IMPROVING THE EFFICIENCY OF A ROCK BREAKER HYDRAULIC WORKING SYSTEM BY CHANGING THE STRUCTURE OF THE HYDRAULIC SYSTEM

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The amount of energy losses in hydraulic systems, determined directly by the efficiency parameter, becomes one of the basic aspects related to the design of these systems. Recently, reducing the energy consumption of hydraulic drives has become one of the most important factors in, and sometimes even the main reason for, development works focused on hydrostatic drive systems in newly designed and already deployed types of machines. The article presents an approach to improve the efficiency of such a hydraulic system based on the analytical model and also includes a case study of redesigning a hydrostatic drive system in a stationary rock breaking machine. The analysis of the calculation results showed benefits resulting from two different concepts of modifying hydraulic systems with an aim at reducing their energy consumption.

KEYWORDS

hydraulic drive efficiency; energy consumption reduction; hydraulic machinery

1 INTRODUCTION

Progress in the field of hydraulic systems has been recently observed in a number of areas. Their newly developed elements allow the systems to operate at a decreased mass and at increased working pressures. A number of research tasks have focused on describing the dynamic phenomena which, in a broader perspective, allow improved reaction times of such systems and a better control over changing pressure and flow values in the system [Padovani 2020]. The potential for applying means of electronic control to hydrostatic drive systems has also been significantly increased [Li 2021; Hui 2011]. Some works have also focused on improved descriptions of thermal phenomena in hydraulic systems [Siwulski 2021]. Research into hydraulic systems has been also additionally motivated by some external factors, such as growing energy costs and the challenges related to the efforts to mitigate climate changes, which are addressed in the European "Green New Deal" [Mahato 2020]. Therefore, designing hydraulic systems has become a multi-dimensional process, which requires a broad perspective and clearly defined goals corresponding to the requirements related to operating safety, energy efficiency, power demand, noise emissions, control accuracy, as well as to the application-specific requirements, such as the potential for encasing the system. As a result, the development of such systems entails an unconventional approach to their architecture and a broad analysis of the influence of such variables as liquid properties, system operating temperature, flow rate, the dynamic and static parameters of individual components in the system, as well as the change speed of control signals etc. [Ma 2020; Zimerman 2007]. In recent years, a significant number of research works dedicated to increasing the efficiency of hydraulic systems have focused on analyzing the potential for developing hybrid electric-hydraulic drive systems [Li 2021; Wang 2014; Qu 2021].

On the other hand, research on improving the control systems comprises works describing system efficiency gains resulting from the soft switching of a directional valve [Mahato 2017], as well as works on improved control algorithms in hybrid systems [Mahato 2020]. Another important publication addresses the genetic-algorithm and neural-network based methods of identifying reasons for increased energy consumption [Ma 2020].

Systems allowing energy recovery may be based on hydraulic accumulators [Ho 2010; Do 2012], on a flywheel [Aljohani 2014; Sonsky 2019] or on other methods [Minav 2013]. In the literature, much attention has been paid to modifying the designs of hydraulic elements with an aim to increase their efficiency [Rituraj 2021; Śliwinski 2019] or to employ new elements previously not used in typical hydraulic systems [Wu 2020].

Extensive research and deployment works carried out at the Mechanical Faculty of Wrocław University of Science and Technology resulted inter alia in a new method for designing or redesigning the already existing systems in order to ensure they meet a broad range of increasingly higher requirements. This paper discusses a representative analysis of the energy consumption in a hydrostatic drive system and some advantageous modifications to its architecture implemented among other things in order to reduce its energy demand. The research is based on data obtained from a previously designed drive system for a rock breaker type URB ZS-3.

2 THEORETICAL GROUNDS FOR ARCHITECTURE MODIFICATIONS AIMED AT INCREASING THE EFFICIENCY OF HYDRAULIC SYSTEMS

2.1 *Types and classification of losses in hydraulic systems*

The basis for an energy-efficiency analysis of a system is to identify the amounts and locations of energy losses, which is understood as the partial conversion of hydraulic energy into heat. In hydraulic systems, such losses are classified as volumetric losses, related to leaks in the system, pressure losses, related to the energy cost of the liquid flowing through individual system components, and mechanical losses in the actuators and in the pumps. The parameter describing the ratio of losses to the transferred power is defined as efficiency (η), which in the case of hydraulic drive systems is the ratio of mechanical power obtained from the hydraulic actuator or motor (P_{out}) to the mechanical power on the pump shaft (P_{in}).

$$\eta = \frac{P_{out}}{P_{in}} \tag{1}$$

Energy losses in the system can be classified by their type, i.e. into liquid stream losses (ΔQ_{st}) and into hydromechanical losses (Δp_{st}). The total losses in the system are the sum of losses which occur on the individual elements of the system.

$$\Delta Q_{st} = \sum Q_I, Q_{II} \dots Q_n \qquad (2)$$

where:

ΔQ_{st} – total liquid stream losses in the system ($\frac{m^3}{s}$),
Q_I, Q_{II}, Q_n – liquid stream losses on the 1st, 2nd and n^{th} element of the system ($\frac{m^3}{s}$),.

$$\Delta p_{st} = \sum p_I, p_{II} \dots p_n \qquad (3)$$

Δp_{st} – total pressure losses in the system, (Pa),
p_I, p_{II}, p_n – pressure losses on the 1st, 2nd, and n^{th} element of the system, (Pa).

