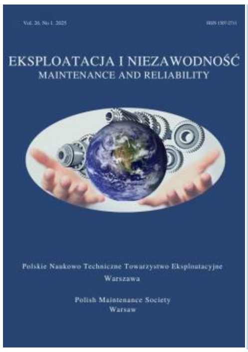

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Data-driven operational pressure estimation for hydraulic actuators fed by fixed displacement pump with variable speed

Indexed by: Web of Science Group

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Highlights

- The data-driven dependence of pressure levels in VSFD systems on operational load and speed.
- The methods of pressure estimation for VSFD systems for given range of loads and speeds.
- Possibility of usage the simplified method of pressure estimation with satisfactory results.

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Abstract

The paper provides the estimation method of the pressure level in the hydraulic drive with actuator fed by variable speed fixed displacement pump (VSFD). The experimental tests were conducted using a typical speed shape in working cycle of the exemplary machine as is the hydraulic indirect elevator. The different piston rod loads and speeds were applied to measure the wide range of pressure values. Obtained data allowed to determine the two-variable estimation functions describing the cylinder inlet pressure values. Pressure mapping quality factors, as the measures of estimation inaccuracy, were introduced. The pressure estimation methods provided satisfactory results in whole range of data. The developed methods allow to support for rapid estimation of operational pressures for hydraulic drives with VSFD pumps, thereby improving the reliability and safety of these systems. Obtained factors can be used in modelling the hydraulic systems and simulations. Further analysis can provide the information for the manufacturers to optimize the hydraulic element sizes.

Keywords
pressure estimation, hydrostatic drive, variable speed pump, frequency converter, elevator

1. Introduction

Nowadays, more and more attention is paid to the energy efficiency of drives already at the design stage. Therefore, the selection of components with an accurate estimate of power demand is important to ensure appropriate forces and moments while maintaining low energy consumption. It is important to know the resistance to movement of the machine, which will operate at different speeds with changing loads.

Investigating the influence of actuator parameters, such as speed and the load of the piston rod, on system pressure is very important from both a control and energy saving point of view. The resistance to motion that occurs significantly affects the operating characteristics. When investigating systems with hydraulic actuators, authors of scientific publications often indicate the relationships between the main operating

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parameters. Recently, a pivotal criterion for assessing hydrostatic systems has become their energy efficiency. Researchers worldwide are actively investigating the energy efficiency of these systems through two distinct approaches: empirical experimentation and computational fluid dynamics (CFD) simulations. The pressure value, and therefore the energy, may be influenced by the valves, accumulators or even pipelines and pipe fitting connection used. The authors reported the findings from an empirical investigation concerning the impact of a hydropneumatic accumulator on a hydrostatic system 1. Additionally, it is possible to employed CFD simulations to analyze hydrodynamic phenomena within hydraulic systems 2.

The working speed of a hydraulic actuator, usually understood as the speed of the piston rod, depends on the area of the working surface on which the hydraulic fluid acts and on other operating parameters, mainly the flow rate 3, 4. In 5, the authors conclude that the response time of the actuator to pressure is dependent on the energy accumulated in the supply line. The output force and piston speed are not the same for the extension and retraction of a piston rod what follows from the difference in working surface areas 6. For double-sided piston rod actuators - the surface areas are the same, so the force and speed are also the same (assuming constant pressure and constant flow rate). Speed can also be affected by different valve types, which can change a flow rate 7, and other methods based on volumetric control 8.

Additionally the speed of the hydraulic actuator may also be influenced by the detrimental impact of mechanical vibrations acting upon the directional control valve that governs the actuator. This issue is thoroughly examined in the referenced paper 9.

Resistance to actuator motion can be related to a number of factors, such as friction or viscosity of the working fluid 10, 11. The occurrence of resistance usually reduces the uniformity of actuator movement and also may affect control accuracy 12. In 11 and 13, a study of the non-uniformity of actuator movement during extension and retraction is discussed. The tests were carried out for four different speeds (as the throttle valve setting increases, the resistance to motion in the system is increased and the speed decreases) and four different loads (achieved by the bypass valve setting). It was shown that the uniformity increases for higher speeds. The value of the friction force increases with the load on the actuator related to the pressure in the working chamber 14. The quality of the working fluid and the flow rate may result in different pressure drop values between the inlet and outlet of the hydraulic actuator or motor. A high flow rate in the motor supplied with water the pressure losses are a dozen or so percent greater than in the motor supplied with oil. However, at low flow rates is the inverse, that is, the pressure losses in the motor supplied with water are about ten percent lower than in the motor supplied with oil 15.

In 16, the effect of speed variation, acceleration and external load on the friction force was investigated. Tests carried out on hydraulic cylinders have shown that the frictional force tested exhibits a typical Stribeck effect, with frictional forces increasing with increasing load during retraction and decreasing with increasing load during extension. For an assumed speed, friction is higher during the advance stroke. In addition, the authors also carried out a dynamic analysis of the friction phenomenon. The friction force increases with the amplitude of the acceleration.

In 17 and 18, authors discuss the issues of modelling the hydraulic system, allowing the creation of dynamic characteristics of the working fluid flow or the working speed of the actuator. The models should take into account the actual factors, friction, possible external load misalignments. Failure to take such issues into account when determining, for example, the safety factors of designed machines, can have disastrous consequences 16, 19-22. In 23, the friction mechanism is discussed using the example of pendulum actuators and a low-speed friction model is built. The model was created on the basis of an analysis of the friction force components The authors proposed a computational model of friction based on deep neural networks. This approach allowed the prediction of friction forces without solving the friction equations from data such as actuator position, pressure and speed. The neglect of certain factors during modelling can lead to the destruction of the actuator. If certain factors are omitted, such as friction, problems may arise during comparing theoretical and experimental results. Modelling often enables early prevention of hydraulic actuator failures, so an inaccurate model may prevent failure prediction.

In 24, the subject of flow rate as an important factor determining the performance and energy efficiency of hydraulic systems is presented. A pilot control of the so-called control pressure was proposed to control the flow rate which is also proportional to the operating speed of the actuator.

In 17 and 25, the relation between the operating parameters of hydraulic systems were determined on the example of the analysis of piston rod braking solutions. A mathematical and geometrical model of a hydraulic cylinder system was created. Pressure and speed characteristics were obtained for various parameters related to damping, so that optimal design parameters for the braking aperture could be determined.

