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Analysis of the Energy Efficiency Improvement in a Load-Sensing Hydraulic System Built on the ISO Plate

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Abstract: The article presents a proposal to reduce energy consumption in a hydraulic system with a single pump and multiple receivers. The proposed Load-Sensing Basic (LSB) solution consists of expanding a typical hydraulic system by using additional logic valves and a dedicated differential valve. The modification is aimed at decrease in operating pressure and, thus, reduction in energy consumption. The LSB system is compact as all components are built on a single ISO plate. A detailed mathematical model of the system was formulated, then a simulation model was built and numerical tests were carried out in the Matlab/Simulink environment. The obtained results indicate that the use of the proposed LSB system for the implementation of typical working cycles with three actuators may reduce energy consumption by 4–30%, and under certain conditions even up to 70%.

Keywords: hydraulic system design; reduction of energy consumption; numerical simulation; load-sensing

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1. Introduction

Heavy-duty machines and working equipment often use hydraulic drives consisting of a single fixed-delivery pump which supplies multiple receivers. This solution is economically viable, as the system is relatively simple and constructed of typical low-cost components. Hence, it is commonly used in heavy duty machines and industrial systems. In the case when the pump's delivery is higher than the total demand of all receivers, the excessive flow is discharged through the relief valve to the tank. Appropriate control of the hydraulic elements makes it possible to obtain even complex working movements of receivers, thus automating the operation process. In particular, solenoid-operated proportional valves offer great possibilities for fast and precise control. The main disadvantage of throttling systems supplied with a single pump of a constant capacity, when the pump's output is not fully utilized, is significant energy loss. The loss results from the fact that the excessive fluid flow rate is drained back to the tank at the pressure set on the relief valve. This wasted energy is converted into heat, which in turn may require the use of additional oil coolers.

The issue of reducing energy losses in hydraulic systems is taken up in scientific works. The research covers various solutions, including structural modifications to the pump, the use of control valves, a converter or a compensator, the storage of excess energy with accumulators, etc. A load-sensing design for a pump in the electro-hydraulic actuator system (EHA) was designed by Chao et al. [1] in order to reduce energy waste. The solution consisted in the use of a specially designed spool valve in order to decrease the volumetric displacement of the pump automatically as the load pressure increases. As a result, the motor torque and heat generation are reduced. Shang et al. [2] proposed a novel integrated Load-Sensing Valve-Controlled Actuator (LSVCA) with high efficiency and low energy consumption, which can reduce the overflow loss by the intermittent operation of the motor and reduce the throttling loss by the variation of the supply pressure. Numerical and experimental studies showed the 1.75-fold throttling efficiency improvement. Analysis of a directional control valve for load-sensing application using CFD numerical modelling was carried out by Bigliardi et al. [3]. Discharge coefficient, efflux angle, flow forces, pressure and velocity distributions in the critical region were estimated based on the simulation results. A wide research on energy saving by means of improvement hydraulic system efficiency was carried out by Gradl and Scheidl. As a result, a hydraulic stepper converter consisting of a hydraulic cylinder piston unit and a fast-switching valve was developed [4] and oscillating mass converter (OMC) with a pure hydraulic control [5] was proposed. Similarly, fast-switching valves were used by Pan et al. [6] in a novel approach for increasing energy efficiency in the fluid power systems, that resulted in an approximately one-seventh reduction in power loss compared to a conventional system based on throttle valves. Another proposal of an energy saving strategy for an electrohydraulic servo valve system was proposed by W. Wang and B. Wang [7]. The strategy, based on the use of load-sensing pump, proportional relief valve together with adaptive backstepping sliding mode control, allowed energy savings of 62% in harmonic tests and up to 90% in multi-step tests. In the field of research on hydraulic systems of heavy-duty machines, solutions for improving the energy efficiency of excavators or manipulators are often presented. Bedotti et al. described several different methods to reduce losses, improve fuel-saving and energy recovery [8] as well as presented energy comparison of system layouts for a hydraulic excavator [9]. The results obtained led to the development of a new hybrid excavator concept with a 15% reduction in fuel consumption. Moreover, minimizing the energy demand of a heavy-load mobile manipulator using a novel electro-hydraulic drive system was the subject of research by Ding et al. [10], while Cheng et al. [11] developed a load-sensing system for an excavator based on two switching valves for controlling flow and pressure separately. A proposal to further improve the efficiency of a typical load-sensing system by reducing inherent pressure losses (SIPL) was formulated by Siebert et al. [12]. The carried out simulations showed that pressure loss may be reduced by up to 44%.

Research on these types of systems is usually carried out with the use of numerical simulation. The principles and methods of hydraulic equipment modelling were summarized in the book by Novak et al. [13]. This methodology was then successfully applied in a number of studies, including research on an adjustable flow control valve by Kuehnlein et al. [14], electro-hydraulic proportional valve by Rybarczyk et al. [15], excavator hydraulic system by Casoli et al. [16] or proportional flow control valve by Lisowski et al. [17]. Moreover, Naseradinmousavi and Nataraj [18] described a non-linear modelling method of solenoid-driven butterfly valves, while Du and Nydal [19] formulated numerical schemes for transient flow in one dimension. A comparison of various numerical modelling approaches related to water hydraulic systems was presented by Antelmi et al. in [20].

This article proposes a method of reducing energy consumption in a Load-Sensing Basic (LSB) system, adapted for installation on a standard ISO 4401 plate. Typical components designed for mounting on the plate were used, including proportional and logic valves and a differential valve. The introduction of an additional differential valve resulted in a pressure drop and hence some energy loss. However, taking into account the global energy demand of the system, it has been significantly reduced in typical practical applications. Using the Matlab/Simulink environment, model of the LSB system with three actuators was built. The carried out simulation tests took into account the implementation of different work cycles by actuators under various loads. The energy demands for the base system and the LSB system were determined and compared.