The amount of losses on the individual elements of the system is frequently described using the efficiency parameter, both for liquid stream losses, which are described in terms of volumetric efficiency (η_v) and for pressure losses, which are described in terms of hydromechanical efficiency (η_{hm}). In this case, the total efficiency of the system is the product of the efficiencies of its individual elements.

$$\eta = \eta_{v(I)} \cdot \eta_{hm(I)} \cdot \eta_{v(II)} \cdot \eta_{hm(II)} \cdot \dots \cdot \eta_{v(n)} \cdot \eta_{hm(n)} \qquad (4)$$

where:

$\eta_{v(I)}$, $\eta_{v(II)}$, $\eta_{v(n)}$ – volumetric efficiency of the2nd and n^{th} element of the system, (-),
$\eta_{hm(I)}$, $\eta_{hm(II)}$, $\eta_{hm(n)}$ – hydromechanical efficiency of the2nd and n^{th} element of the system, (-).

The above values are not constant and depend mainly on the flow rate of the liquid, viscosity and pressure. As the viscosity of hydraulic oils is strongly related to temperature, it can be assumed that the amount of losses in the system is influenced by the above two factors, i.e. the flow rate and the temperature of the liquid. The above equations also demonstrate that from the perspective of energy efficiency limiting the number of elements passed by the liquid stream is advantageous and that all of the elements should have high volumetric and hydromechanical efficiency. This condition is particularly difficult to be met in the case of systems with different and scattered hydraulic control, in which individual functions of the system are effected with the use of independent valves or sometimes even valve systems. Therefore, it seems practical to base the research direction on first defining the original function of the system and only subsequently on performing the design process in such a manner that all the functions of the system are performed with the use of the smallest possible number of elements provided with appropriate characteristics. In this analysis, the valve blocks with an adapted structure are treated as one element, as their functions are usually analogous to those of the spool valve blocks.

2.2 Guidelines on system analysis

The process of reducing the number of system elements is limited mainly by changes of the dynamic responses after modifications in the system and by the resulting negative phenomena such as hydraulic resonance, system instability caused by switching the valves, system self-excitation caused by fast signal changes on the control circuits and also changing stiffness of the operating elements [Siwulski T. 2020]. All of the above aspects have a direct influence on how systems function, although not all of them can be predicted in the state of the art. The adverse dynamic phenomena can be limited by avoiding serial connections of elements controlled by pressure signals from the system and by using hydraulically independent electric control means in which the function of the signal in time can be manipulated. It is also advantageous to use cartridge valves, whose moving elements have small masses and relatively short response times. The basic guidelines can be thus presented in the following points:

- the system architecture must ensure that all of the specified functions, or all of the functions prior to system modifications, are enabled,
- the number of control elements, understood as a number of valves with the flow, should be maximally reduced,
- the elements should be advantageously controlled by electric signals,
- hydraulic control signals generated in the system should be avoided,
- an increase in the actuator stiffness, followed by increased maximum pressure values in the system when switching occurs, should be allowed for,
- the potential for encasing the elements of the system must be dictated by the mechanical design and by external factors.

A broad approach to designing the architecture of hydraulic systems may be viewed as one of the development directions in this field. Importantly, due to the multidimensional character of the analyses, some of the requirements need to be prioritized. Increasingly often, the greatest priority is given to the efficiency parameter.

3 MATHEMATICAL MODEL OF THE SYSTEM FOR EFFICIENCY ANALYSIS

The mathematical model of the system was developed in order to identify the influence of the individual system elements on the measured efficiency levels. The model was implemented in MATLAB, in the SIMULINK environment. The software allowed a detailed analysis of the influence exerted by the individual parameters of the system elements and by the variables which describe the working characteristics of the hydraulic system. The results of this analysis are provided further in the paper. The individual variables and the characteristic parameters of the model were implemented on the basis of both the data provided to the authors, which were partly described in the previous section, and the general knowledge on hydrostatic drive mechanisms. What follows are detailed descriptions of mathematical models representing the phenomena and elements involved in the system, which were used to develop the final model of the system.

3.1 Working liquid model

A broad model analysis of energy losses was enabled by introducing liquid temperature as a variable in the model. The liquid density parameter was assumed to be constant with respect to temperature, and the viscosity parameter as a function of temperature is described with the Herschel equation [1-3], which, after transformation, is expressed as follows:

$$\mu = \mu_0 \left(\frac{T_0}{T}\right)^k \quad (5)$$

where:

μ – dynamic viscosity of liquid at a temperature T,

μ_0 – dynamic viscosity of liquid at a temperature T_0,

k – slope.