Friction can also be related to the characteristics of the hydraulic actuator seals 11, 13, 26, 27. In 28, issues related to seal wear due to friction were addressed. Seal friction forces during the reciprocating motion of the actuator can result in a deterioration of the actuator's response speed to control signals, a deterioration of the seal quality and, consequently, a reduction in its service life. In the paper, the authors note how important it is to study the characteristic parameters of friction at the seal in order to analyze the effect of compensating pressure on it. In 27, authors described the effect of piston seal modification on the resistance to motion of the piston rod of a hydraulic actuator were described. Low operating pressures and viscous fluids, a tight fit between the piston and cylinder can provide adequate sealing. However, such situations do not occur very often, so the issue of seals is very important in today's hydraulic applications. Seals, however, cause resistance to movement depending on the pressure in the chamber.

Issues related to pressure estimation depending on the operating parameters are interesting both in the aspects of the design of hydraulic systems, modeling and ensuring their adequate durability.

In the available literature, there are no information on the methods of determining pressure during steady motions of the hydraulic drives with possibilities of free shaping the piston rod speed. Currently available research results only include pressure measurements in constant speed systems and are much simpler than in systems where the speed changes during the work cycle. The pressure can only depend on the load and/or the throttle valve settings in the case of throttle regulation. The possibility of a volumetric change of speed during the work cycle introduces another factor that affects the working pressure of the actuator supply system, namely speed.

This paper presents tests covering the entire permissible range of operating speeds and payload masses. The developed, in this paper, methods allow for the estimation of pressure values for an arbitrarily loaded actuator that moves at any assumed speed.

The most common machines with hydraulic drive which operate with constant loads and speeds are lifting mechanisms. In turn, hydraulic elevators are devices that, due to the operational requirements specified by the relevant provisions and standards (EN 81-20, EN 81-50, ISO 8100 and ASME A17.1CSA B44), operate at different speeds during a single work cycle (travel from one stop to another). Due to different piston rod speeds, different loads on the hydraulic system may be generated due to e.g. different flow resistances. A typical system where piston rod moves at different speeds in one cycle is an hydraulic elevator. The speed change can be realized in various ways described e.g. in [5], including also the new control of the indirect elevator using a hydraulic drive with variable speed fixed displacement (VSFD) pump obtained using a frequency converter. The use of such a drive also makes it possible to check the operating parameters of a hydraulic actuator moving at different speeds. These types of drives are increasingly entering the hydraulics market, hence the ability to estimate working pressures in the full range of available pump and motor speeds is important both for designers of hydraulic systems and in the development of accurate models of hydrostatic drive systems.

This paper presents experimental research on the hydraulic indirect elevator with a rope system, which allowed the development of equations describing the pressure levels at the inlet of the actuator in the wide range of loads and speeds of the fixed displacement pump, for both directions of movement. The structure of this paper is presented as follows: Chapter 2 presents the test stand and the course of a typical work cycle, Chapter 3 presents experimental research and the method of calculating the pressure as a function of the load of the piston rod and speed of the hydrostatic pump, along with estimating its accuracy. Chapter 4 contains an analysis of the results as well as conclusions from the conducted research.

2. Test Stand

The indirect rope-elevator supplied by the hydraulic actuator is equipped with a drive allowing free shaping of lifting and lowering speed using the volumetric method of control. The scheme and photos of the test stand are presented in Fig. 1 and Fig. 2.

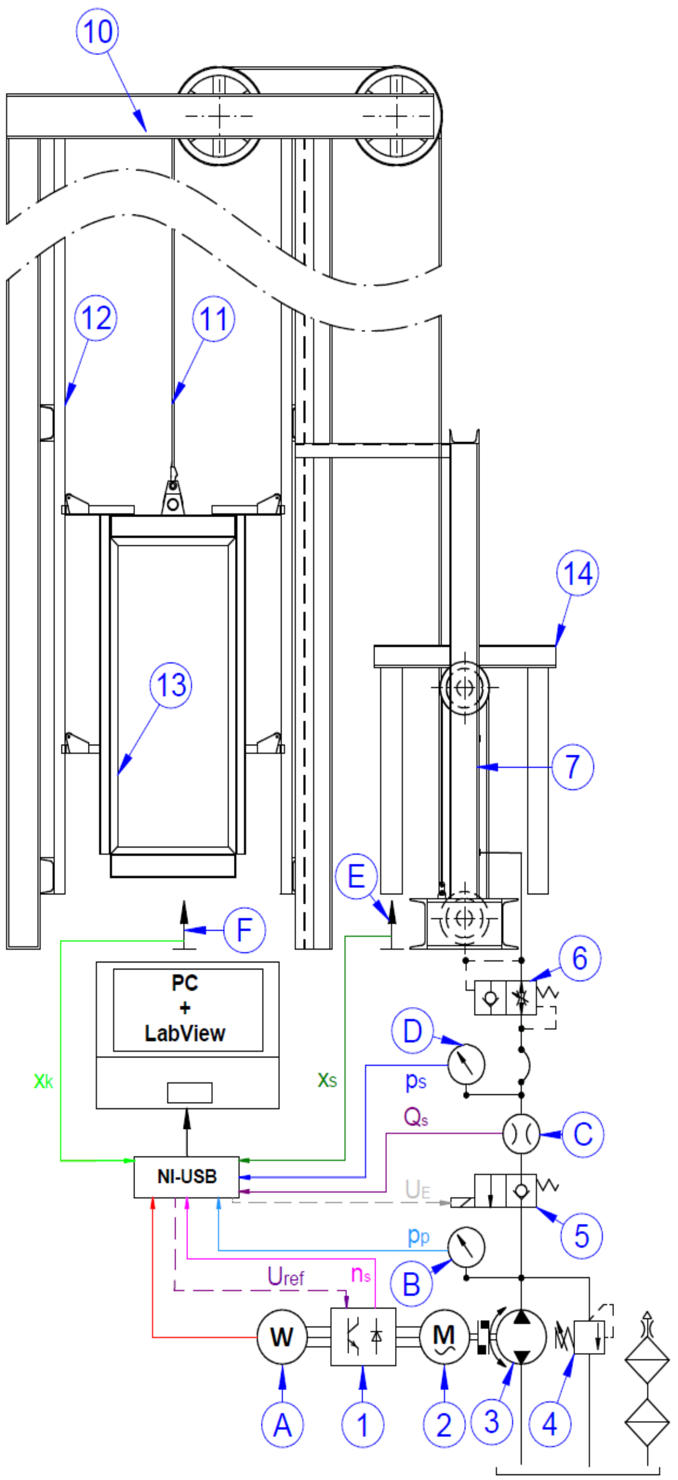
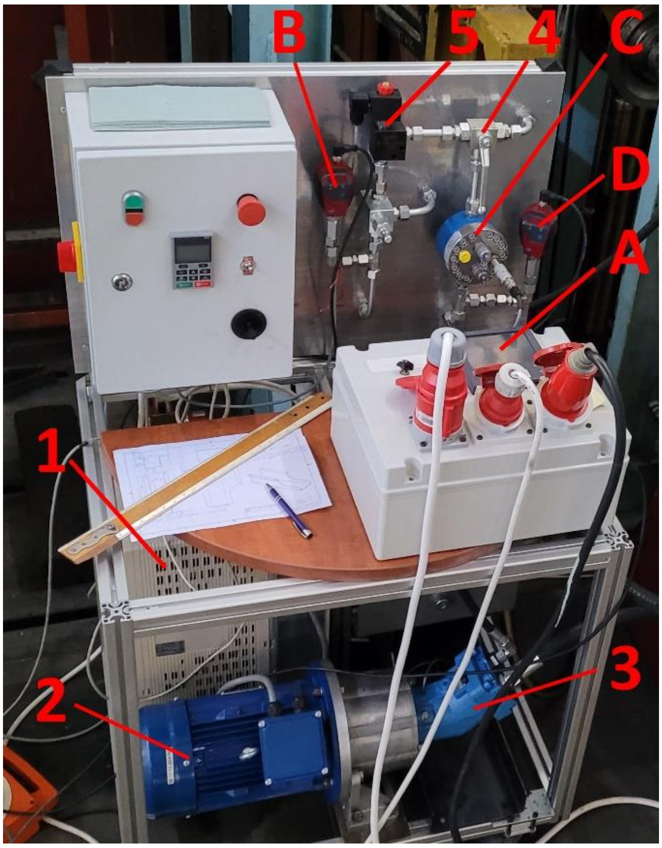


