculate the flow rate to the level of working pressure.

\[
\eta = \frac{P}{p \cdot Q_p}
\]  

(2)

where \( P \) – power [W], \( p \) – working pressure [Pa], \( Q_p \) – volume flow rate of compressed air \([m^3.s^{-1}]\).

Specific air consumption is air consumption related to output power. In this case, it is necessary to use flow rate recalculated to normal atmospheric conditions (ANR – atmospheric normal reference).

\[
\bar{m} = \frac{Q_N}{\bar{P}} \left[ dm^3 \cdot min^{-1} (ANR) \cdot W^{-1} \right]
\]  

(3)

where \( \bar{P} \) – power [W], \( Q_N \) – air consumption \([dm^3.min^{-1} (ANR)]\).

Figure 5 shows the torque characteristic of the measured motor. The torque significantly decreases from a maximum value at zero speed (starting torque). The value of the starting torque depends on the size of the pressure. At the inlet pressure of 3 to 6 bar the starting torque varies from 2 to 3.8 Nm. From the course of the power in Figure 6 it is obvious that the maximum power is achieved in the range of revolutions from 1100 to 1200 min\(^{-1}\).
Figure 7 shows the course of overall efficiency of the measured motor. The maximum efficiency for different pressure values is in the range of 48-55%. These values are achieved in the speed range from 500 to 750 min⁻¹.

From the above it is clear that the area of maximum efficiency is shifted in comparison with the area of maximum power to the lower speed. Generally, the optimum working range of the motor is located between maximum efficiency and maximum power. The maker states that the optimum range is from 700 to 900 min⁻¹, which corresponds with the measured values.

As mentioned above, the torque decreases depending on the speed. Theoretically, the torque does not depend on the speed, but the problem is the pressure. With increasing speeds, the work space of the motor is not filled enough, see Figure 8. This leads to a reduction in the pressure and hence the torque. Into the calculation of parameters then enters so called filling efficiency. Figure 9 shows course of measured motor filling efficiency.

If we know displacement of the motor, the real value of torque can be calculated from equation 4 and the power from equation 5, more see [2, 3]

\[
M = \frac{1}{2\pi} \cdot V_g \cdot p_{it} \cdot \eta_p \cdot \eta_m
\]  

\[
P_s = V_g \cdot n \cdot p_{it} \cdot \eta_p \cdot \eta_m
\]

where

- \(V_g\) – displacement of motor [m³],
- \(p_{it}\) – theoretical indicated pressure [Pa],
- \(n\) – speed [min⁻¹],
- \(\eta_p\) – filling efficiency,
- \(\eta_m\) – mechanical efficiency.
3 Modeling of a pneumatic system with a rotary motor

In addition to the measurements we carried out verification of the possibility of mathematical modeling of pneumatic systems with the rotary motor. The Matlab-Simulink Simscape was used for modeling. Simscape contains models of the basic pneumatic components including the rotary air motor model "Rotary Pneumatic Piston Chamber". Into this model, it is necessary to specify Displacement (volume per unit angle), Initial angle, Dead volume and Chamber orientation (direction of rotation). The model consists of three following equations [4].

The continuity equation is

\[ G = \frac{V_0 + D \cdot \theta}{R \cdot T} \cdot \left( \frac{d\rho}{dt} - p \cdot \frac{dT}{dt} \right) + \frac{D}{R \cdot T} \cdot p \cdot \frac{d\theta}{dt} \]  

where
- \( G \) – mass flow rate at input port [kg s\(^{-1}\)],
- \( V_0 \) – initial chamber volume [m\(^3\)],
- \( D \) – piston displacement (volume per unit angle) [m\(^3\) rad\(^{-1}\)],
- \( \Theta \) – piston angle [rad],
- \( p \) – absolute pressure in the chamber [Pa],
- \( R \) – specific gas constant [J kg\(^{-1}\) K\(^{-1}\)],
- \( T \) – absolute gas temperature [K],
- \( t \) – time [s].

The energy equation is

\[ q = \frac{c_v}{R} \cdot (V_0 + D \cdot \theta) \cdot \frac{d\rho}{dt} + \frac{c_p}{R} \cdot p \cdot \frac{d\theta}{dt} - q_w \]  

where
- \( q \) – heat flow due to gas inflow in the chamber [Js\(^{-1}\)],
- \( q_w \) – heat flow through the chamber walls [Js\(^{-1}\)],
- \( c_v \) – specific heat at constant volume [J kg\(^{-1}\) K\(^{-1}\)],
- \( c_p \) – specific heat at constant pressure [J kg\(^{-1}\) K\(^{-1}\)].

The torque equation is

\[ \tau = p \cdot D \]  

Mechanical, flow neither filling efficiency is not included into the mathematical model of the motor. It causes that the simulation results are not good. The torque at zero speed corresponds to the real motor but the torque characteristics depending on the speed do not decrease linearly. Larger differences are then in power characteristics. Maximum calculated power is several times greater than the real power and it is also achieved at higher speeds. With the above model we have not achieved real results.

The manual for Simulink (Help) contains another model, "Pneumatic motor". There is chosen a completely different approach of motor parameters definition. Into the model it is necessary to enter vector of rotational speeds, vector of torque values, vector of volumetric flow rates and pressure differential at which the torque and flow data were measured. By this way the
characteristics of the motor can be very precisely defined. The disadvantage is in obtaining these values by measurement.

An example of measured and calculated values is in Figure 10. In this case, at the time of 1.4 s the valve has opened. At the time of approximately 2 s the steady speeds was achieved. During the acceleration the torque has risen to 1.9 Nm due to the inertia of the rotating parts, after the acceleration the torque reached the level of 1.1 Nm.

Subsequently, the accuracy of the simulation was validated in the case where during the rotation the load was increased, for example from 0.7 Nm to 0.95 Nm. This change in torque has resulted in the decrease of speed from 1245 min⁻¹ to 980 min⁻¹, see Figure 11.

**Figure 10 - Torque during motor acceleration**

**Figure 11 - Speed during load increase**

### 4 Conclusions

One of the disadvantages of pneumatic rotary motors is the change in speed depending on load variation. It is therefore advisable to know the characteristics of the motor when choosing a motor for various applications. The characteristics are listed in the motor's catalogs. By the help of software, such as Matlab-Simulink, based on the measured characteristics it is possible to predict the behaviour of the system, for example, when changing the torque, moment of inertia, etc. It has been mentioned in the article that the model “Rotary Pneumatic Piston Chamber” does not deliver so good results. If this basic model would supplemented mainly with filling efficiency, better results can be achieved. In the future, we will continue to address this issue.

### 5 References


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