In order to determine the slope of a straight line k, the viscosity of a liquid must be identified in two different temperatures. In practice, viscosity is typically determined at 40 and 100 °C. The slope is described with the following relationship:

$$k = \frac{\ln \mu_1 - \ln \mu_2}{\ln T_2 - \ln T_1} \quad (6)$$

where:

μ_1 – dynamic viscosity of liquid at a temperature T_1,

μ_2 – dynamic viscosity of liquid at a temperature T_2,

By allowing for the constant value of the viscosity parameter and by defining the relationship of the kinematic viscosity of the liquid as:

$$\nu = \frac{\mu}{\rho} \quad (7)$$

where:

ν – kinematic viscosity,

ρ – density of the liquid.

the following final relationship was obtained, which describes the change of kinematic viscosity of liquid in the function of temperature:

$$\nu = \nu_{40} \left(\frac{T_{40}}{T}\right)^k \quad (8)$$

where:

ν – kinematic viscosity of liquid at a temperature T [cSt],

ν_{40} – kinematic viscosity of liquid at a temperature of 40°C [cSt],

T_{40} – reference temperature of 40 [°C].

In this relationship, the slope is described with the following relationship:

$$k = \frac{\ln \nu_{40} - \ln \nu_{100}}{\ln T_{100} - \ln T_{40}} \quad (9)$$

where:

ν_{100} – kinematic viscosity of liquid at a temperature of 100°C [cSt],

T_{100} – reference temperature of 100 [°C].

Moreover, the temperature of the oil was assumed to be identical in the entire system, and therefore each of the elements was assumed to work with a liquid of identical viscosity parameter at a certain time.

3.2 Hydraulic lines model

When developing the model of flows in hydraulic lines, the liquid stream movement was assumed to have both a laminar and a turbulent character. In order to identify this character, the Reynolds number (Re) was first determined from the following relationship:

$$Re = \frac{v \cdot d_w}{\nu} \quad (10)$$

where:
d_w – hydraulic line internal diameter,
v – mean liquid velocity in the hydraulic line.

Mean liquid velocity in the hydraulic line was determined from the following relationship:

$$v = \frac{Q}{\pi \cdot \frac{d_w^2}{4}} \quad (11)$$

where:
Q – flow rate in the hydraulic line.

In order to identify the character of the flow in the line, the limit value of the Reynolds number was defined at 2300. In the case when this value was exceeded, the flow in the line was assumed to be turbulent, and in the case when the values were below the limit, the flow was assumed to be laminar.
The losses in the line, described as pressure differences, were described with the following relationship:

$$\Delta p = \lambda \frac{l}{d_w} \frac{\rho}{2} v^2 \quad (12)$$

where:
λ – linear loss coefficient,
l – line length.

The above relationship may be used to define pressure losses in both the turbulent and the laminar flow, the difference being that depending on the flow character, the linear loss coefficient λ is described differently. In the case of the laminar flow, the value of the coefficient was assumed to be:

$$\lambda = \frac{75}{Re} \quad (13)$$

Importantly, the value of 75 assumed in the numerator describes the hydraulic lines quite well, allowing for the heat exchange with the environment and for minor turbulences in the connections.
In the case of the turbulent flow, the linear loss coefficient was determined from the Blasius equation:

$$\lambda = \frac{0{,}316}{\sqrt[4]{Re}} \quad (14)$$

The model was provided with a module which allows and indicates a change of the flow type in individual lines.

3.3 Valve model

The valves are provided with characteristics which describe pressure decrease in the function of flow rate, defined for a certain directly given parameter of the kinematic viscosity of liquid. As individual valves are described by flow characteristics defined for one value of kinematic viscosity (*ν*) and these values may be different for each of the valves, the influence of liquid viscosity on the pressure difference values for a constant flow was assumed in the form of the following relationship:

$$\Delta p_Q = \Delta p_{cQ} \cdot \frac{\nu}{\nu_{cQ}} \quad (15)$$

where:
Δp_Q – pressure drop on the element represented as a function of liquid kinematic viscosity (ν) and defined for a constant flow rate,
Δp_{cQ} – pressure drop on the element characterized for a constant liquid kinematic viscosity (ν_{cQ}) and at a constant flow rate,
ν_{cQ} – liquid kinematic viscosity which served as reference for the characteristics offered by the manufacturer.

Moreover, the model was provided with a module allowing for the calculations of pressure loss values as a function of flow in the case when the value of the flow rate in the element exceeds the nominal value. For this purpose, it was assumed that at the maximum flow rate predicted and allowed by the manufacturer, the flow in the valve is fully turbulent. This assumption can be described with the following equation:

$$\Delta p_{Q\_t} = \zeta \frac{\rho}{2} \frac{Q^2}{A^2} \quad (16)$$

where:
Δp_{Q_t} – pressure loss in the case of turbulent flow through the valve,
ζ – flux loss coefficient,
Q – flow rate,
A – area of flow cross-section.

