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VERIFICATION THE MATHEMATICAL MODEL OF ENERGY EFFICIENCY OF HYDRAULIC SYSTEM

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Abstract

The article presents the laboratory verification of the mathematical description of losses and energy efficiency of the hydraulic transmission with proportionally controlled cylinder supplied by the constant capacity pump in the system of constant pressure is presented. The axial piston pump with pivoting rotor supplied to the system consisted of proportional directional control valve and linear motor – hydraulic cylinder at constant pressure, cooperating with an overflow valve. The choice of the analysed system is not accidental. There is always a view in literature about the very limited energy capabilities of a proportional control system. For this purpose, measurement methods were developed and a test stand was adapted. It consists of two systems: tested and loading. Measurements during the tests were recorded up to date on the computer hard disk. In order to allow for comparison of the total efficiency of the system with the efficiency derived from the simulation, the k_i coefficients determining the energy losses of the individual components were calculated. Investigations have shown a high convergence mathematical description of energy losses in the elements of the system efficiency and reality. This allows accurate simulation determining the energy efficiency of the field at every point in its operation, i.e. at any speed and any load-controlled hydraulic cylinder. The speed and load range of the hydraulic cylinder can also be accurately simulated.

Keywords: *hydraulic systems, hydraulic transmission, energy efficiency, constant capacity pump, energy losses, mathematical model*

1. Introduction

The article aims to show the energy efficiency of a typical hydraulic system with proportional control of a two-rod hydraulic cylinder, with a constant capacity pump cooperating with an overflow valve obtained from laboratory tests and comparing it with the efficiency resulting from the simulation tests.

The article is based on the work done so far [1-3], aimed at describing specific problems related to the determination of structural efficiency, and aims to show with what accuracy the simulation model of energetic behaviour of the propulsion system and hydrostatic control maps the reality.

The study includes the impact of power losses on the pump, the cylinder and the conduits, the drop speed of the pump driving motor, the overflow valve characteristics, the load and the motor speed.

The analysis of the efficiency of the individual elements of the studied structure and the comparison of the efficiency determined simulatably with the obtained laboratory were possible thanks to the elaboration by prof. Z. Paszota mathematical model loss and energy efficiency of the considered system. Based on the mathematical model as well as models related to other structures, energy efficiency simulation programs for hydrostatic drive and control systems were developed. In addition to obtaining an image of the performance of the system, resulting from the hydraulic engine operating parameters and operating conditions of the entire system, it is possible to compare and evaluate the impact of different structures.

The choice of the analysed system is not accidental. There is always a view in literature about the very limited energy capabilities of a proportional control system.

2. The test stand

As the analysed hydraulic system (Fig. 1), a serial throttling control system with a constant capacity pump 4, an overflow valve 5 and a proportional directional control valve 3 was adopted. By connecting the pump 4 and the hydraulic cylinder 1 to the control unit, the test system was run at constant pressure.

The pump draws liquid from the tank, delivers its energy and directs the liquid through the proportional directional control valve to the cylinder, which, at the expense of the liquid pressure energy, performs mechanical useful work.

