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STUDY OF LOSSES AND ENERGY EFFICIENCY OF HYDROSTATIC DRIVES WITH HYDRAULIC CYLINDER

Grzegorz Skorek
Gdynia Maritime University, Poland

ABSTRACT

Energy efficiency of hydrostatic transmissions, and especially efficiencies of drives with motor speed controlled by throttle, as well as efficiency of hydraulic servomechanisms can in fact be higher than the efficiency values most frequently given by the respective literature in this field. With the progress achieved in recent years in the development of hydraulic systems it is becoming necessary to develop methods for precise energy efficiency calculation of such systems. It is difficult to imagine that more and more, better and better machines and control elements could be used without the possibility of a mathematical tool at our disposal to enable an accurate analysis and assessment of behavior of the system in which such machines and control elements have been applied. The paper discusses energy savings using mathematical model of losses in elements, the energy efficiency of the system. There are possibilities to reduce energy losses in proportional control systems (in the pump, in the throttle control unit, especially in the cylinder), and thus to improve the energy efficiency of the throttling manifold. The considerations allow for comparison of the loss power resulting from the applied hydraulic control structure of the hydraulic cylinder and the power consumed by the pump from the electric motor that drives it, the power necessary to provide pump-driven hydraulic cylinder. The article shows the impact on the output (useful) power consumed in the considered systems, and the impact on the power consumed of the loss power in the individual elements. The paper presents also formulas of loss power, formulas of energy efficiency connected with investigated hydrostatic drives, two schematic diagrams of hydraulic systems, their principle of operation and problems of studying losses in elements and energy efficiency characteristics of systems consisting of a feed assembly, control set and cylinder. It also includes a subject matter connected with an energy loss power of hydrostatic systems with hydraulic cylinder controlled by proportional directional control valve. Diagrams of loss power of two hydraulic systems worked at the same parameters of speed and load of a cylinder, which were different due to structure and ability of energy saving, were presented and compared.

Keywords: energy losses, energy efficiency, hydrostatic system, control structures, proportional directional control valve, cylinder

INTRODUCTION

The hydrostatic systems play a very important role in modern machines. A great number of the machines constructed nowadays has more or less developed hydrostatic or electric-hydrostatic drive systems and in many cases those systems are the most important parts of the machines. Component elements – hydraulic linear motors (cylinders) – are widely applied in machines used on land and aboard ships. Unquestioned advantages of cylinders are: capability of performing the translational motion, reliability, simple construction, the effective force to weight ratio [2]. A control system with a directional control servo valve or a proportional directional throttling control valve, controlling a linear hydraulic motor (cylinder) is used in the ship steering gear drive, in the controllable pitch propeller control, in the variable capacity pump control system for hydraulic deck equipment motors or fixed pitch propellers in small ships (e.g. ferries) [1].

The electro-hydraulic system as one of the fundamental components has been applied in many other equipment too, such as construction machines, agricultural machines, and airplanes. Common to these applications is that high power is often required to perform the desired work, for example moving material or lifting heavy weights. The power for such drives is often generated by a centralized source, usually an internal combustion engine or a high power electric machine. Using fluid power systems the power is easily distributed via hydraulic lines to either linear or rotary drives. It is estimated that by the year 2000, the world market of electro-hydraulics is about 30–35 billion dollars per *year,* and is in steady growth. Meanwhile, energy saving concerning a hydraulic system has been raised with the numerical application of heavy equipment. The electro-hydraulic system mounted equipment often works around the clock and outputs high power in handling heavy loads. The energy consumption and the waste gas emission of such systems therefore stays high. Take one of the most popular construction machines, 20-t load sensing controlled hydraulic excavator as an example. Such excavator type usually requires a diesel engine of more than 110 kW, which consumes at least 33 liters fuel in an hour. The amount of NOx and CO emitted by this machine is considerable. But only 30% of the consumed energy is used for moving loads. While more than 60% of the energy is consumed in power losses and driving of hydraulic systems. Therefore, even a small improvement in the hydraulic system efficiency will have a significant impact on the total energy efficiency of the machine [16].