Fig. 1. Scheme of the test stand.

a)



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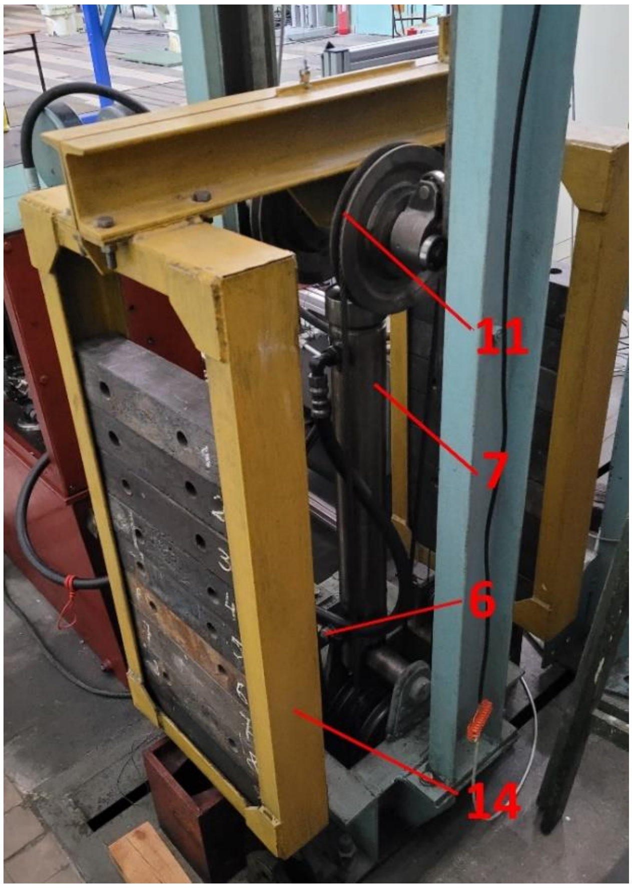


c)

Fig. 2. The test stand with the new drive system.

The elevator structure consists of the steel structure (10) and guides (12) of the car (13). The car (13) is connected with an actuator with the rope system (11) with the transmission ratio equal to 4. The actuator is loaded with the fixed mass (14) attached directly to the piston rod of the cylinder (7). The electric motor (2), controlled by the vector controlled frequency converter (1), is driving the fixed displacement piston pump (3).

In case of lifting the load, an electric motor controlled by the frequency converter drives the fixed displacement pump, so that the working fluid is directed to the actuator (7) by the solenoid-operated check valve 2/2 (5) and through the rupture valve (6). When the car (13) is lowering, the solenoid-operated check valve (5) is in the position allowing the oil to flow from the actuator to the pump. The frequency converter (1) controls the speed of the motor and hydraulic pump shaft, allowing controlled operation of the car including standstill. The hydraulic system is also equipped with the pressure relief valve (4), limiting the maximum operating pressure. The test stand is equipped with the measurement system, including:

- grid parameters analyzer (A),
- pressure sensors (B, D),
- flowmeter (C),
- position cable sensors of the hydraulic cylinder and the car (E, F)

The signals are sent to control and measurement card (9) and then to a computer (8). All components of the test stand and their parameters are presented in Table 1. The element numbers were correlated with the markings in Fig. 1 and Fig. 2. The operational parameters of the elevator are presented in Table 2. With such components, the total displacement of the pump at the nominal motor speed is 17.16 lpm, which gives the piston rod speed of 0.056 m/s (3.41 m/min) and the car speed of 0.23 m/s (13.65 m/min).

Table 1. Components of the test stand.

No	Component	Parameters
1	Yaskawa CIMR-UC4E0014AAA	400 V, 5 kW, 14 A
2	MS100L3-4 B5	400 V 4kW, 1430 rpm
3	Hydro Leduc XPi 12 0523820	12 cm^3/rev, 38MPa, 0-3150 rpm
4	Ponar A-VMP-PIB-12-SP	35 MPa, 70 lpm
5	Winner Hydraulics EP-08W-05-M-05	35 MPa, 30 lpm
6	VUBA-03-FF-S	35 MPa; 67 lpm
7	Hydro-Com UCJ2F-16-80/45/500Dz	80/45/500, 16 MPa
8	PC + LabVIEW Software	Dell Inspiron 17 G3 3779
9	NI USB 6434	16 AI, 2 AO, 24 DIO USB
A	LUMEL ND45	400 V AC; 20/5A; acc. ± 0.1%
B, D	Hydrotechnik MultiEpc 100	0 – 40 MPa; acc. ±0,2%
C	QG 110 3185-03-S-35.030	0,2 – 30 lpm; acc. ±0,7%
E, F	Hohner Automation 90.1404.FX+58-11112-2000	4 m; acc. ±0,03%

Table 2. Operational data.