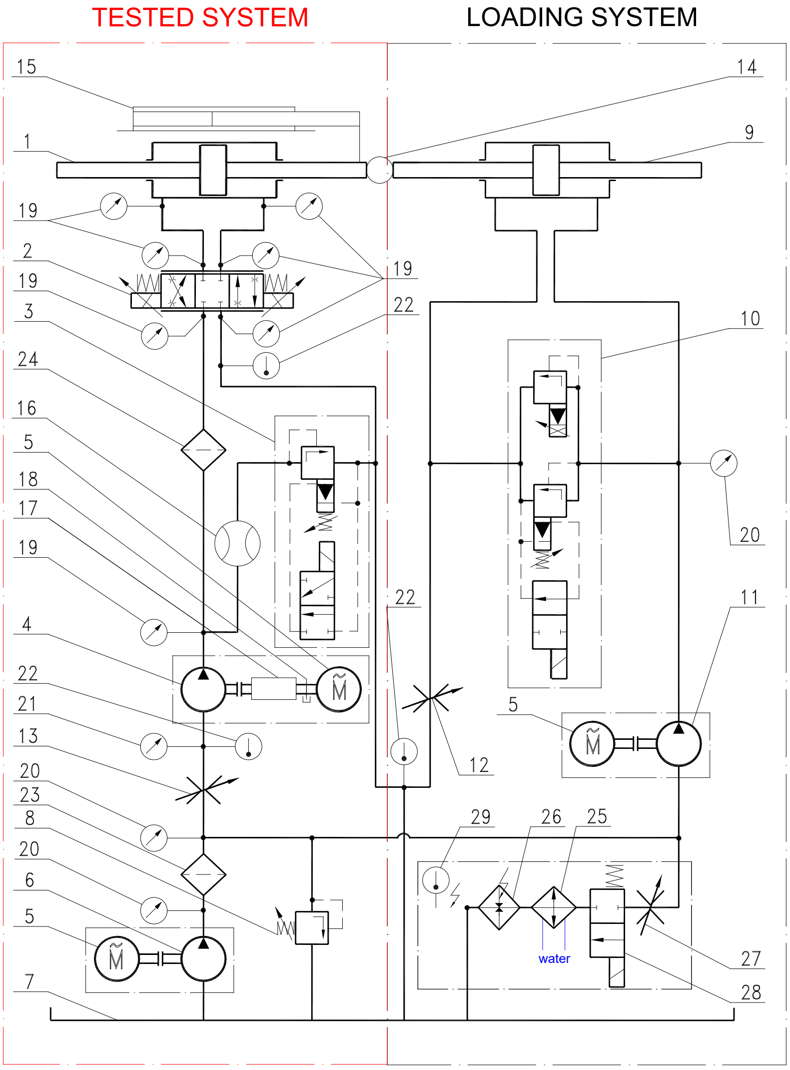

Fig. 1. A test stand layout: 1 – hydraulic cylinder, 2 – proportional directional control valve, 3, 8, 10 – overflow valve, 4, 11 – axial piston pump, 5 – asynchronous motor, 6 – pump, 14 – force sensor, 15 – displacement sensor, 16 – piston flow meter, 17 – torque sensor, 18 – angular velocity sensor, 19 – pressure sensor, 20 – pressure gauge, 22 – temperature sensor, 23, 24 – filter, 25 – heat exchanger, 26 – electric heater, 28 – manifold, 29 – thermostat

3. Characteristics of elements, hydraulic system operation parameters

In order to be able to compare the total efficiency η of the overall system with the efficiency obtained is calculated on the basis of the simulation coefficients k_i, proposed by Professor

Z. Paszota and defining the energy losses of individual system components. The formulas for calculating coefficients are defined in literature positions [1-4].

To assess the maximum value of the simulation test the total efficiency η of the hydraulic cylinder controlled by proportional directional valve (value reached the maximum cross-throttle valve), was mathematically determine the relationship pressure p_{M2i} in the output chamber of the cylinder from indication force F_{Mi} on the piston.

The increase of pressure in the cylinder output chamber caused by the simultaneous throttling of the flow at its inlet and outlet forced a change in the description of the loss and energy efficiency of the cylinder compared to the model of these losses for the hydraulic rotary motor. This is because, for example, it is impossible to describe the machine's volumetric efficiency directly on the basis of the leakage in the cylinder due to the fact that the pressure drop in the cylinder operating as the hydraulic rotary motor may under certain conditions assume a negative value. Due to the fact that the description of efficiency must take into account not the magnitude of the volume losses (which can be negative) and the volume of the volume losses, in monographs [1, 2] define the operating parameters and the power of individual losses characterizing the cylinder.