There are uninvestigated areas connected with behavior of elements in hydraulic systems with different structures. Unawareness of proportions of the energy, volumetric, pressure and mechanical losses in elements is often the case. Problems connected with energy efficiency are essential for improvement of functionality and quality of hydrostatic drive systems, characterized by unquestioned advantages but also by relatively low efficiency in comparison with other types of drive. Energy efficiency of hydrostatic transmissions, particularly those with throttling control of the motor speed, and also efficiency of the hydraulic servo-mechanism systems may be in fact higher than the values most often quoted in publications regarding the subject. At present, for instance, the efficiency of a system with a servovalve is still often presented incorrectly. Namely, the maximum efficiency of such a system (with ideal pump and hydraulic motor) is quoted to be equal to $\eta = 0{,}385$ at a motor supply pressure equal to $p = 2/3\ p_{max}$. Such an approach leads to the use of higher than required pump unit, operation at lower efficiency, increase in costs of the system itself and its operating costs. In too low efficiency systems there is an increase in load, mainly in pump load, and this can lead to a greater risk of failure, the necessity of repair or replacement and also to decrease of operating life of the system. Too low system efficiency resulting most often from intensive throttling of liquid flow is also the source of quick deterioration of operating properties, mainly lubrication properties of hydraulic oil, due to too high operating temperature of the hydrostatic transmissions. Possibility of calculating the real value of the hydraulic system overall efficiency as a function of many parameters influencing it, becomes a tool of complete evaluation of the designed system quality [3].

The paper presents definitions of power losses and energy efficiencies occurring in two hydraulic systems. It compares laboratory investigated efficiencies of systems with cylinder proportional control and efficiency of the system with volumetric control by a variable capacity pump. Presented are also two schematic diagrams of these hydrostatic systems, their principle of operation and problems of studying losses in elements of systems consisting of a feed assembly, control set and cylinder. The analysis was performed comparing the selected parameters of operation of the hydraulic cylinder power lines of energy losses in the elements of these structures.

In searching for the energy saving solutions, computer-aided methods of calculating the energy efficiency of systems have been developed and improved.

The required speed v_M and load F_M of the driven machine are a result of its operation cycle and tasks to be performed. The driven machine current speed and load values are independent of the type and structure of the machine driving system.

The current speed and load of the hydrostatic system driven machine have a direct or indirect impact on the mechanical, volumetric and pressure losses in the hydraulic motor, pump and other elements of the system with a given motor speed control structure, the losses resulting also from the hydraulic oil viscosity.

Laboratory verification of energy savings was concerned with two hydraulic systems controlled with a proportional directional control valve supplied with a constant capacity pump:

a) with overflow valve – constant pressure structure [p=cte] (Fig. 1),
b) using a pressure-controlled overflow valve from the cylinders' inlet line – variable pressure structure [p=var] (Fig. 2).

The most popular solution is a system (Fig. 1) where a throttling control valve is fed by a constant capacity pump cooperating with an overflow valve. This system, working with a constant pressure, achieves a high energy efficiency value η only at the point of maximum motor load coefficient $\overline{M}_M$ and maximum motor speed coefficient $\overline{\omega}_M$. The pressure drop in the cylinder balances the load acting on the hydraulic cylinder. The proportional throttling valve generates two pressure drops at the inlet and outlet of the cylinder. The pump in the constant pressure system (p=cte) must generate pressure before the overflow valve, which will not be less than the pressure required by the hydraulic cylinder. The cylinder, which is an executive element in the system, may require pressure depending on its load, changing from zero to nominal value. When it reaches the nominal value of the load, the pressure drop in the throttle slots of the manifold tends to zero.

The unit, which consists of a pump and an overflow valve in constant pressure system p=cte is ready to supply the system at maximum pressure and maximum capacity. However, it is not usually used to such an extent that the cylinder at the moment is loaded with force which requires a lower than nominal pressure drop.

This system achieves high energy efficiency, equal to the efficiency of the system without the throttling control, only at the point with the maximum values of the load coefficient $\overline{M}_M$ and the speed coefficient $\overline{\omega}_M$. With decreasing engine load, especially with engine speed dropping, the efficiency η decreases rapidly [6].

There are possibilities to reduce energy losses in proportional control systems (in the pump, in the throttle control unit and in the hydraulic cylinder, and thus to improve the energy efficiency of the throttling control valve.

The hydraulic drive system and the proportional control of the hydraulic cylinder can be supplied with a constant capacity pump cooperating with an overflow valve stabilizing a pressure in proportional directional control valve to the nominal pressure level (Fig. 1), or a pump cooperating with a pressure-controlled overflow valve at the inlet to the receiver – hydraulic cylinder. The variable pressure system p=var (Fig. 2) allows for the reduction of losses in the pump, in the control unit and in the hydraulic cylinder.

In the variable pressure system p=var, the structural pressure losses and structural volume losses in the throttle control unit, mechanical losses in the cylinder and pump, and volume losses in the pump can be seriously reduced. The mathematical description of loss and energy efficiency is presented in the paper.

The variable pressure structure p=var represents the system with a constant-capacity pump cooperating with an overflow valve controlled by the cylinder supply pressure (Fig. 2). It is a cost-effective solution for both the cylinder and the pump as well as the entire control system. Variable pressure system p=var with control overflow valve SPS, the actual throttling valve discharge pressure to the inlet chamber of the cylinder, allows the pressure level in the pump discharge line to be adjusted to the prevailing load of the cylinder so as to limit the pressure loss in the discharge opening of the distributor liquid to the tank. In addition, this system maintains a constant piston speed independent of the load. This is a result of keeping practically constant pressure drop Δp_{DE1} in the throttle slit of the proportional distributor (proportional directional control valve) [15].