Components	Parameters
Tower height	~7.5 m
Cylinder stroke	0.5m
Car displacement	2 m
Piston rod nominal speed	0.06 m/s
Car nominal speed	0.24 m/s
Nominal pressure	16 MPa
Mass of the car	158 kg
Fixed mass mounted to the piston rod	366 kg
Nominal car load	732 kg
Rope system transmission ratio	4

Duty cycle

The considered duty cycle consists, according to Fig. 3, of the lifting phase, standstill phase and lowering phase. The lifting and lowering phases are performed with variable speeds thanks to the frequency converter. The level speeds are changed using virtual limit switches described respectively H_1, H_2, H_3 and H_4.

The line above the time axis represents the lifting phase and the line below the time axis represents the lowering phase. Signatures from t_1 to t_6 mean the times of speed changes corresponding both the virtual limit switches operation (H_1-H_4) and times of acceleration (deceleration) set on the frequency converter. The standstill duration is signed as t_p. The test stand is controlled using the LabView graphical programming environment.

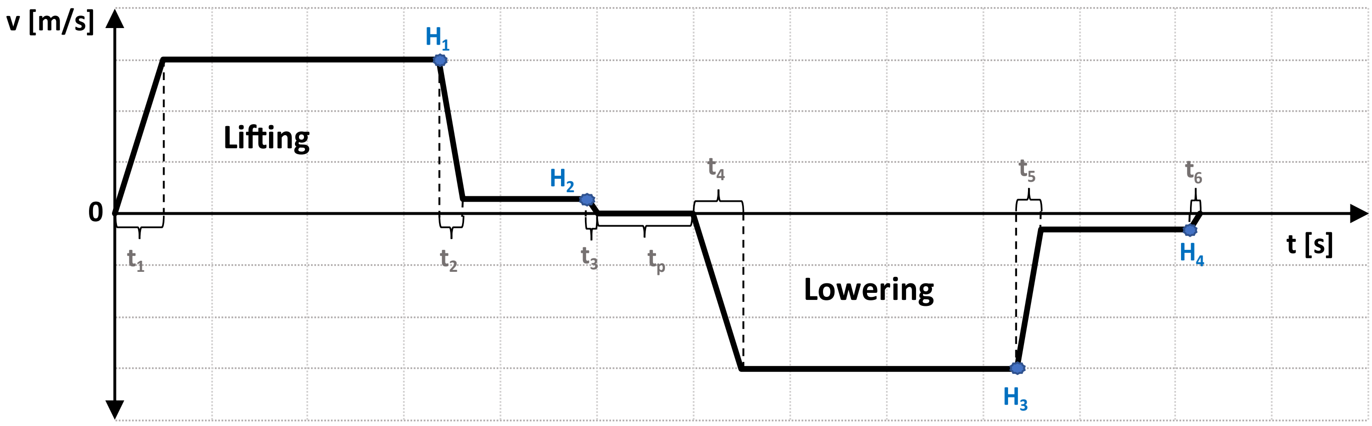


Fig. 3. Scheme of car speed during lifting and lowering.

3. EXPERIMENTAL TESTS

The tests were carried out using the duty cycle shown in Fig. 3 with different loads in the car from zero to the nominal payload mass and different motor speeds causing different piston rod speeds. The speed of the piston rod is not always possible to measure, what could be caused by technological, design or, at least, economic reasons. Due to the increasingly popular use of frequency converters in hydrostatic drives, it becomes possible to easily measure the speed of motors driving pumps. Therefore the speed of the motor was considered in further calculations. In Fig. 4 the charts of actual motor rotational speed n, piston rod speed v_s and pressure on the cylinder inlet p are presented. The tests were conducted at oil temperatures between 22 and 28 degrees Celsius. Average values of motor speed and actual pressure were determined for steady movement during lifting (red rounded square) and lowering (green rounded square). These values are presented in Table 3 and Table 4, respectively.

Based on the data obtained, the approximation functions for each tested payload mass were developed, using LSA (least squares approximation) method. These functions and corresponding equations for lifting and lowering the payloads with different speeds are presented in Fig. 5.