If assumed that that the values of the flux loss coefficient (ζ), flow cross-sectional area (A) and viscosity (ρ) are constant, the last point of the characteristic curve can be indicated as the basis for calculating pressure losses at higher flow rates. With the area of the flow cross-section assumed as the basis for the comparison, the following relationship was possible:

$$A^2 = \zeta \frac{\rho}{2} \frac{Q_I^2}{\Delta p_{Q\_t\_I}} = \zeta \frac{\rho}{2} \frac{Q_{II}^2}{\Delta p_{Q\_t\_II}} \quad (17)$$

where:
Q_I – maximum flow rate specified by the manufacturer,
$\Delta p_{Q\_t\_I}$ – pressure loss defined in the specification for the maximum flow rate (Q_I),
Q_{II} – flow rate exceeding the values specified by the manufacturer,
$\Delta p_{Q\_t\_II}$ – pressure loss defined for the flow rate exceeding the values specified by the manufacturer (Q_{II}).

After reductions and transformations, the above relationship can be expressed as:

$$\Delta p_{Q\_t\_II} = \frac{Q_{II}^2}{Q_I^2} \Delta p_{Q\_t\_I} \quad (18)$$

The above relationship was implemented in the numerical model and served to define pressure losses for flow rates exceeding the maximum values specified for a particular valve or filter type, while also allowing for the above relationships which describe the influence of viscosity on the level of pressure losses. Moreover, the software directly identifies exceeded values during calculations.

3.4 Simplified models of the hydraulic cylinder, the pump and the auxiliary elements

This paper presents the influence of the architecture of the system on its efficiency and therefore the models of both the actuator in the form of a hydraulic cylinder and the pump were significantly simplified, while the auxiliary elements, such as filters and radiators, were completely omitted. Therefore, the volumetric efficiency of the actuators (μvs) was assumed to be equal to unity, while the hydromechanical efficiency (μhms) – to be constant and equal to 0.9. The operation of the cylinder was analyzed on the basis of the ratio of active surface areas on both sides of the piston (ϕ), which is the basic geometric parameter and which is calculated from the following equation:

$$\varphi = \frac{D^2}{D^2 - d^2} \quad (19)$$

where:
D – diameter of cylinder piston,
d – diameter of cylinder piston rod.

In the case of cylinder extension, when the liquid is supplied to the piston chamber, and with the assumed absence of leaks, the outlet flow rate is described by the following relationship:

$$Q_{out\_I} = \frac{Q_{z\_I}}{\varphi} \quad (20)$$

where:
Q_{out_I} – flow rate of liquid removed from the extending cylinder,
Q_{in_I} – flow rate of liquid in the line supplying the extending cylinder.

The value of the pressure in the piston chamber is expressed with the following equation:

$$p_{p\_I} = \frac{4F}{\mu_{hms} \pi D^2} + \frac{p_{r\_I}}{\mu_{hms} \varphi} \quad (21)$$

where:
p_{p_I} – pressure on the piston side of the extending cylinder (supply pressure),
p_{r_I} – pressure on the piston rod side of the extending cylinder,
F – force acting on the extending cylinder.

In the case of retraction, the analogical relationships for the cylinder are as follows:

$$Q_{out\_II} = \varphi \cdot Q_{in\_II} \quad (22)$$

where:
Q_{out_II} – flow rate of liquid removed from the retracting cylinder,
Q_{in} – flow rate of liquid in the line supplying the retracting cylinder.

The value of the pressure in the piston rod chamber is expressed with the following equation:

$$p_{p\_II} = -\frac{4F}{\mu_{hms} \pi (D^2 - d^2)} + \frac{\varphi \cdot p_r}{\mu_{hms}} \quad (23)$$

where:

p_{p_II} – pressure on the piston rod side of the retracting cylinder (supply pressure),
p_{r_II} – pressure on the piston side of the retracting cylinder.

In the discussed case, an extended pump model was not used, and instead it was replaced with a source of constant flow rate (Q_p) independent of the load and viscosity.

4 MODEL OF HYDRAULIC SYSTEM FOR ROCK BREAKER TYPE URB ZS-3

The above mathematical model was used to analyze the energy-consumption of a drive system for cylinders powering the work system of the URB ZS-3 rock breaker manufactured by KGHM ZANAM S.A. and used in underground copper ore mines (Figure 1).

Figure 1 Rock breaker URB ZS-3

In the first step, the system components were changed in order to better adjust the parameters to each other and to the hydraulic quantities present in the system. As a result, the initial specification of the system was obtained for the analysis of the influence of the system architecture on its efficiency. This article does not describe the above process of reselection of hydraulic components. In the second step, a simplification of the modified hydraulic system with pressure-controlled valves was made. As the analysis focused on the control system, the interchangeable elements, such as oil cooler or filters, were omitted. The following important system functions and limitations were identified:

- the possibility to supply all actuators from one pump,
- the necessity to ensure a free movement of the cylinder under load in the case of system decompression,
- the total lengths of the hydraulic lines should not change, as the modification of the cylinder positions and of the pump system were not considered,
- the valves should have minimized geometric dimensions,
- the system should have good dynamic properties which would allow a precise identification of the position of the work element.