2. Working Principle of the Analysed System

The subject of the analysis is a hydraulic system which is shown in Figure 1. The valve block has been mounted on a standard ISO 4401 plate with a nominal size of DN 6. There are three lines with hydraulic actuators in the system. The lines are powered from a fixed delivery pump (1) of nominal flow rate $Q_{nom} = 80~\mathrm{dm^3\,min^{-1}}$, with maximum outputs to the receivers $Q_{max,i} = 40\ \mathrm{dm}^3\,\mathrm{min}^{-1}$ at a maximum pressure of $p_{max} = 32$ MPa. Relief valve (5) provides protection against overpressure. The flow to individual receivers (3) is controlled by means of three proportional valves (2), and the tightness is ensured by hydraulic locks (4). The presented system is simple, reliable in operation and hence often used in practice. However, in the event that not all actuators are running simultaneously or the total flow requirement of the actuators is less than the pump delivery, significant energy losses may occur in the system. The loss is related to keeping the high supply line pressure, which is only limited by the operation of the relief valve.

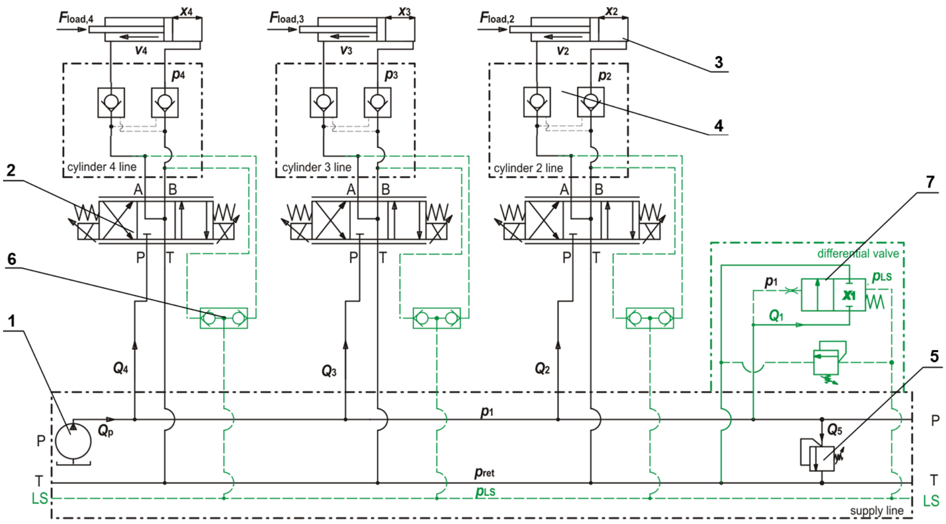

Figure 1. Scheme of the analysed hydraulic system: 1—pump, 2—proportional control valve, 3—hydraulic cylinder, 4—hydraulic lock, 5—relief valve, 6—logic valve, 7—differential valve; black colour—base system, green colour—proposed LSB modification.

In order to reduce energy losses, it was proposed to modify the system by adding the LSB module, which is shown in green in Figure 1. For economic and practical reasons, the same ISO connection plate and control valves with hydraulic locks were used in the extended system. The LSB module contains three sets of logic valves (6) which receive load signals from individual actuator lines and a differential valve (7) to decrease supply line pressure. All the included elements are typical hydraulic components with known flow characteristics and connection dimensions. This ensures low costs and short expansion time. The LSB control signals are taken directly from the outputs of the proportional control valves in the proposed system. Plate mounted control valves were used in accordance with ISO 4401-03-03 which, in the neutral position, provide an open connection between the actuator lines and the return line. Thus, when all control valves are in neutral position, no pressure appears on the LSB line, so the differential valve opens and the supply line is discharged. Similarly, when the total flow requirement of all actuators is lower than the pump output, and the pressure in the line of the most loaded actuator is lower than that set on the relief valve (5), the required power supplied to the pump is reduced compared to the classic system without LSB. This study uses the results of previous work on flow calculations and determining the characteristics of proportional distributors, e.g., [21,22]. For the special needs of this system, however, a dedicated differential valve adapted to be mounted on an ISO plate with an integrated safety relief valve was developed.

A key element for the correct operation of the LSB system is the differential valve. The view of the created 3D model of the valve is shown in Figure 2, while the cross-section by a plane passing through the spool axis is presented in Figure 3. The flow through the differential valve takes place between the P and T ports. In the considered design, the A and B ports are not used and both should be plugged. However, in industrial applications there is a possibility to mount, for example, one of the control valves directly on top of the differential valve, since all the channels are pass-through.

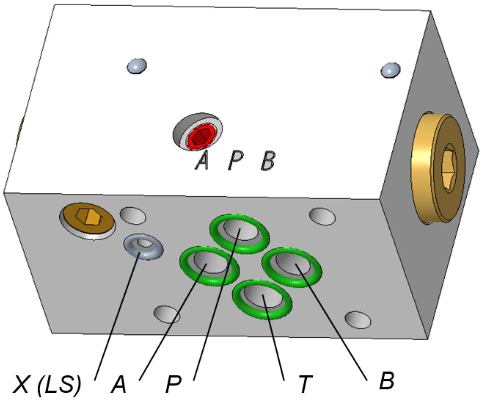

Figure 2. 3D model of the differential valve: 1—valve body, P, A, B, T, LS—connecting ports.

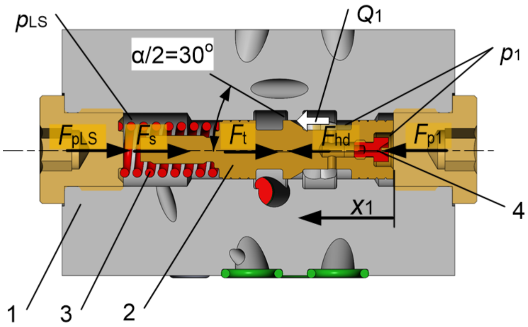

Figure 3. Longitudinal cross-section of the differential valve: 1—valve body, 2—spool, 3—spring, 4—damping nozzle; in, out—connecting ports, $\alpha/2$—half of the cone opening angle, x_1—spool position.