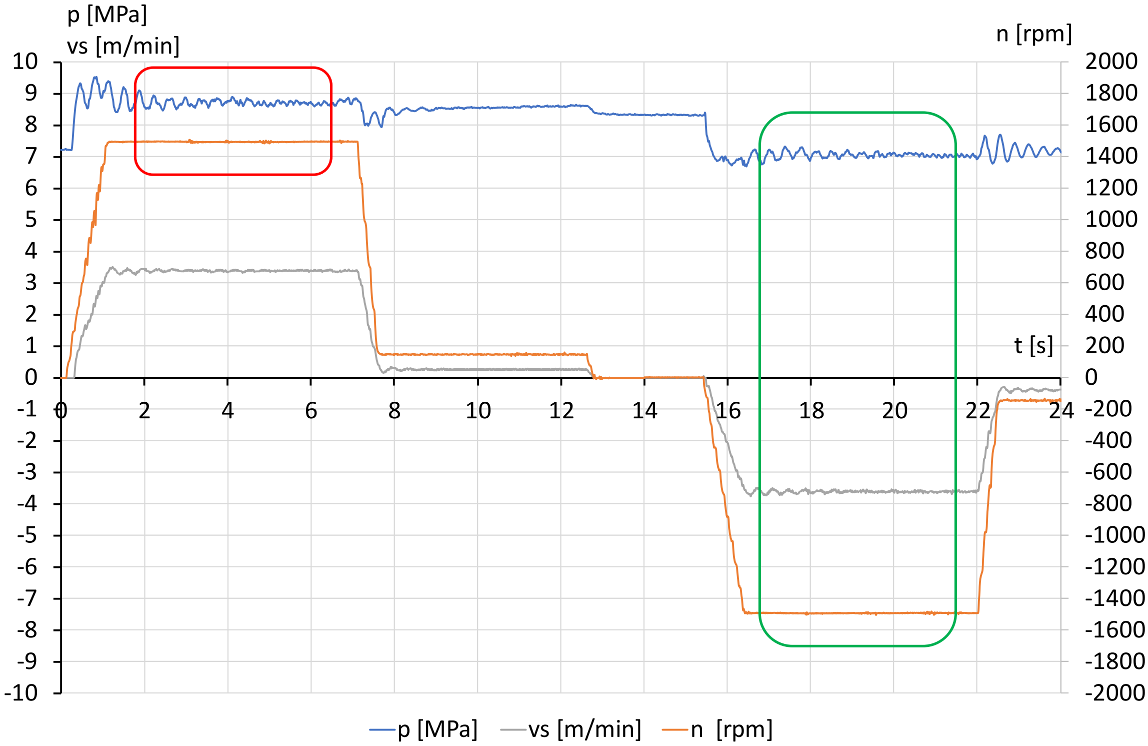


Fig. 4. Charts of actual motor, piston rod speed and pressure on the cylinder inlet.

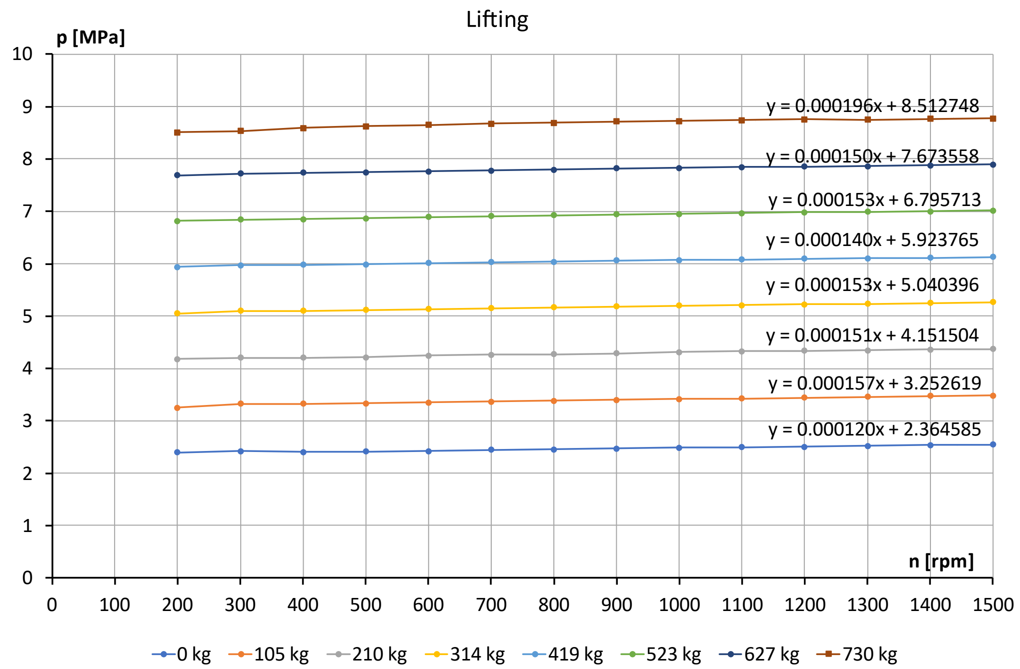
Table 3. Data for lifting.

Series		Motor speed [rpm]													
No	m_Q [kg]	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
1	0	2.40	2.42	2.41	2.42	2.43	2.44	2.45	2.47	2.48	2.49	2.51	2.52	2.54	2.55
2	105	3.25	3.32	3.32	3.33	3.35	3.37	3.38	3.40	3.41	3.43	3.44	3.46	3.47	3.48
3	210	4.18	4.21	4.21	4.21	4.24	4.26	4.27	4.29	4.31	4.33	4.33	4.35	4.36	4.37
4	314	5.05	5.10	5.10	5.11	5.13	5.15	5.17	5.18	5.20	5.21	5.23	5.24	5.25	5.26
5	419	5.94	5.97	5.98	5.99	6.01	6.03	6.04	6.06	6.07	6.08	6.09	6.10	6.11	6.13
6	523	6.82	6.85	6.85	6.86	6.89	6.91	6.92	6.94	6.95	6.97	6.98	7.00	7.00	7.02
7	627	7.69	7.72	7.73	7.74	7.77	7.78	7.80	7.82	7.83	7.84	7.86	7.87	7.87	7.89
8	730	8.51	8.53	8.60	8.62	8.65	8.67	8.70	8.72	8.73	8.74	8.76	8.75	8.77	8.78

Table 4. Data for lowering.

Series		Motor speed [rpm]													
No	m_Q [kg]	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
1	0	1.93	1.92	1.91	1.88	1.87	1.86	1.84	1.83	1.82	1.79	1.77	1.75	1.74	1.73
2	105	2.68	2.66	2.65	2.62	2.62	2.61	2.57	2.57	2.55	2.52	2.52	2.48	2.49	2.47
3	210	3.45	3.43	3.41	3.40	3.39	3.38	3.34	3.35	3.33	3.28	3.29	3.26	3.25	3.25
4	314	4.19	4.17	4.16	4.14	4.14	4.13	4.10	4.10	4.09	4.07	4.06	4.02	4.02	4.00
5	419	4.95	4.93	4.93	4.91	4.91	4.89	4.87	4.86	4.86	4.83	4.82	4.79	4.80	4.78
6	523	5.74	5.68	5.69	5.65	5.67	5.65	5.62	5.62	5.62	5.57	5.59	5.55	5.57	5.54
7	627	6.48	6.47	6.42	6.38	6.38	6.37	6.38	6.38	6.37	6.35	6.33	6.30	6.30	6.28
8	730	7.44	7.27	7.26	7.26	7.24	7.30	7.23	7.21	7.19	7.15	7.20	7.14	7.11	7.11

a)