The elements used in the studied system are characterized by average characteristics of energy losses and average performance characteristics (coefficients k_i of energy losses), namely:

1) Axial piston pump with swing rotor with energy loss coefficients:

$$k_1 = 0.057, \quad k_3 = 0.002, \quad k_{4.1} = 0.039, \quad k_{4.2} = 0.015.$$

2) Electric motor driving a pump with a coefficient of drop rotational speed:

$$k_2 = 0.004,$$

corresponding to a nominal electric engine power of the order of 42 kW.

3) Hydraulic linear motor – double piston rod cylinder with energy loss coefficients:

$$k_{7.1} = 0.031, \quad k_{7.2} = -0.022, \quad k_8 = 0, \quad k_9 = 0.$$

4) Hydraulic conduits with energy loss coefficients:

$$k_5 = 0.021, \quad k_{6.1} = 0.016, \quad k_{6.2} = 0.017.$$

5) Overflow valve with pressure rise coefficient:

$$a = 0.023.$$

It was assumed that the proportional directional control valve used on the test stand was characterized by a nominal intensity equal to the theoretical pump capacity – $Q_{DEn} = Q_{Pt}$. Therefore, at 100% of the control signal current (at maximum cross-section $f_{DE1\max}$ of the throttle slot P → A of the manifold), the pressure drop Δp_{DE1} in this slot is required to reach the intensity $Q_{DEn} = Q_{Pt}$. Then it was determined to be equal to:

$$\Delta p_{DE}\Big|_{Q_{Pt}}^{f_{DE\max}} = 2\Delta p_{DE1}\Big|_{Q_{Pt}}^{f_{DE1\max}} , \tag{1}$$

which was treated as the nominal pressure drop of Δp_{DEn} required by proportional directional control valve.

The nominal pressure drop Δp_{DEn} was thus:

$$\Delta p_{DEn} = \Delta p_{DE}\Big|_{Q_{Pt}}^{f_{DE\max}} = 1.184\ MPa. \tag{2}$$

The coefficient k_{11} of the proportional directional control valve, which is the ratio $k_{11} = \Delta p_{DEn} / p_n$ obtained, at a nominal pressure $p_n = 16$ MPa, the value $k_{11} = 0.074$, and the splitter parameter $\bar{f}_{DE\max}$ of the proportional directional control valve the value:

$$\bar{f}_{DE\max} = \left(\frac{1}{k_{11}}\right)^{0,5} = 3.68\,. \tag{3}$$

4. Description and results of tests

In the work on the mathematical model of energy efficiency of the system, consideration has been given to the energy conservation of the system consisting of the following elements (Fig. 2):

1) double piston rod cylinder with $S_{M1} = S_{M2}$,
2) proportional directional control valve with identical throttling slots, that is $f_{DE1} = f_{DE2}$,
3) real overflow valve, i.e. with $a > 0$,
4) actual hydraulic conduits, i.e. with $\Delta p_{C1} > 0$, $\Delta p_{C2} > 0$, $\Delta p_{C3}' > 0$, $\Delta p_{C3}'' > 0$,
5) constant capacity pump,

with the following assumptions:

6) the pressure p_{P1} at the inlet to the pump $p_{P1} = 0$,
7) the resistances Δp_{Mp1} and Δp_{Mp2} in the cylinder channels are equal to zero – $\Delta p_{Mp1} = 0$, $\Delta p_{Mp2} = 0$, Q_{Mf} volume losses in the cylinder are zero – $Q_{Mf} = 0$.