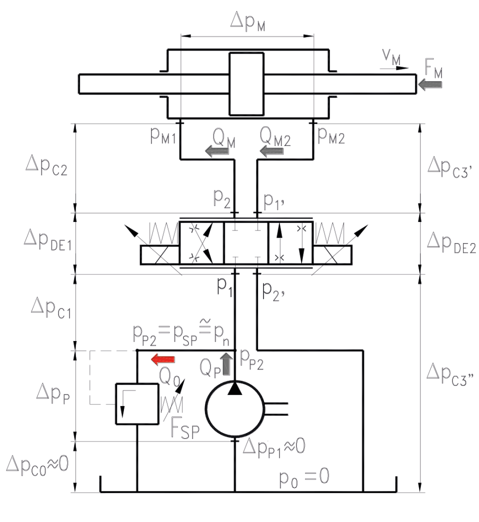

Fig. 1. Diagram of the test system fed at constant pressure – structure p=cte

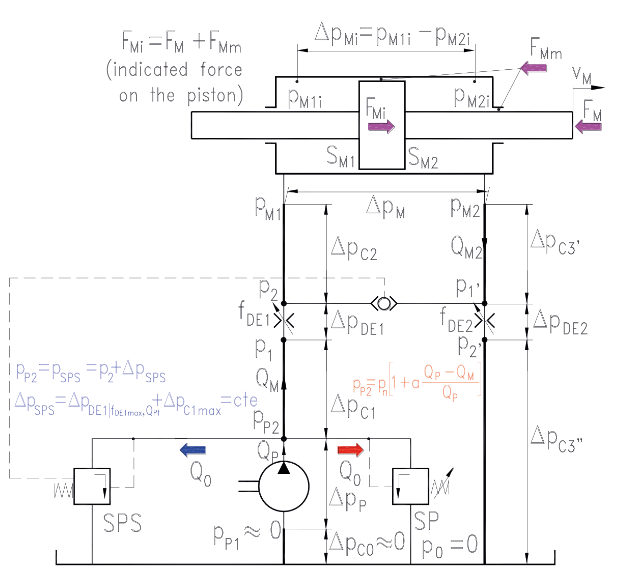

Fig. 2. Schematic diagram of a proportional valve system supplied by a constant capacity pump working with a controlled overflow valve in a variable-pressure system – p=var

The following components were used in the tested systems:

- axial piston pump with displaceable rotor HYDROMATIC type A7.VSO.58DR, operating with fixed theoretical capacity Q_{Pt}=0,000805m^3s^{-1} (48,30dm^3min^{-1}),
- directional proportional control valve, REXROTH type 4WRA10E60-21/G24N9K4, with identical throttling slots $f_{DE1}=f_{DE2}$,
- double piston cylinder HYDROSTER type CD-63/36x500, piston diameter D=63mm and piston rod diameter d=36mm,
- indirect operation overflow valve REXROTH type DBW10A3-52/315XU GE 62 4N9K4,
- controlled overflow valve REXROTH type ZDC10PT-23/XM (only in the variable pressure – p=var system).

The nominal pressure of the tested systems was p_n=16MPa, the hydraulic oil Total Azola 46 was used with kinematics viscosity of υ=35mm^2s^{-1} (at the temperature ϑ=43°C) and volumetric mass of ρ=873,3kgm^{-3}.

The studied structures worked at the same parameters of the linear hydraulic cylinder, i.e. its load F_M and speed v_M.

The considerations allow for comparison of the loss power ΔP of the individual losses resulting from the applied structure supply and the power P_{PC} consumed by the pump from the electric motor that drives it, the power required to provide the unchanged useful power $P_{Mu}=F_M \cdot v_M$ for hydraulic cylinder.

DEFINITIONS OF POWER AND LOSS POWER OCCURRING IN SYSTEM COMPONENTS

Definitions of parameters of work, power and power of particular losses and the record of efficiency are the content of many articles, papers and monographs of prof. Z. Paszota [1÷13]. Model of loss and energy efficiency of a hydraulic drive with proportional control of the cylinder, which would use a full description of loss and efficiency of the hydraulic motor itself, would become too complex. Therefore, pressure losses (flow resistance) in the inlet and outlet conduits of the cylinder are omitted, as well as negligible volume losses (leaks) in the cylinder.

The following describes the powers that characterize the operation of particular elements occurring in the tested hydrostatic drive systems.