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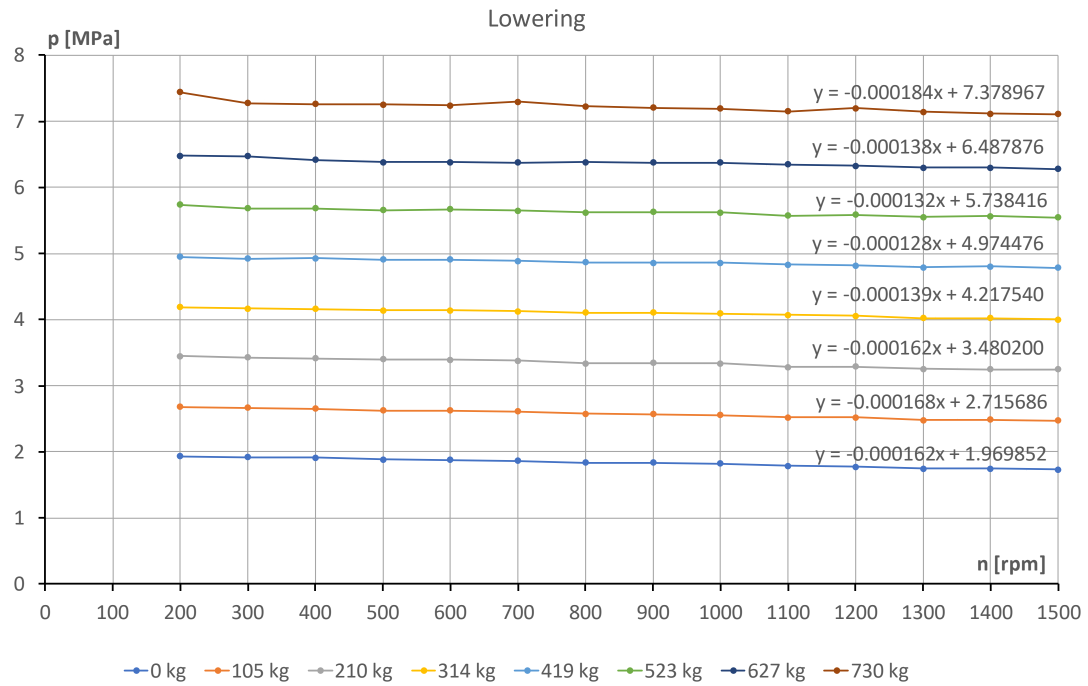


Fig. 5. Pressure values shown as linear functions depend on the rotational speed of the electric motor shaft. a) lifting b) lowering.

As it can be seen in Fig.5 for each payload mass the pressure is described as linear function:

$$y = C_i \cdot x + p_{n0i}$$

where y means the estimated pressure, x is the measured speed, C_i are the slope coefficients of the linear function and p_{n0i} means the pressure at zero speed both for each series with different payload masses.

4. PRESSURE ESTIMATION METHODS

Due to the same nature of the graphs in Fig. 5 and similar values of the slope coefficients of the equations C_i, an attempt was made to generalize the description of the working pressures in the system. The result was a two-variable function which describe the pressure values for the planned range of payload masses in the car and electric motor shaft speeds.

$$p_U = p_{U0} + A_U \cdot m_Q + B_U \cdot n$$

$$p_D = p_{D0} + A_D \cdot m_Q + B_D \cdot n$$

Equations and parameters designation for lifting and lowering phases are signed, respectively with index U and D, where

p_U, p_D - estimated pressure values [MPa],

p_{U0}, p_{D0} - estimated pressure values at zero speed and without payload (p_{n01}) [MPa],

m_Q - mass of the payload in the car [kg],

n - rotational speed of the electric motor shaft [rpm].

$A_{U(D)}$, $B_{U(D)}$ - coefficients of the equations

The above coefficients are calculated as averaged from all considered series, where l_m is a number of payload mass measured, and calculated as follow:

$$A_{U(D)} = \frac{\sum_{i_2}^{i_m} \frac{(p_{n0i} - p_{n01})}{m_Q}}{l_m} \qquad \left[\frac{N}{m^2 \cdot kg}\right]$$

$$B_{U(D)} = \frac{\sum_{i_2}^{i_m} C_i}{l_m} \qquad \left[\frac{N}{m^2 \cdot rpm}\right]$$

The coefficients A, B and p_0 for both movement directions and all measured data (series 1-8), were calculated and are presented in Table 5.

Table 5. Equation coefficients basing on all data.

Coefficient	Lifting (U)	Lowering (D)
p_0	2.364585	1.969852
A	0.0084797	0.0072081
B	0.0001525	-0.0001506

The differences between the measured (with index r) and estimated values of pressure Δp in the whole measured range were presented Fig 6. The differences were calculated as follow:

$$\Delta p_U = p_{rU} - p_U$$

$$\Delta p_D = p_{rD} - p_D$$

Pressure mapping quality factors, as the measures of estimation inaccuracies, were calculated according to the following relationships:

- for lifting

$$W_{pU} = \frac{\sum_{i_1}^{i_m} \frac{\sum_{j_1}^{j_k} |\Delta p_U|}{k}}{l_m}$$

- for lowering

$$W_{pD} = \frac{\sum_{i_1}^{i_m} \frac{\sum_{j_1}^{j_k} |\Delta p_D|}{k}}{l_m}$$