The valve consists of a body (1) in which a spool (2) is placed. The stability of the valve operation is achieved by means of a damping nozzle (4) and a spring (3). All connection ports are pass-through, which enables the potential installation of other valves in the sandwich system. The position of the spool x_1 is determined by the F_{p1} and F_{pLS} forces resulting from the p_1 and p_{LS} pressures, respectively, the F_t viscous friction force, the F_hd flow force and the F_s spring force. In the case when F_{p1} value exceeds the sum of F_{pLS} and F_s, the Q_1 flow is opened and thus the supply line is discharged.

3. Formulating a Mathematical Model

Mathematical model of the studied system was built in accordance with the methodology of modelling hydraulic components, which was introduced by Novak et al. [13]. The created mathematical model consists of a system of differential equations and algebraic formulas. It includes the pump flow rate function, equations of fluid mass conservation formulated for the distinguished volumes, equations of motion of valve spools and hydraulic cylinder pistons, flow equations through throttle gaps based on Bernoulli's law and geometrical relations. The pump was modelled as an ideal source of flow rate. The flow rate Q_p is assumed to be constant at $Q_{nom} = 80\ \mathrm{dm^3\,min^{-1}}$ except for a short start-up time t_s when it rises linearly from 0:

$$Q_p(t) = \begin{cases} Q_{nom} \cdot \frac{t_s}{t} & \text{for } t_s < t \\ Q_{nom} & \text{otherwise} \end{cases} \tag{1}$$

The mass conservation equation formulated for the supply line has the following form:

$$\frac{dp_1(t)}{dt} = \frac{B_f}{V_1} \cdot (Q_p(t) - Q_1(t) - Q_2(t) - Q_3(t) - Q_4(t) - Q_5(t)), \tag{2}$$

where Q_1 refers to the differential valve, Q_5 to the relief valve, and Q_2, Q_3, Q_4 to the lines of actuators, respectively. The supply line volume V_1 and the fluid bulk modulus B_f were assumed to be constant. Analogous equations have been formulated for three lines of actuators based on the flow rates of control valves Q_i as well as positions x_i and velocities v_i of the actuator pistons:

$$\frac{dp_i(t)}{dt} = \frac{B_f}{V_{0,i} + x_i(t) \cdot A_{C,i}} \cdot (Q_i(t) - \frac{dx_i(t)}{dt} \cdot A_{C,i}), \text{ where } i = \{2,3,4\}, \tag{3}$$

where the output flow rates of three control valves Q_i were calculated on the basis of characteristics obtained within the previous research (Figure 21 in [22]). The flow rates were determined based on the current pressure drop $\Delta p_i = p_1 - p_i$ and the percentage value of the control signal I_i:

$$Q_i = f(\Delta p_i, I_i), \text{ where } i = \{2,3,4\}. \tag{4}$$

The p_i pressures generate hydrostatic forces acting on the pistons of hydraulic cylinders. The equations take into account the hydrostatic forces acting on both sides of the piston, the viscous friction force and the load force. The following equations of motion were formulated for the pistons:

$$m_{p,i} \cdot \frac{d^2x_i(t)}{dt^2} = p_i(t) \cdot A_{C,i} - p_{ret} \cdot A_{R,i} - \varphi_i \cdot \frac{dx_i(t)}{dt} - F_{load,i}, \text{ where } i = \{2,3,4\}. \tag{5}$$

Protection against overpressure is provided by a relief valve $RV5$. The valve ensures that the pressure is limited to $p_{1,max} = 30$ MPa. Flow rate through the relief valve gap of the A_5 area is described by the following equation:

$$Q_5(t) = \mu_5 \cdot A_{g,5}(x_5) \cdot \sqrt{\frac{2 \cdot (p_1 - p_{ret})}{\rho}}. \tag{6}$$

where the poppet position x_5 can be determined from the equation of motion:

$$m_5 \cdot \frac{d^2x_5(t)}{dt^2} = (p_1(t) - p_{ret}) \cdot A_5 - \varphi_5 \cdot \frac{dx_5(t)}{dt} - C_5 \cdot (x_5(t) + x_{5,start}). \tag{7}$$

The proposed solution for reducing energy consumption required the installation of a logic valve system that ensured the determination of the maximum pressure occurring in the actuator lines. The pressure signal p_{LS} is the maximum value of all actuator lines which, however, is limited to the relief valve pressure $p_{1,max}$:

$$p_{LS}(t) = MIN(MAX(p_2, p_3, p_4), p_{1,max}). \tag{8}$$

The obtained p_{LS} signal is directed to the differential valve. Flow through the valve depends on the pressure difference acting on both sides of the valve spool: $\Delta p_1 = p_1 - p_{LS}$. The complete spool motion equation, based on Figure 3, has the following form:

$$m_1 \cdot \frac{d^2x_1(t)}{dt^2} = F_{p1} - F_{pLS} - F_t - F_s + F_{hd}. \tag{9}$$

After substituting the expressions which define the individual forces, respectively, F_{p1}, F_{pLS}, F_t, F_s, the equation can be written as follows:

$$m_1 \cdot \frac{d^2x_1(t)}{dt^2} = p_1(t) \cdot A_1 - p_{LS} \cdot A_1 - \varphi_1 \cdot \frac{dx_1(t)}{dt} - C_1 \cdot (x_1(t) + x_{1,start}) + F_{hd}(t), \tag{10}$$

and the flow force was estimated based on previous research by Valdes [23] and Lisowski et al. [21]:

$$F_{hd}(t) = Q_1(t) \cdot \lambda_1(x_1) \cdot \sqrt{2 \cdot \rho \cdot (p_1(t) - p_{ret})}. \tag{11}$$