The tested system, with a constant capacity pump working with the overflow valve, is a constant pressure system, i.e. regardless of the external load of the cylinder; the pump will operate at the nominal pressure p_n resulting from the pressure of the overflow valve. If assume that the pressure drop Δp_M in the hydraulic cylinder is zero and that the losses Δp_{C1}, Δp_{C2}, $\Delta p_{C3}'$, $\Delta p_{C3}''$ of the pressure in the connecting lines is also zero, then the pressure $p_{P2} = p_{SP} \approx p_n$ generated by the pump and the associated overflow valve it will be cut in proportional divider, causing drops:

$$\Delta p_{DE1} = \Delta p_{DE2} = \frac{p_n}{2}\text{, (Fig. 3).} \tag{4}$$

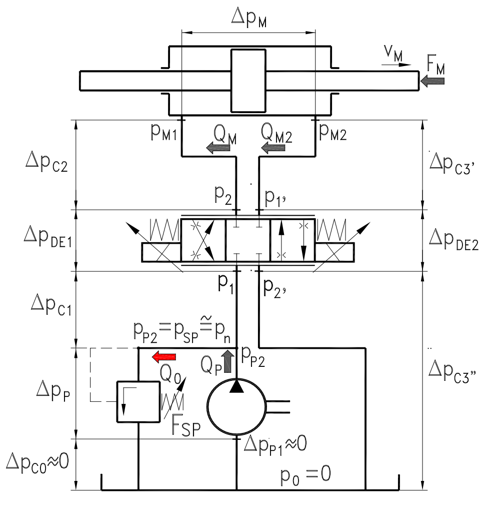

Fig. 2. Hydraulic system with throttling on the inlet and outlet from a cylinder

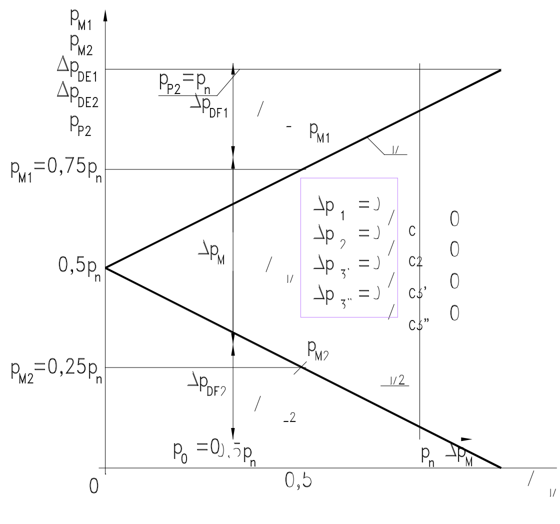

Fig. 3. Pressure and pressure drops in the tested system as a function of pressure drop Δp_M – constant pressure structure

Figure 4 shows a test stand with connection hydraulic conduits.

5. Verification of mathematical model of energy efficiency

Field of work (Fig. 5) of an ideal system with a directional control valve with an unlimited throttle gap (as well as with a pump and an ideal motor) with a relative speed $\bar{\omega}_M$ change of 0 to 1, and a change in relative load $\bar{M}_M$ from 0 to 1 is a rectangle. On the other hand, in a real system in which there are losses, the field of work will decrease. Line 2-3 defining the field of work on the graph $\overline{M}_M = f(\overline{\omega}_M)$ is affected by mechanical and pressure losses in the cylinder, pressure losses in the lines and pressure losses in the proportional directional control valve, and line 3-4 – mainly volume losses in the pump and the cylinder (if present).