where

k number of speed measured

l_m number of mass measured

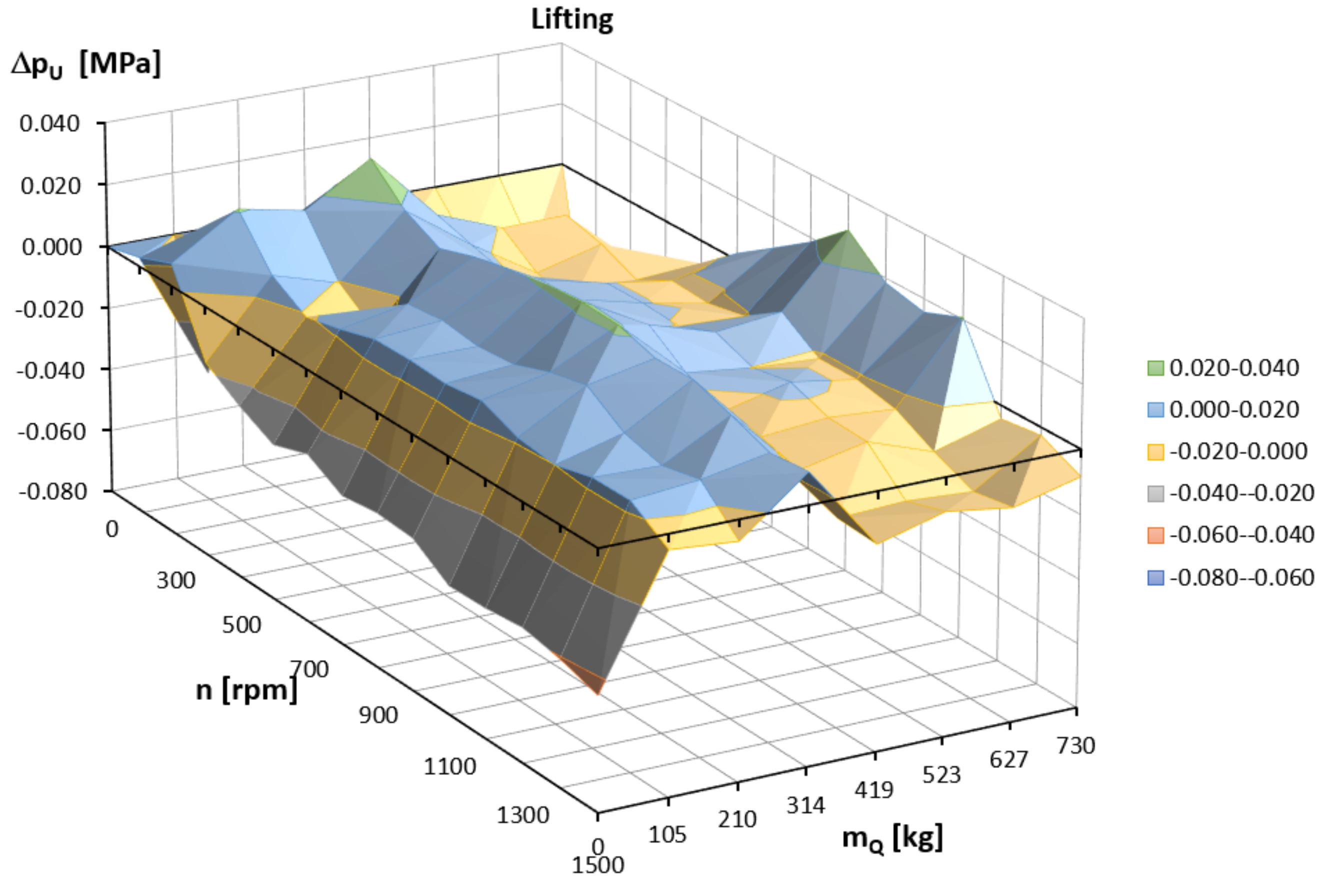
The following results were obtained: W_{pU} = 0.012 MPa and W_{pD} = 0.028 MPa.

Analyzing the waveforms shown in Fig. 6, it was noticed that for most of the measured series the results for Δp were similar. Therefore, it was decided to reduce the number of analyzed runs to half. The half of the data (with index ½), related to smaller loads from 0 kg to 314 kg (series 1-4) was taken as the basis for determining the equations. The remaining data were used to determine the quality of the developed equations and were presented in Table 6. Satisfactory values of introduced factors were achieved as follows: W_{pU} = 0.013 MPa and W_{pD} = 0.041 MPa.

Table 6. Equation coefficients basing on half of data.

Coefficient	Lifting (U)	Lowering (D)
$p_{1/2}$	2.364585	1.969852
$A_{1/2}$	0.008496	0.0071512
$B_{1/2}$	0.00011453	-0.0001455

An attempt was also made to compare two series, in which the first one is always an unloaded course. Factors presented in Table 7 were calculated for equations determined from all, a half of the data and also for seven cases of coefficient calculations basing on two measuring series (consisting of obligatory 0 kg payload mass movement – (marked as 1) and chosen payload mass movement series – from 105 – 732 kg marked as, respectively 2-8).



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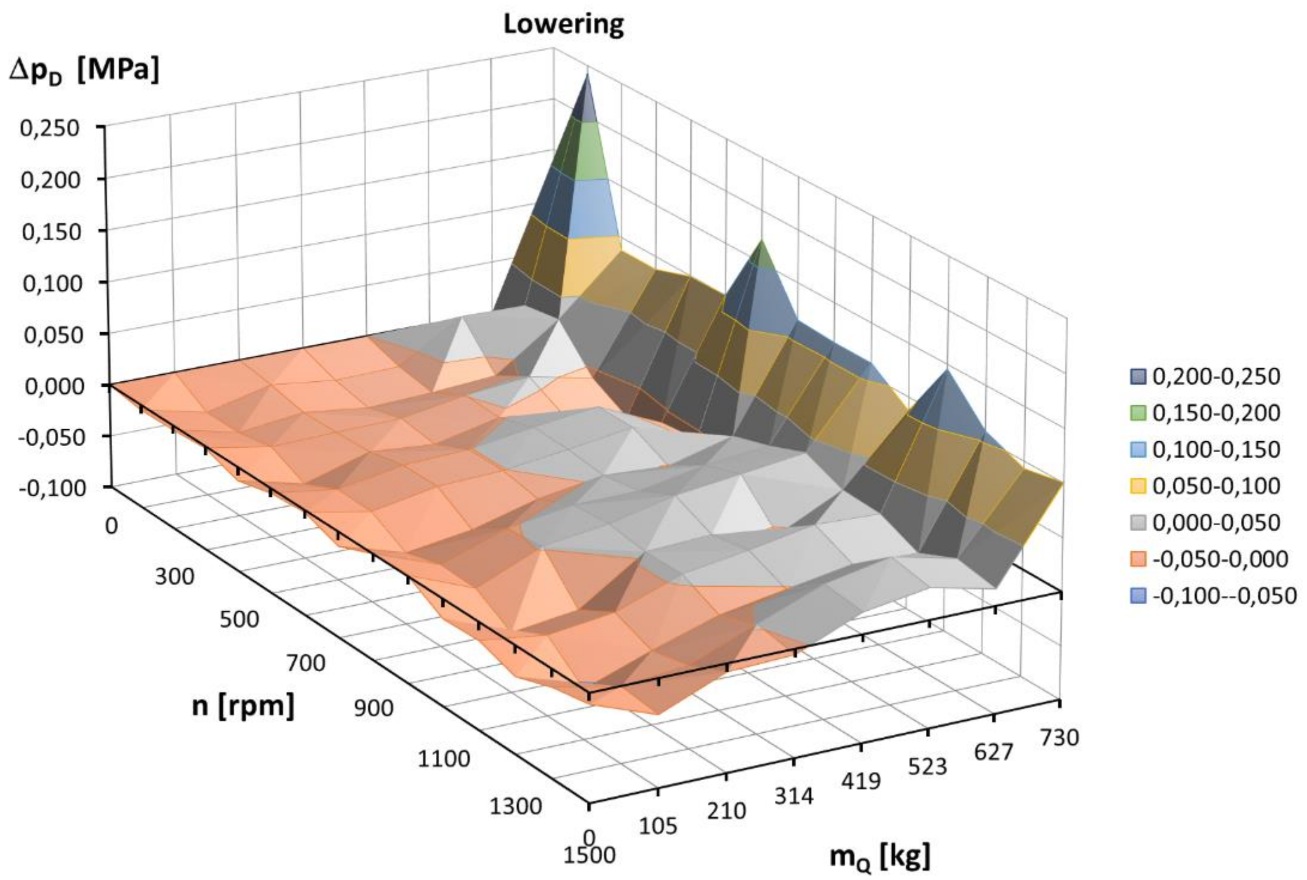