Having determined the valve spool position $x_1(t)$, the gap area can be calculated on the basis of geometrical relations, taking into account the spool overlap $x_{1,ovr} = 0.5$ mm and the opening angle $\alpha_1 = 60^\circ$:

$$A_1(x_1) = \begin{cases} 0 & \text{for } x_1 < x_{1,ovr} \\ \pi \cdot [d_1 - \frac{(x_1 - x_{1,ovr})}{2} \cdot sin(\alpha_1)] \cdot (x_1 - x_{1,ovr}) \cdot sin(\frac{\alpha_1}{2}) & \text{otherwise.} \end{cases} \tag{12}$$

Finally, the main flow through the differential valve Q_1 can be calculated as:

$$Q_1(t) = \mu_1 \cdot A_{g,1}(x_1) \cdot \sqrt{\frac{2 \cdot (p_1 - p_{ret})}{\rho}}. \tag{13}$$

The presented system of Equations (1)–(13) constitutes mathematical model, which is the basis for the simulation model created in the Matlab-Simulink environment.

4. Structure and Parameters of Simulation Model

A general scheme of the LSB system simulation model in the Simulink environment is presented in Figure 4. The diagram shows the supply line, the load-sensing subsystem and three actuator lines. The *MS*1 manual switch allows the operator to simply change over between the base system and the LSB system. The percentage signal value of the proportional control valves can be set using the *I*2, *I*3 and *I*4 fields, while the load force acting on the actuators should be entered into the *F*2_*cyl*, *F*3_*cyl* and *F*4_*cyl* fields. It should be noticed that previous studies have proved that the practically applicable range of control signals is between 50% and 100%. The blocks representing the individual hydraulic components and lines are connected by signals of pressure, flow, as well as displacement and velocity of the movable parts.

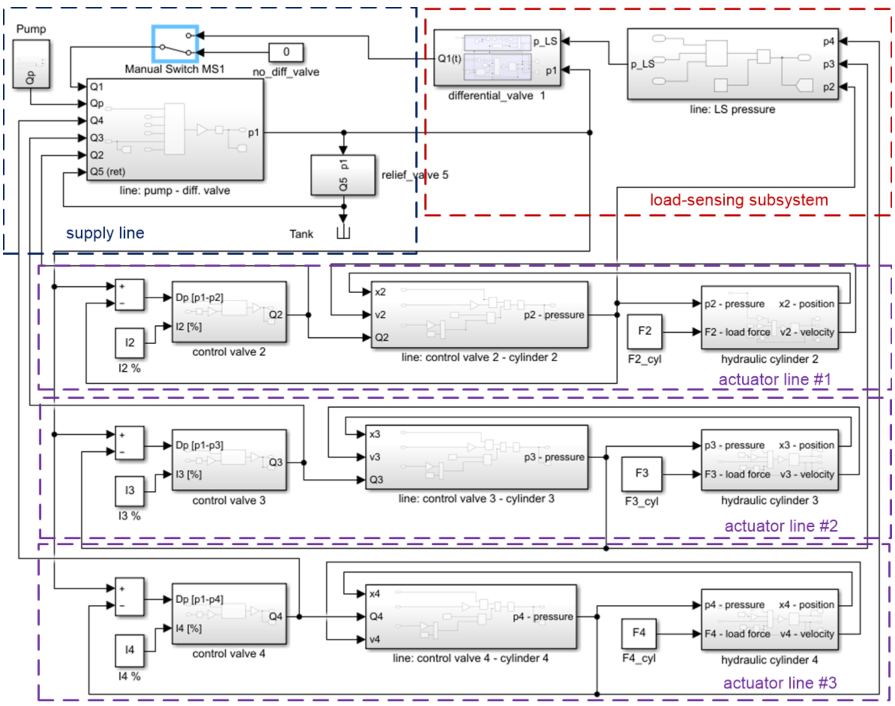

Figure 4. General scheme of the LSB simulation model in Simulink.

As previously mentioned, one of the most important hydraulic components in terms of the proper operation of the system is the differential valve. The contents of the valve subsystem are shown in Figure 5.

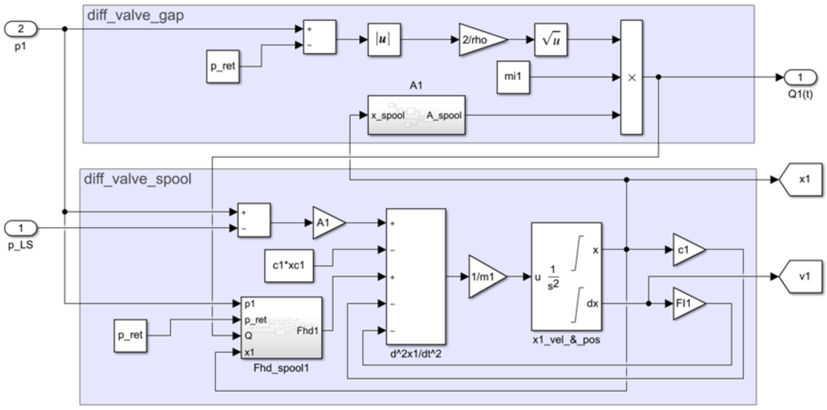

Figure 5. Differential valve subsystem in Simulink.

The diff_valve_spool panel contains a block diagram of the spool motion Equation (Equation (10)) which includes a block for calculating the flow forces according to Equation (11). Similarly, the diff_valve_gap panel represents an implementation of the equation of flow through the spool gap (Equation (13)), where the $A1$ block is used to determine the gap area as a function of the spool position $A_1 = f(x_1)$ (Equation (12)). The values of the physical model parameters are summarized in Table 1, while the solver configuration settings are presented in Table 2.