The above conditions clearly show that the efficiency of the system can be improved primarily by changing its architecture, and not by just replacing its individual elements with elements having lower flow resistance values, i.e. with geometrically larger elements. Moreover, the cylinder efficiency was assumed to be equal to unity. As a result, all of the defined parameters and the output efficiency were related only to the modified drive system and therefore it can be viewed as equivalent to the efficiency of the hydraulic drive system from the source to the actuator. In order to directly identify the efficiency, the output power from the system was assumed to be the mechanical power on the cylinder, which is equal to:

$$P_{out} = v \cdot F \tag{24}$$

where:
v – velocity of the piston rod [m/s],
F – force acting on the cylinder [N].

The input power, on the other hand, was defined as hydraulic power on the source, in the following form:

$$P_{in} = p \cdot Q \tag{25}$$

where:
p – liquid pressure on the source [Pa],
Q – flow rate on the source [$\frac{m^3}{s}$].

The analyzed systems comprised the same cylinder, with a nominal extension force equal to 250 kN and a nominal retraction force equal to 185 kN. The determination of the efficiency of the systems was carried out on the basis of the above values in order to determine the maximum efficiencies, the value of which depends on the adopted values of the output power. In the discussed analyses, the cylinder efficiency was equal to unity.

The original system comprised three cylinders having identical parameters and driven by analogical systems, and therefore the analysis was limited to the drive system of only one cylinder. Figure 2 shows a reduced original schematic diagram of the system , indicated with No. 1.

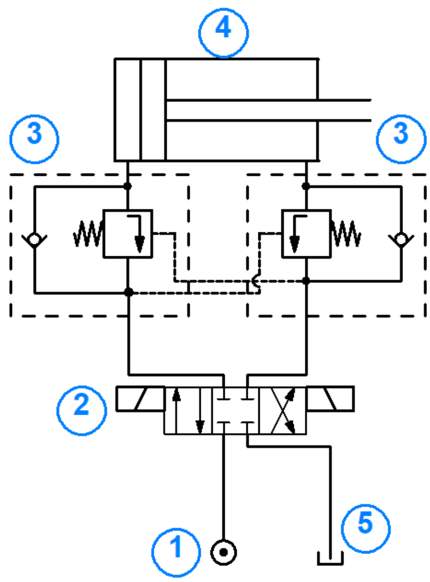

Figure 2 Reduced original hydraulic scheme which served as the basis for the current model – model 1

The working liquid used in the system is a refined mineral oil HYDROL L - HV 68. The oil has the following basic parameters:

- density: 850 ÷ 900 kg/m^3 – the value used in the calculations was 875 kg/m^3,
- kinematic viscosity at a temperature of 40°C equal to 66.27 cSt,
- kinematic viscosity at a temperature of 100°C equal to 10.56 cSt,

- ignition temperature: 226°C,
- flow temperature: -30°C.

The analyses were performed for a number of structures of the system. However, the influence of modifications to the system architecture on its energy-consumption, as well as on other aspects of the system, will be demonstrated on two illustrative structures. The first is a classic structure, in which pilot operated check valves were implemented in the place of standard pilot assisted load control valves (Figure 3). This solution is designated with number 2. The actual design did not incorporate pilot operated check valves controlled by pressure signals, because they did not allow the tool at the end of the work system to be positioned with an adequate repeatability and accuracy.

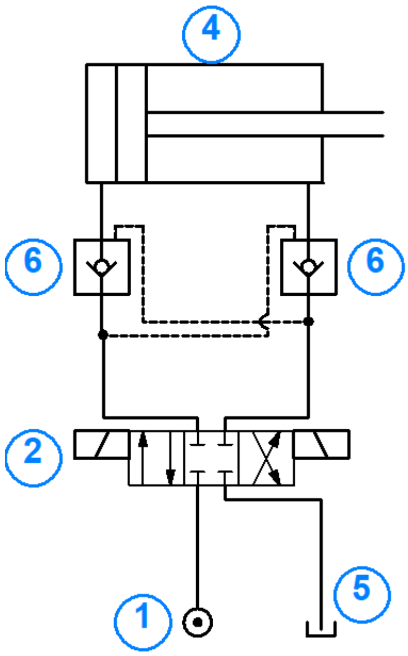

Figure 3 Modified hydraulic scheme incorporating pilot operated check valves – model No. 2

The third system, proposed for implementation in the actual design and designated with No. 3, incorporates hermetic pilot operated 2/2 way valves controlled by electric signals. In this system, the control valve switching the power supply is an additional element used in order to protect the system against the loss of tightness and to ensure an option to disconnect the hydraulic drive. In order to reliably identify the influence of structural changes in the system on its efficiency, the analysis was based on the parameters of a valve identical to the valve in the first two systems, although it should be importantly installed possibly close to the hydraulic drive system and may be replaced with a 2/2 way valve. Also, the number of external control signals in the third system is increased, leading to a complete elimination of hydraulic signals inside the system. Moreover, the range of functions available during operation is broadened by:

- the ability to implement fast movement,
- the ability to simultaneously connect the two cylinder chambers with the reservoir and thus to move freely,
- the ability to wash the lines supplying liquid to the cylinder by enabling the liquid to flow from the pump through the valves to the reservoir.