b)

Fig. 6. differences between the measured and estimated values of pressure. a) lifting b) lowering.

Table 7. Pressure mapping quality factors.

Factor	W_{pU} [MPa]	W_{pD} [MPa]
Full data	0,012	0,028
Half data	0,013	0,041
1&2	0,027	0,065
1&3	0,015	0,033
1&4	0,017	0,040
1&5	0,018	0,038
1&6	0,021	0,036
1&7	0,021	0,029
1&8	0,024	0,041

As it can be seen in Fig. 7, the levels of accuracy of the estimations made using the simplified methods (half of data and two selected measurement series) differs slightly from the results obtained using all series, but it does not exceed a few hundredths of MPa, which is considered a satisfactory result. It follows that the proposed method will be suitable for estimating pressure based only on a limited number of measurement series. It is enough to perform measurements for the system without load and with any load, which shortens the time of experimental tests.

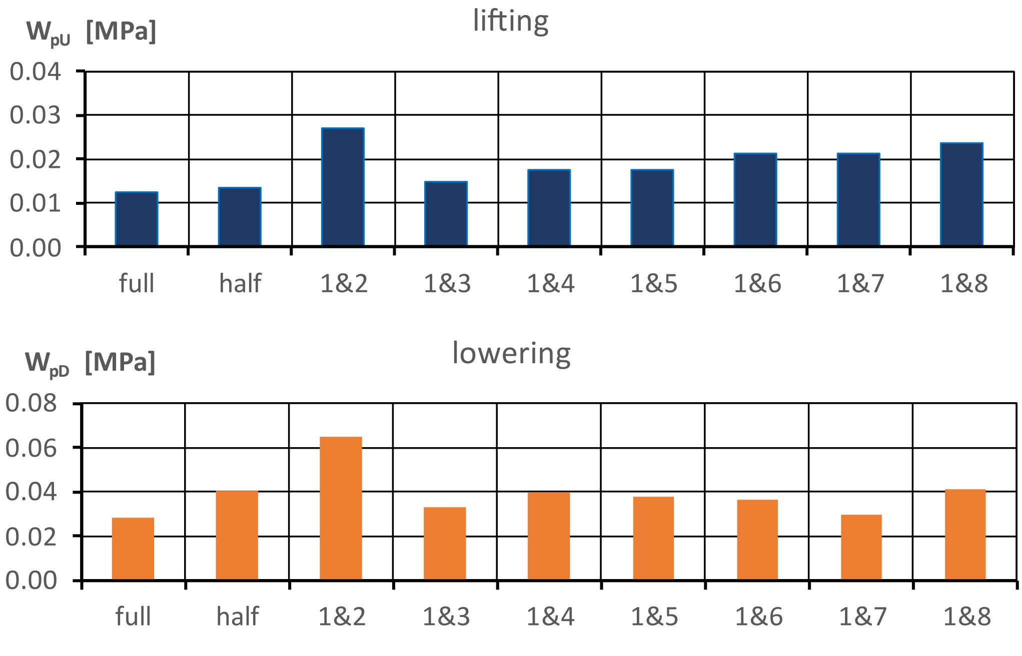

Fig. 7. Comparison of pressure mapping quality factors for different methods of calculation.

5. Conclusion

A new types of hydraulic drives using pump rotational speed shaping to generate an oil flow provide new possibilities for achieving the assumed speeds (positions) of the actuator. Fully conscious design and operation of this type of drives requires, among others: knowledge of the influence of both load and speed of the piston rod on working pressures. The presented research indicate that in drives that are characterized by high repeatability of movements both due to the mapping of speed and the nature of the load, data from the course of the unloaded system and the system loaded with any payload may be sufficient to estimate the pressure map in the full range of operation of the device.

Pressure mapping quality factors, as the measures of estimation inaccuracy, were determined. The presented in this paper pressure estimation methods showed some differences in accuracy, although the results obtained are still at the level of hundredths of MPa, which should be considered satisfactory. The inaccuracies for all pressure estimation methods considered do not exceed 0.27 MPa for lifting and 0.65 MPa for lowering phase.

Testing the system in a wider temperature range is not considered. This results from its nature of operation, where the motion control is not throttling. The operating time and the frequency of use of the elevator is not long so little heat is released, therefore temperature increase is not significant.

The developed methods provide the support for rapid estimation of operational pressures for hydraulic drives with VSFD pumps, thereby improving the reliability and safety of these systems. The presented method of pressures estimation in hydraulic systems with VSFD pumps allows for predicting the system pressure, taking into account the resistances resulting from the load and flow in the entire speed range.

Obtained two-variable functions can be used in modelling the hydraulic systems and simulations to determine resistance coefficients to modeling hydrostatic systems and range of pressures acting on the hydraulic actuator. Further analysis can provide the information for the manufacturers to optimize the hydraulic element sizes and also to provide guidelines for the most favorable operating parameters of hydraulic systems equipped with VSFD and variable displacement pumps.

This may also have a significant impact on the analysis of pressure changes in lifting systems, where the throttle valve is responsible for the speed of the lowering piston rod of the hydraulic cylinder. The obtained pressure range can be used to determine the settings of such valves and determine the appropriate speeds of movement of the machine's actuator.

Further analysis of the developed methods is the subject of the authors' next research, with particular emphasis on their use in dynamic models and simulations of hydraulic systems.

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