Table 1. Physical model parameters.

Name	Denotation	Value	Unit
Nominal pump delivery	Q_p	80	$dm^3 \cdot min^{-1}$
Fluid density	ρ	840	$kg \cdot m^{-3}$
Fluid bulk modulus	B_f	1000.0	MPa
Relief valve pressure	p_5	30.0	MPa
Return line pressure	p_{ret}	0.1	MPa
Differential valve spool diameter	d_1	10.0	mm
Differential valve spool movement range	$x_{1,max}$	3.5	mm
Differential valve spring stiffness	C_1	20.9	$N \cdot mm^{-1}$
Differential valve damping coefficient	φ_1	0.95	$N \cdot s \cdot mm^{-1}$
Hydraulic cylinder piston diameter	$d_{c,2}, d_{c,3}, d_{c,4}$	80.0	mm
Hydraulic cylinder rod diameter	$d_{r,2}, d_{r,3}, d_{r,4}$	40.0	mm

Table 2. Solver configuration.

Name	Denotation	Value	Unit
Solver type and name	variable-step, ode45 (Dormand-Prince)	-	-
Initial time step size	Δt_0	10^{-5}	s
Max and min time step size	$\Delta t_{max}, \Delta t_{min}$	10^{-4}, 10^{-8}	s
Relative tolerance	r_t	10^{-3}	-

5. Results of Numerical Simulation and Discussion

The system can operate at the maximum load force exerted on the actuators $F_{i,max} = 150$ kN. All simulations were carried out for both the base system and for the proposed LSB extension. Time series of system parameters were obtained, and the total energy consumption per each operating cycle was determined as:

$$E_{total} = \int_{t_{sim}} p_1(t) \cdot Q_p(t)dt. \tag{14}$$

The research plan covers three main cases of system operation:

- equal actuator loads and fixed control signals for proportional valves,
- various actuator loads and fixed control signals for proportional valves,
- proportional valve control for obtaining equal speeds of actuators with different loads.

In the first case, it was assumed that each actuator is loaded with the same force, while control signals for proportional valves, according to their characteristics obtained during the previous works [22], may vary in the range from $I_{min} = 50\%$ to $I_{max} = 100\%$ every $\Delta I = 10\%$. Time series of parameters such as pressure, flow rate, displacement of actuator pistons were obtained and the total energy demand for each cycle was calculated for the load force: $F_{load,i} = 15, 50, 75, 100, 135$ kN, which corresponds to the percentage values 10, 33, 50, 66, 90%, respectively. Exemplary results for the load $F_{load,i} = 15$ kN are shown in Figure 6 (base system) and Figure 7 (LSB system). Analogous results for $F_{load,i} = 135$ kN are presented in Figures 8 and 9. The summary of total energy consumption determined for all considered cases is shown in Table 3.

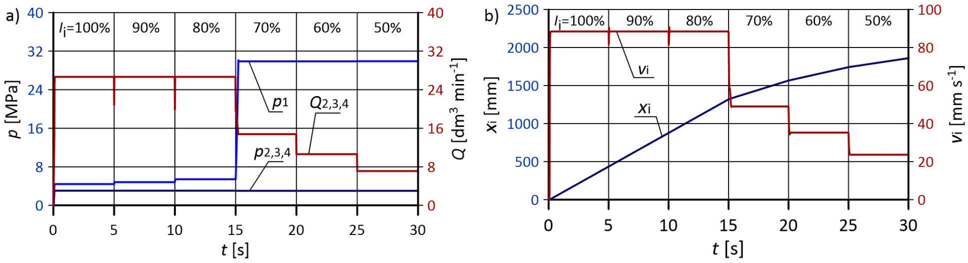

Figure 6. Results of the base system, equal load force $F_{load,i} = 15$ kN; (**a**) pressures and flow rates, (**b**) speeds and positions of actuators.

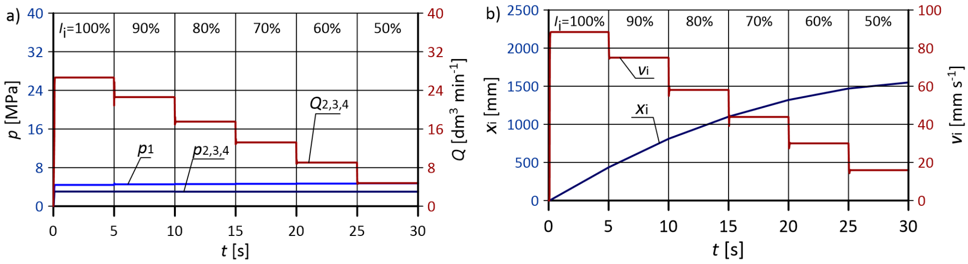

Figure 7. Results of the LSB system, equal load force $F_{load,i} = 15$ kN; (**a**) pressures and flow rates, (**b**) speeds and positions of actuators.

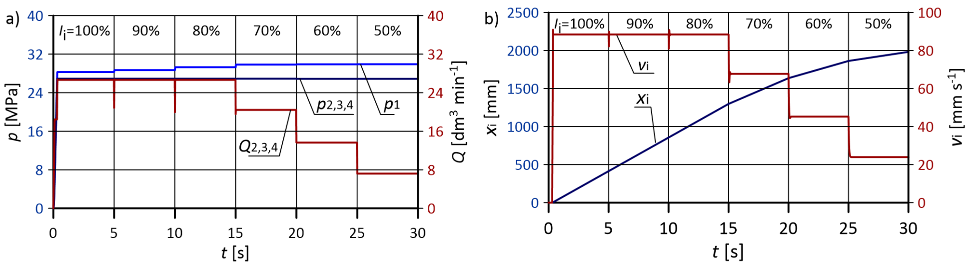

Figure 8. Results of the base system, equal load force $F_{load,i} = 135$ kN; (**a**) pressures and flow rates, (**b**) speeds and positions of actuators.