A significantly increased control potential is the result of using the new structure and of basing practically all control functions on electric signals, whose behavior can be easily modified. It should be mentioned that in this system a certain disadvantage may be the necessity to provide additional four control signals to the valve block mounted on the cylinder.

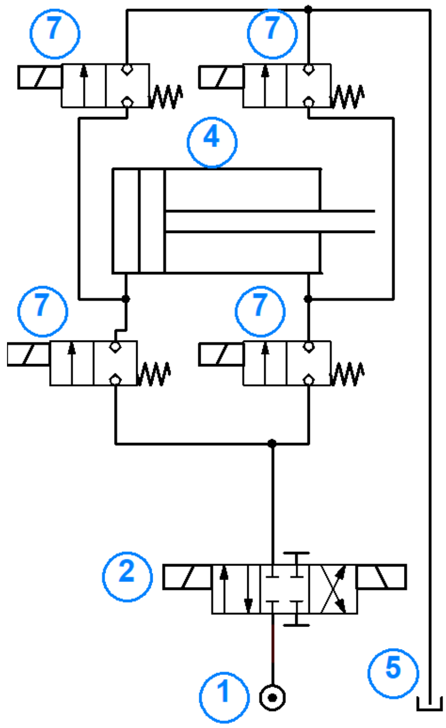

Figure 4 Modified hydraulic scheme of the system following the new architecture – model No. 3

Table 1 presents the description of basic elements used in all of the discussed systems. It should be noted that the objective was to obtain the best possible similarity of the elements in terms of their operating parameters, with a particular emphasis on the maximum flow rate value. The valves at the cylinder are cartridge type, which facilitates their installation in the valve block. Moreover, as they are installed in the block arranged on the cylinder body, the analysis ignored the influence of internal flow resistances in the valve block and in the cylinder, based on an assumption that they are relatively small and also identical for all of the systems. Therefore, the models include specified hydraulic lines in the form of conduits having parameters as provided by the manufacturer of the device (Table 2).

No.	Element	Model	Rated flow [dm^3/min]
1	Hydraulic drive	-	80
2	Solenoid Operated Spool Valve	Wandfluh WD M F A10	160
3	Standard Pilot Assisted Load Control Valve	Parker E2B060ZNMK2	120

4	Hydraulic Cylinder	SHJ2F-140/70	340
5	Reservoir	-	-
6	Pilot operated check valve	Wandfluh RNXPM33	150
7	Pilot operated 2/2-way valve	Wandfluh SVSPM33	150

Table 1 Element specification

No.	Symbols of the connected elements	Internal line diameter [mm]	Line length [mm]
1	1 - 2	20	5.000
2	2 - 3 (1) 2 – 6 (2) 2 – 7 (3)	16	5.390
3	3 – 2 (1) 6 – 2 (3)	16	5.390
4	2 – 5 (1, 2)	25	8.190
5	7 – 5 (3)	20	13.580

Table 2 Specification of hydraulic lines

The length of the return line in hydraulic system 3 was identified from the parameters of the return lines originally installed in systems 2 and 3. It was assumed that all three systems have a similar structure, based on a directional valve at the supply level (No. 2) and a system of valves built into one valve block mounted on the cylinder.

5 TEST RESULTS

The mathematical model was implemented in the Matlab environment, and the presented results refer to a stabilized flow and load. All of the analyses of the energy losses in individual systems were performed on the basis of an assumption that the liquid temperature was constant at 55˚C. First, the analysis focused on the systems driven by a constant flow rate of 80 dm^3/min. This flow rate corresponded to the actual flow rate in the hydraulic system of the URB ZS-3 rock breaker. The analyses were performed for two cylinder movement directions. The results are shown in Tables 3 and 4.

<table>
<tr><th>Energy losses [kW]</th><th>System 1</th><th>System 2</th><th>System 3</th></tr>
<tr><td>Line 1-2 supply</td><td colspan="3">0.118</td></tr>
<tr><td>Valve 2 supply</td><td colspan="3">0.529</td></tr>
<tr><td>Line 2 – 3, 6, 7 supply</td><td colspan="3">0.368</td></tr>
<tr><td>Valve 3, 6, 7 supply</td><td>0.977</td><td>0.797</td><td>1.024</td></tr>
<tr><td>Valve 3, 6, 7 return</td><td>0.438</td><td>0.194</td><td>0.551</td></tr>
<tr><td>Line 3, 6, 7 – 2 return</td><td colspan="2">0.120</td><td rowspan="3">0.124</td></tr>
<tr><td>Valve 2 return</td><td colspan="2">0.292</td></tr>
<tr><td>Line 2-5 return</td><td colspan="2">0.031</td></tr>
<tr><td>Sum total</td><td>2.873</td><td>2.449</td><td>2.714</td></tr>
<tr><td>Difference relative to system 1</td><td>-</td><td>-0.424</td><td>-0.159</td></tr>
</table>