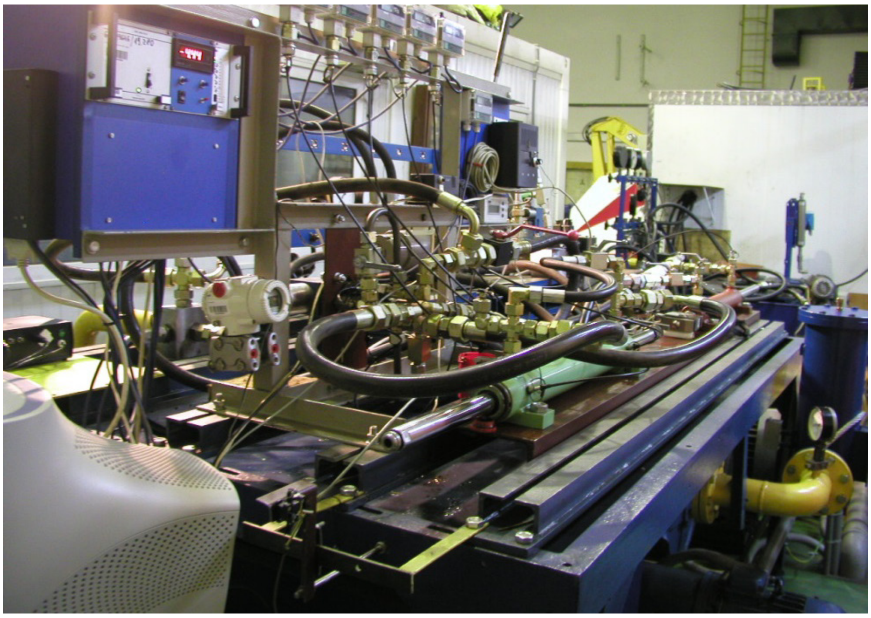

Fig. 4. View of the test stand

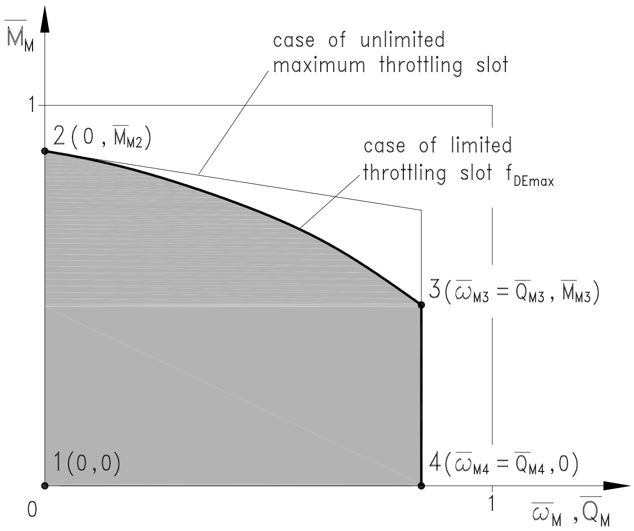

Fig. 5. The range of change flow intensity coefficient $\overline{Q}_M$, speed $\overline{\omega}_M$ and load $\overline{M}_M$ the cylinder in the individual system with constant capacity pump and proportional directional control valve with limited maximal throttle cross section f_{DEmax}; $\overline{\omega}_M = \overline{Q}_M$, because the internal leakage current in cylinder is considered negligible – $Q_{Mf} = 0$

In contrast, in the actual distribution system in which directional control valve cooperating with the overflow valve, the maximum throttle gap of the manifold is limited and causes an additional reduction in the operating field, i.e. the maximum possible load and hydraulic motor speed that can be achieved. The proportional distributor field f_{DE} decides, at the given cylinder load $\overline{M}_M$, how much the intensity Q_M of the stream in the manifold will be, what is the speed $\overline{\omega}_M$ of the cylinder. The increase in pressure drop Δp_{DE} in the manifold is accompanied by the rising intensity Q_M of the liquid in its throttling gaps. Consequently, with the theoretically constant pressure p_1 before the manifold and the decreasing pressure drop Δp_M in the cylinder, the pressure drop Δp_{DE} in the manifold increases. At the maximum load $\overline{M}_M$ of the cylinder, the value of the load factor $\bar{M}_M$ can reach the value $\bar{M}_{M\max} = 1$ in the ideal system. Due to pressure losses in the lines and mechanical losses in the hydraulic cylinder and pressure losses in the manifold, the value $\bar{M}_{M\max}$ is less than 1 and decreases with increasing speed $\bar{\omega}_M$ of the cylinder. Exception may be a situation in which the overflow valve characteristic (coefficient "a" of the characteristic $a > 0$) causes an increase in p_{P2} pressure above p_n. Then, at $p_{P2} > p_n$, the value of the load coefficient $\bar{M}_M$ of the cylinder can, at small values of $\bar{\omega}_M$, exceed $\bar{M}_{M\max} = 1$.