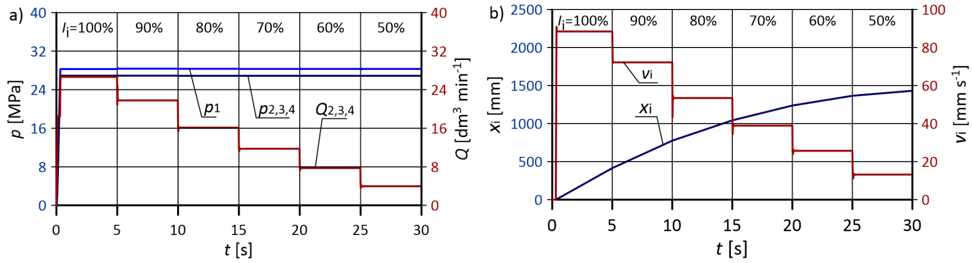

Figure 9. Results of the LSB system, equal load force $F_{load,i} = 135$ kN; (**a**) pressures and flow rates, (**b**) speeds and positions of actuators.

As shown in Figure 6, if the load force in the base system is relatively low, and control signal I_i is higher than the threshold value (70%), the whole pump flow rate is directed to the actuators and p_1 pressure is slightly higher than pressures in actuator lines. However, if the signal falls below the threshold, the supply line pressure is significantly increased to the value set on the relief valve. The energy loss is increasing and the flow rate is dropping rapidly. A similar phenomenon occurs at a high load force (Figure 8), however, in this case the energy losses are relatively lower due to the higher pressure in the actuator lines.

Table 3. Energy consumption, equal load force.

Load Force $F_{load,i}$ kN	15	50	75	100	135
Energy consumption—base, kJ	691.1	830.9	930.1	1028.7	1165.7
Energy consumption—LSB, kJ	183.3	457.7	654.3	850.9	1125.8
Relative reduction, %	73.5	44.9	29.6	17.3	3.4

In contrast, the supply line pressure in the LSB system is slightly higher than the actuator line pressure regardless of the load force. Thus, energy loses are reduced. Additionally, the flow rate can be regulated throughout the range provided by the proportional valves. Thus, energy loses are reduced. Additionally, the flow rate can be regulated throughout the range provided by the proportional valves. The comparison presented in Table 3 shows that the LSB system allows a significant energy saving. The greatest savings, up to 70% may be achieved when the system operates at a minimum load. However, even with typical work cycles, a 20–30% reduction can be obtained.

In the second stage, the system with different actuator loads was tested. Figures 10 and 11 show exemplary results obtained for $F_2 = 37.5$ kN, $F_3 = 75$ kN and $F_4 = 112.5$ kN (25%, 50% and 75% of the maximum load, respectively) in both the base system and the LSB system. The energy consumption for the different load cases is summarized in Table 4.

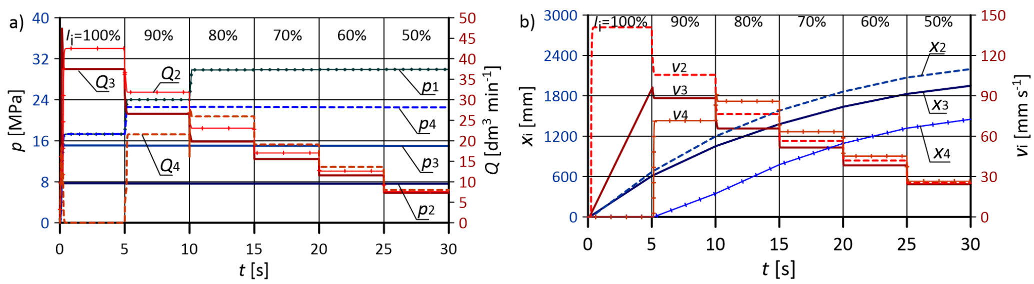

Figure 10. Results of the base system with variable load: $F_{load,2} = 37.5$ kN, $F_{load,3} = 75$ kN, $F_{load,4} = 112.5$ kN; (**a**) pressures and flow rates, (**b**) speeds and positions of actuators.

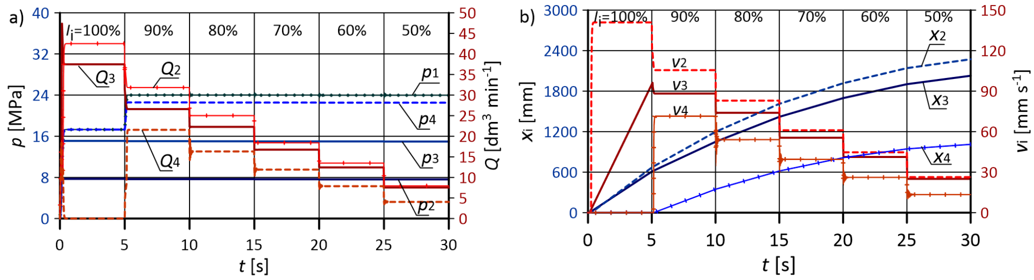

Figure 11. Results of the LSB system with variable load: $F_{load,2} = 37.5$ kN, $F_{load,3} = 75$ kN, $F_{load,4} = 112.5$ kN; (**a**) pressures and flow rates, (**b**) speeds and positions of actuators.

It can be noticed, that the courses of most parameters over time has similar form in both systems, except that the LSB ensures reduction in p_1 supply line pressure. Besides, if the proportional valve control signal is above 90%, the entire pump delivery is used by the less loaded actuators, while the most heavily loaded one is stationary. The comparison in Table 4 shows that during the execution of work cycles under typical loads (20–80% of maximum force), the LSB system can reduce the energy consumption up to 27%. Savings can be even higher in the case of low loads on all the actuators.