Table 3 Energy losses in the systems for the flow rate on the source equal to 80 dm^3/min and for oil temperature equal to 55 ˚C during cylinder extension

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<tr><td>Valve 3, 6, 7 return</td><td>2.327</td><td>0.91</td><td>1.962</td></tr>
<tr><td>Line 3, 6, 7 – 2 return</td><td colspan="2">0.812</td><td rowspan="3">0.709</td></tr>
<tr><td>Valve 2 return</td><td colspan="2">1.646</td></tr>
<tr><td>Line 2-5 return</td><td colspan="2">0.148</td></tr>
<tr><td>Sum total</td><td>6.925</td><td>5.328</td><td>4.710</td></tr>
<tr><td>Difference relative to system 1</td><td>-</td><td>-1.597</td><td>--2.215</td></tr>
</table>

Table 4 Energy losses in the systems for the flow rate on the source equal to 80 dm^3/min and for oil temperature equal to 55 ˚C during cylinder retraction

The efficiency of the drive system was also defined on the basis of the rated cylinder load equal to 250 kN for the compressive force and to 185 kN for the tensile force. The results are shown in Table 5.

Efficiency of the drive system	System 1	System 2	System 3
Cylinder extension	0.883	0.898	0.889

Cylinder retraction	0.755	0.800	0.819

Table 5 Obtained efficiencies of the drive systems relative to the rated loads for the flow rate on the source equal to 80 dm³/min and for the oil temperature equal to 55˚C

The obtained results allow a conclusion that the main energy losses occur in the valves installed at the cylinder. These valves are an element of the protection system and therefore their implementation is obligatory. However, in the analyzed case the change of the system architecture (system 3) always reduced the energy-consumption of the system relative to the original system (system 1) while also improving efficiency relative to system 2 in the case of cylinder retraction. During cylinder extension, systems 2 and 3 showed very similar energy consumption: the difference was only 0.265 kW in favor of system 2, with a gain of 0.618 kW during the cylinder retraction. Importantly, when compared to the original system 1, the energy loss in system 3 during cylinder retraction was reduced by more than 2 kW.

As the parameters of individual system elements allowed the flow rate on the source to be increased to 90 dm³/min, which corresponded to the flow rate on the return line during cylinder retraction equal to 120 dm³/min, in the next step the simulations were performed with an increased flow rate. The results of these simulations are presented in Tables 6, 7 and 8.

Energy losses [kW]	System 1	System 2	System 3
Line 1-2 supply	0.164		
Valve 2 supply	0.753		
Line 2 – 3, 6, 7 supply	0.509		
Valve 3, 6, 7 supply	1.403	0.973	1.320
Valve 3, 6, 7 return	0.617	0.253	0.710
Line 3, 6, 7 – 2 return	0.231		0.157
Valve 2 return	0.417		
Line 2-5 return	0.039		
Sum total	**4.133**	**3.339**	**3.613**
Difference relative to system 1	**-**	**-0.794**	**-0.520**

Table 6 Energy losses in the systems for the flow rate on the source equal to 90 dm³/min and for oil temperature equal to 55 ˚C during cylinder extension

Energy losses [kW]	System 1	System 2	System 3
Line 1-2 supply	0.164		
Valve 2 supply	0.753		
Line 2 – 3, 6, 7 supply	0.509		
Valve 3, 6, 7 supply	1.403	0.973	1.320
Valve 3, 6, 7 return	3.303	1.297	2.656
Line 3, 6, 7 – 2 return	1.122		0.98
Valve 2 return	2.380		
Line 2-5 return	0.205		
Sum total	**9.839**	**7.403**	**6.382**
Difference relative to system 1	**-**	**-2.436**	**-3.457**

Table 7 Energy losses in the systems for the flow rate on the source equal to 90 dm³/min and for oil temperature equal to 55 ˚C during cylinder retraction

Efficiency of the drive system	System 1	System 2	System 3
Cylinder extension	0.855	0.880	0.871
Cylinder retraction	0.710	0.765	0.790

Table 8 Obtained efficiencies of the drive systems relative to the rated loads for the flow rate on the source equal to 90 dm³/min and for the oil temperature equal to 55˚C

The results for the flow rate equal to 90 dm³/min confirmed advantageous differences in energy consumption between the original system 1 and systems 2 and 3. However, a relatively limited increase in the flow rate resulted in the energy loss increasing by more than ten percent. This increased energy loss was in both cases observed to be the least significant in system 3.

The energy losses and the system efficiency during operation at different flow rates were more precisely investigated by simulating the energy losses in the case of supplying the systems with the highest permissible flow rate. An assumption was made that the simulation will only include the extension of the cylinder supplied with a flow rate equal to 120 dm³/min, which was the limit value in the implemented valve systems. These tests were performed only for cylinder extension, as supplying such a flow rate to the piston rod side would generate a flow rate of 160 dm³/min on the return line and this would significantly exceed the rated values for the system elements. The results are shown in Tables 9 and 10.