The increased load on the piston rod of the cylinder is accompanied by its decreasing maximum v_M speed. The lower limit of the maximum load $\bar{M}_M$ coefficient of the hydraulic cylinder (line 2-3 in Fig. 5) is due to the fact that the pressure drop Δp_{DE} in the manifold increases. In the case of increasing load $F_M(\bar{M}_M)$, the pressure drop Δp_{DE} in the manifold will fall to zero and the cylinder will stop.

In the system without losses in the pump, in the cylinder, in the conduits and in the throttling manifold (proportional directional control valve), the load coefficient $\bar{M}_M$ and the cylinder speed coefficient $\bar{\omega}_M$ could be varied from 0 to 1. The sum of the losses in the components, with increasing speed, decreases the load capacity of the cylinder to the limit of line 2-3 (Fig. 5), which drops with increasing speed.

Vertical line 3-4 (Fig. 5) is the second line that limits the hydraulic circuit. The pump works virtually at constant pressure, so the volume losses in it are almost constant. There are no volume losses in the cylinder, so line 3-4 is a vertical line.

The effect on the range of change of load coefficients $\bar{M}_M$ and cylinder speed $\bar{\omega}_M$ has: the slot field $f_{DE\max}$ of the maximum proportional flow slot of the proportional distributor (coefficient k_{11}), the volume losses in the pump (k_1), the decrease of the rotational speed of the pump driving electric motor (k_2), coefficients k_5, k_6 pressure losses in connecting conduits, coefficients $k_{7.1}$, $k_{7.2}$ mechanical losses in hydraulic cylinder, coefficient k_8 of cylinder pressure losses and coefficient k_9 the cylinder volume losses.

In the constant pressure system, the basic coefficients determining the instantaneous value of η energy efficiency are current coefficients $\bar{M}_M$ and $\bar{\omega}_M$ of the hydraulic cylinder performance. In addition, coefficients k_1, k_3, $k_{4.1}$, $k_{4.2}$ – losses in the pump and coefficient k_2 of the rotational speed of the motor driving the pump are also decisive.

Figure 6 shows the work area and the stability lines of the tested system. These are the results that have been achieved using the mathematical model prof. Z. Paszota with coefficients "k_i" designated during the research at the test stand.

By enlarging the throttle slot $f_{DE\max}$ in the proportional directional control valve, an increase in the maximum value of η is achieved. In the ideal (without loss) pressure system, the slot expansion to $\bar{f}_{DE\max} = 5.6$ ($p_n = 320$ bar, $\Delta p_{DEn} = 10$ bar) relative to the reference f_0 gives the maximum theoretical efficiency $\eta_{\max}$ of the order of 0.97. Further enlargement of the throttle slot f_{DE} in the manifold (proportional directional control valve), for example up to $\bar{f}_{DE\max} = 10$, results in an efficiency value of η of 0.99. This results in a 2% increase in efficiency, where in this system you have to use a proportional directional control valve twice as large.

6. Conclusions

1. Description of dependency, created by prof. Z. Paszota, describing the energy efficiency η of the system and the range of changes in operating parameters of the hydraulic cylinder (coefficients of speed and load), taking into account the influence of the control and drive characteristics (coefficient "a" of the overflow valve characteristic, coefficient "k_2") of the speed of the motor driving the pump, coefficients k_i of the mechanical, pressure and volumetric losses in the components (in the pump, in the cylinder and in the conduits), k_{11} coefficient or

f_{Demax} parameter of the proportional directional control valve, made it possible to carry out simulations of the energy behaviour of the actual throttle manifold.

2. Model of loss and energy efficiency of hydraulic drive with proportional control of the cylinder, which would use a full description of loss and efficiency of the cylinder itself, would become too complex. Therefore, the pressure losses (flow resistance) in the inlet and outlet ducts of the cylinder ($k_8 = 0$) and the negligible volumetric losses ($k_9 = 0$) are omitted.