Table 4. Energy consumption, different load forces.

Loads: $F_{load,2}$-$F_{load,3}$-$F_{load,4}$, kN	10-50-50	10-50-100	30-60-90	37.5-75-112.5	100-100-135
Energy consumption—base, kJ	828.5	1018.2	1014.3	1067.2	1124.3
Energy consumption—LSB, kJ	455.1	791.1	738.4	911.0	1084.0
Relative reduction, %	45.0	22.3	27.2	14.6	3.6

To date, the simulations have been carried out with the assumption of the same control signals for all proportional valves at a given time. As a result, the pistons under different loads moved at different speeds. In order to obtain equal speeds of all the actuators, it was necessary to appropriately adjust the values of the I_i control signals. Hence, the third stage of the simulations consisted in obtaining the equal speed of the actuators at different loads by setting the control signals of the proportional valves. Load cases analogous to those used in the previous stage were studied. Figures 12 and 13 show results for the case 4: $F_{load,2} = 37.5$ unit kN, $F_{load,3} = 75$ kN and $F_{load,4} = 112.5$ unit kN and the expected piston speed $v_r = 60.0$ mm/s. Table 5 presents the values of control signal set on the individual proportional valves and the energy consumption received.

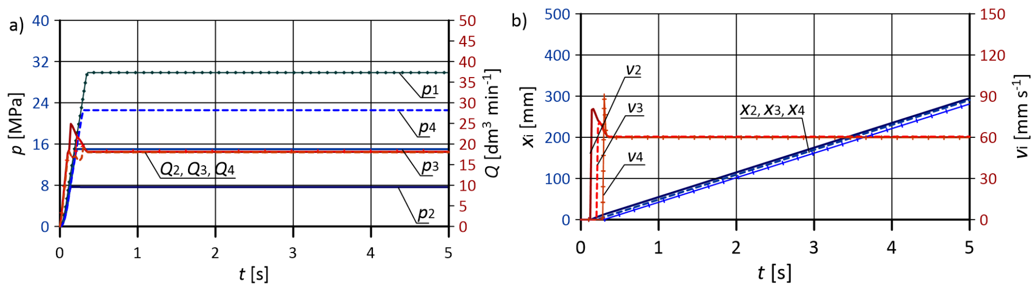

Figure 12. Results of the base system with variable load: $F_{load,2} = 37.5$ kN, $F_{load,3} = 75$ kN, $F_{load,4} = 112.5$ kN and the required piston speed $v_i = 60.0$ mm/s; (**a**) pressures and flow rates, (**b**) speeds and positions of actuators.

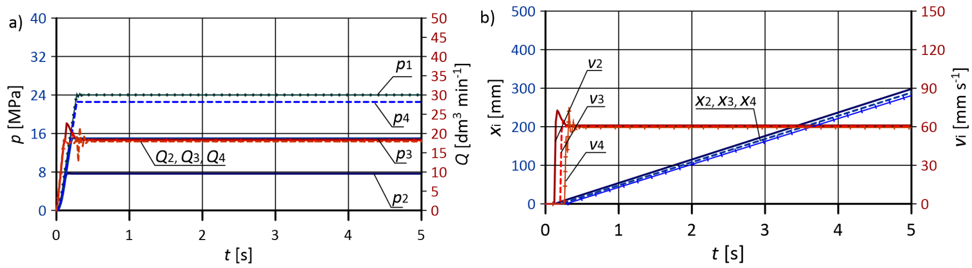

Figure 13. Results of the LSB system with variable load: $F_{load,2} = 37.5$ kN, $F_{load,3} = 75$ kN, $F_{load,4} = 112.5$ kN and the required piston speed $v_i = 60.0$ mm/s; (**a**) pressures and flow rates, (**b**) speeds and positions of actuators.

The obtained results shown in Figures 12 and 13 indicate that the assumed speed of the actuators was achieved in both, base and LSB system. In steady state, the maximum speed difference does not exceed $1.0\ \mathrm{mm \cdot s^{-1}}$ However, the supply line pressure p_1 in the LSB system is approximately 20% lower and reaches the value of 24 MPa. Table 5 summarizes all the tested load cases with percentage values of the individual control signals and the energy consumption. The obtained results show that the reduction in energy consumption obtained by the LSB system compared to the base one is from 4.7% to 60%, with an average value of 29%.

Table 5. Energy consumption, different loads, equal speeds of all actuators.

Loads $F_{load,2}$-$F_{load,3}$-$F_{load,4}$, kN	10-50-50	10-50-100	30-60-90	37.5-75-112.5	100-100-135
Control signals I_2, I_3, I_4—base, %	79, 74, 74	78, 74, 70	77, 73, 71	76, 72, 68	70, 70, 66
Energy consumption—base, kJ	189.7	191.9	191.2	191.4	191.8
Control signals I_2, I_3, I_4 - LSB, %	70, 82, 82	71, 68, 82	72, 68, 83	73, 70, 83	69, 69, 84
Energy consumption—LSB, kJ	75.3	138.3	126.2	155.1	182.6
Relative reduction, %	60.3	28.1	34.0	18.8	4.7

The presented solution is an innovative design of a load-sensing hydraulic system with respect to the state of the art. A comparison with other approaches to load-sensing systems is shown in Table 6. In particular, the following features indicate the novelty:

- although the load-sensing technique is generally well known, it has so far not been used industrially with ISO 4401 (formerly the CETOP standard) in a modular (sandwich) arrangement. Thus, the presented solution is an innovative application in industrial technology,
- the presented system is more universal than the currently used section valves due to the wide possibilities of expansion in a sandwich system by installing various types of valves and advanced configuration of individual sections,
- the number of actuators can be arbitrarily large, limited only by the pump capacity,
- unlike other solutions, only standard, typical hydraulic components were used. There is no need to modify them or use advanced electronic systems and control algorithms, which is crucial for the possibility of practical applications.