Energy losses [kW]	System 1	System 2	System 3
Line 1-2 supply	0.361		
Valve 2 supply	1.867		
Line 2 – 3, 6, 7 supply	1.122		
Valve 3, 6, 7 supply	3.303	1.605	2.656
Valve 3, 6, 7 return	1.403	0.553	1.32
Line 3, 6, 7 – 2 return	0.509		0.444
Valve 2 return	0.971		
Line 2-5 return	0.069		
Sum total	**9.605**	**7.057**	**7.770**

Difference relative to system 1	-	-2.548	-1.835

Table 9 Energy losses in the systems for the flow rate on the source equal to 120 dm³/min and for oil temperature equal to 55 ˚C during cylinder extension

Efficiency of the drive system	System 1	System 2	System 3
Cylinder extension	0.772	0.822	0.807

Table 10 Obtained efficiencies of the drive systems relative to the rated loads for flow rate on the source equal to 120 dm³/min and for oil temperature equal to 55 ˚C

The obtained values of energy losses were compared to the results obtained for cylinder retraction at the supply flow rate of 90 dm³/min. This fact serves as a confirmation of how important, from the perspective of an energy-efficiency analysis, it is to allow for the changes of liquid flow in cylinders and for the flow resistances on the return lines. It is particularly important in the case of using valves which prevent the cylinder from moving independently under load and when the flow in the system needs to be throttled in order to increase pressure values in the control lines.

In order to facilitate the interpretation of the results, Figures 5 and 6 present a comparison of the efficiencies of the drive systems, calculated in accordance with equation 1 and with allowance for the assumptions shown in section 4.

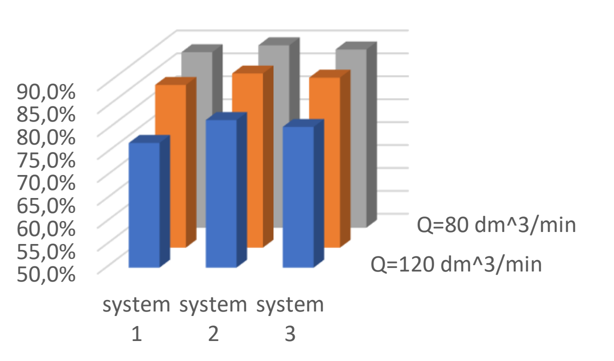

Figure 5 System efficiencies obtained during cylinder extension

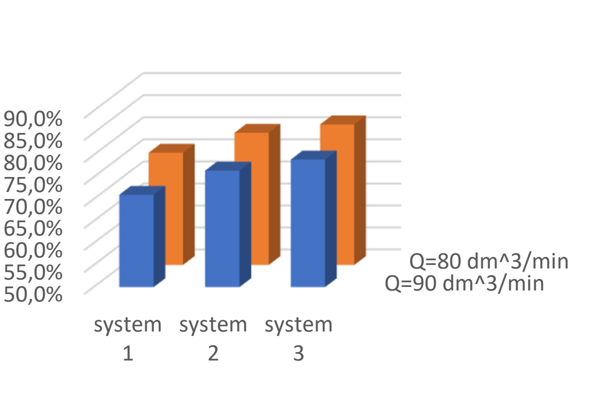

Figure 6 System efficiencies obtained during cylinder retraction

The comparison of the obtained efficiency values clearly indicates that the modification of the system architecture proposed in system 3 not only broadens the scope of potential system functions and effects advantageous changes with respect to the control systems, but also may increase efficiency.

6 CONCLUSIONS

The paper presents an illustrative successful process of modifying a hydraulic system with an aim to increase its efficiency. The implementation of the mathematical model of the system allowed the energy losses to be defined and compared in the individual analyzed systems. The original model for further analyses was developed on the basis of the actual hydraulic drive system of the work system in the URB ZS-3 rock breaker manufactured by KGHM ZANAM S.A. operated in underground copper ore mines. The results clearly show that the energy losses can be significantly limited: in the analyzed case by as much as 3.457 kW for the retraction of a cylinder supplied by a flow rate of 90 cm³/min in the system with modified architecture. Moreover, as compared to the original system, the system following the new architecture was observed to have improved efficiency within the entire investigated range and to have an efficiency greater than the efficiency of the simplified system in the case of cylinder retraction. This phenomenon is likely due to increased flow rate on the return line. The results here presented also clearly show that a modification of hydraulic system architecture may limit its energy-consumption without the need to limit its functions, and even may actually broaden the range of the available functions. Moreover, the elimination of the relatively unstable system elements and of the internal pressure signal lines is expected to result in improved parameters of the system movement and in the elimination of adverse dynamic phenomena in the system. The authors believe that the presented direction for the development of hydraulic lines is directly related to the current industrial and social requirements and therefore should be pursued further.

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