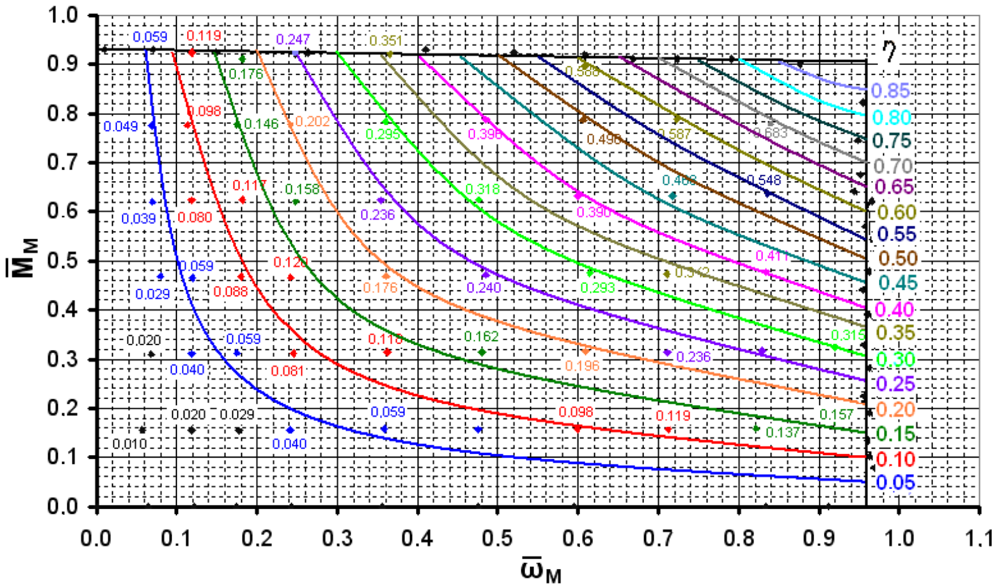

Fig. 6. The field of work and constant efficiency lines η = cte of the tested constant pressure system (lines – defined as simulation and laboratory points)

3. Laboratory experiments with a double piston rod cylinder proportionally controlled allowed: verification of mathematical energy efficiency η of the studied structure and determination of the dependence of the force F_{Mi} indicated on the piston from the force F_M on the piston rod of the cylinder to determine the coefficients $k_{7.1}$ and $k_{7.2}$ describing the simulation, $F_{mi} = k_{7.1} F_{Mm} + (1 + k_{7.2}) F_M$.
4. The absolute error of the simulation method for determining the energy efficiency of a hydraulic system with a proportional control of a cylinder driven by a constant capacity pump (as a difference in laboratory and simulation results) is of the order of 0.003-0.06%.
5. The relative error of the simulation method for determining the energy efficiency of the system (as the ratio of absolute error to the simulation result) is of the order of 0.01-0.66%.
6. It can therefore be stated that the simulation method for determining the energy efficiency of a hydraulic system with a proportional control of a servomotor powered by a constant capacity pump, developed by Z. Paszota, reproduces reality with great accuracy.

References

1. Paszota, Z., *Energy losses in hydrostatic drive*, Monography, LAP Lambert, Acadamic Publishing, 2016.
2. Paszota, Z., *Aspects Energétiques des Transmissions Hydrostatiques*, Monography, Gdansk Univeristy of Technology Publishing, 2002.
3. Paszota, Z., *Układ hydrauliczny z rozdzielaczem dławiącym – model sprawności energetycznej*, Seminar material Hydropneumatyka'96, World Trade Center Gdynia Expo Ltd., 23-25.10.1996, s. 1-8, 1996.
4. Skorek, G., *Charakterystyki energetyczne układu hydraulicznego o sterowaniu proporcjonalnym siłownika zasilanego pompą o stałej wydajności w systemie stałego i zmiennego ciśnienia*, PhD Thesis, 2008.