As arises from Table 6, the proposed LSB system needs additional hydraulic components, however, neither geometrical modifications, nor advanced control systems nor base system redesign are required. However, in practical applications some certain limitations must also be taken into account. The most important restrictions are:

- practical use is limited to components with ISO 4401 connectors (which means that it is dedicated to a specific type of connection according to the standard),
- the system can only work with distributors which, in their neutral position, relief A and B channels in order to send a proper signal to the load-sensing line (as shown in Figure 1).

- there is a necessity to use a hydraulic lock (double pilot-operated check valve) in the case when the actuator is loaded with force in the neutral position.

Table 6. Comparison of the features of load-sensing solutions.

Name, Publication	Additional Components	Geometrical Modifications	Advanced Control System	Hydraulic System Redesign	Avg/Max Energy Saving Ratio (%)
L-S for EHA pump [1]	Yes	Yes	No	No	n.a.
LSVCA [2]	Yes	No	Yes	Yes	efficiency improved 1.75 times
L-S proportional valve [3]	No	Yes	No	No	n.a.
Stepper converter [4]	Yes	No	No	Yes	30%/60%
L-S for EHSS [7]	Yes	No	Yes	No	62.5%/90%
LS for excavator [9]	No	No	Yes	Yes	fuel saving over 20%
multi-DOF manipulator [10]	Yes	No	Yes	Yes	25.8%/35.3%
LS with reduced SIPL [12]	Yes	No	No	Yes	SIPL reduced by 44%
the studied LSB	Yes	No	No	No	29%/70%

6. Conclusions

In typical industrial applications there are many simple, throttle-operated hydraulic systems controlled by directional or relief valves installed on a standard plate according to ISO 4401. It is a technologically feasible and relatively inexpensive solution. However, if multiple actuators are supplied from a single pump, significant energy losses can occur, particularly in the case when the actuators are loaded unequally. The article shows that the proposed load sensing system (LSB) composed of typical elements can be a beneficial solution ensuring significant energy savings. The LSB system, compared to conventional solutions, requires the use of an additional differential valve and information about the current load of the actuators. Typical logic valves and the 'X' port of the ISO plate were used to provide load information, based on the pressures in the individual actuator lines. The reduction of energy consumption was achieved by using a differential valve, also mounted on the plate. An important issue affecting the dynamical aspects of the system's operation was the appropriate selection of the spring stiffness and the damping coefficient of the differential valve spool. In order to compare the LSB system parameters with a typical throttling solution, an appropriate model was built in the Simulink environment, taking into account the flow and motion equations, and a number of numerical analyses were carried out. Based on the obtained results, the following final conclusions were formulated:

- the presented solution is universal, the LSB system can be easily implemented as well as disconnected,
- the use of standard ISO plates is a convenient solution because it allows interchangeability of components from different manufacturers and easier configuration of control elements,
- the proposed system, depending on the load conditions of the actuators, allowed for energy savings from several to even 60–70%,
- the formulated mathematical model and the developed simulation model make it possible to quickly define various work cycles and assess the dynamics of the system and energy consumption in a given cycle,

- to obtain stable operation of the system, it is necessary to carefully select the spring stiffness and the damping coefficient of the spool in the differential valve,
- the use of an additional differential valve is associated with the appearance of some pressure losses, however, the benefits obtained from its use exceed this cost significantly,
- the proposed LSB system uses the currently available industrial design of a differential valve. In the future, research is planned to reduce pressure losses across the differential valve by optimizing the geometry of the spool and flow paths.

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Nomenclature

Indices	
1	differential valve and supply line
i	actuator line, where $i = 2, 3, 4$
5	relief valve
ret	return line
Parameters	
A_1 A_5	cross-sectional area: differential valve spool, relief valve poppet (m^2)
$A_{C,i}$, $A_{R,i}$	hydraulic cylinder piston area, piston area with the rod area excluded (m^2)
$A_{g,1}$ $A_{g,5}$	flow gap area: differential valve, relief valve (m^2)
B_f	fluid bulk modulus (Pa)
C_1 C_5	spring stiffness: differential valve, relief valve (N m^{-1})
E_{total}	total energy consumption per operating cycle (kJ)
$F_{load,i}$	external force exerted on hydraulic cylinders (N)
I_i	control valve control signal (percentage) (%)
Q_p, Q_{nom}, Q_{max}, Q_1, Q_i, Q_5	pump, nominal, maximum flow rate, flow rate at a certain point (dm^3 min^{-1}, m^3 s^{-1})
$V_{0,i}$, V_1, V_i	volume: initial, supply line, actuator lines (m^3)
d_1	differential valve spool diameter (m)
$d_{c,i}$, $d_{r,i}$	hydraulic cylinder piston diameter and rod diameter (mm)
m_1, $m_{p,i}$, m_5	mass: differential valve spool, pistons with rods, relief valve poppet (kg)
p_1, p_i, p_{ret}, p_{LS}, Δp	pressure: supply line, actuator lines, return line, load-sensing; pressure drop (MPa)
t, t_s	time, start-up time (s)
x_1, x_i, x_5,	position: differential valve spool, hydraulic cylinder piston, relief valve poppet (m)
$x_{1,start}$, $x_{5,start}$	initial spring compression: differential valve, relief valve (mm)
α_1	differential valve poppet opening angle (^{o})
λ_1	differential valve jet angle coefficient (-)
μ_1, μ_5	contraction coefficient: differential valve gap, relief valve gap (-)
ρ	fluid density (kg m^{-3})
φ_1, φ_i, φ_5	damping coefficient (N s m^{-1})

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