The effective power of the cylinder on its piston rod is the product of the external load F_M and velocity v_M of the piston rod travel:

$$P_{Mu} = F_M v_M \quad [7]. \tag{1}$$

The P_{Mc} power consumed by the cylinder is the difference between the power of liquid in the inlet line and the power of liquid in the discharge line of the cylinder:

$$\begin{aligned} P_{Mu} &= P_{M1} - P_{M2} = Q_M p_{M1} - Q_{M2} p_{M2} \\ &= Q_M (p_{M1} - p_{M2}) \quad [7]. \end{aligned} \tag{2}$$

Power ΔP_{Mm} mechanical losses in the cylinder is the product of the velocity v_M of the piston and the force F_{Mm} of friction between the piston and the cylinder on one side and between the piston rod and the gland on the other hand:

$$\begin{aligned} \Delta P_{Mm} &= P_{Mc} - P_{Mu} = Q_M (p_{M1} - p_{M2}) - F_M v_M \\ &= F_{Mm} v_M \quad [7]. \end{aligned} \tag{3}$$

The useful power of the P_{Pu} pump can be defined as the product of its Q_P capacity and pressure increase Δp_P (pressure p_{P2} when $p_{P1} = 0$):

$$\begin{aligned} P_{Pu} &= Q_P \cdot \Delta p_P = Q_P \cdot (p_{P2} - p_{P1}) \\ &= Q_P \cdot p_{P2} \quad [7]. \end{aligned} \tag{4}$$

The P_{Pc} power consumed by the pump on its drive shaft is the product of the moment M_P and the angular velocity ω_P:

$$P_{Pc} = M_P \cdot \omega_P = 2\Pi \cdot M_P \cdot n_P \quad [7]. \tag{5}$$

The power loss ΔP_P of the pump is the sum of the power ΔP_{Pp} of the pressure losses, the power ΔP_{Pv} of the volume losses and the power ΔP_P of the mechanical losses in the pump:

$$\Delta P_P = \Delta P_{Pp} + \Delta P_{Pv} + \Delta P_{Pm} \quad [7]. \tag{6}$$

Power ΔP_{Pv} of pressure losses in the pump:

$$\Delta P_{Pp} = Q_P p_{Pp} = Q_P (\Delta p_{Pp1} + \Delta p_{Pp2}) \quad [7]. \tag{7}$$

Power ΔP_{Pp} of volume losses in the pump:

$$\Delta P_{Pv} = Q_{Pf} \Delta p_{Pi} = Q_{Pf} (p_{P2i} - p_{P1i}) \quad [7]. \tag{8}$$

Power ΔP_{Pm} of mechanical losses in the pump:

$$\Delta P_{Pm} = M_{Pm} \omega_P \quad [7]. \tag{9}$$

Power ΔP_C of pressure losses in conduits:

$$\Delta P_C = Q_M \Delta p_C \quad [7]. \tag{10}$$

DEFINITIONS OF ENERGY EFFICIENCIES

Problems connected with energy efficiency are of basic importance for improvement of functionality and quality of the hydrostatic drive systems, characterized by unquestioned advantages but also by relatively low efficiency in comparison with other types of drives. Publications describing the influence of particular design and operating parameters on the hydrostatic system efficiency are valuable. They make it possible to work out system configurations with losses reduced to a minimum [8].

Energy efficiency of hydrostatic transmissions, particularly those with throttling control of the motor speed, and also efficiency of the hydraulic servo-mechanism systems may be in fact higher than the values most often quoted in publications on the subject. Possibility of calculating the real value of the hydraulic system overall efficiency as a function of many parameters influencing it, becomes a tool of complete evaluation of the designed system quality. The capability of making such an evaluation is important because the hydrostatic control systems are used in various machines and equipment, and also due to increasing power of the hydrostatic drive at the time of constantly increasing costs of energy generation [5].

In a system with too low efficiency, the load of the pump increases, which leads to increased risk of failure and the necessary repair or replacement, as well as to a shorter service life. The too low system efficiency, most often resulting from intensive throttling of the stream of liquid, is also a source of rapid deterioration of operational characteristics, particularly the hydraulic oil lubricating properties, which is an effect, among other reasons, of too high temperature of the working liquid – the hydrostatic transmission power medium.

The energy efficiency of components and systems can be determined by the following formulas:

- mechanical efficiency η_{Pm} of a pump:

$$\eta_{Pm} = \frac{X + k_3(1-k_1)^2(1-k_2)^2}{k_{4.1} + (1+k_{4.2})[X + k_3(1-k_1)^2(1-k_2)^2]^2} \quad [13], \tag{11}$$

where:

$$X = 1 + a[(1-k_1)(1-k_2) - \overline{Q}_M] \quad [13],$$

- pressure efficiency η_{Pp} of a pump:

$$\eta_{Pp} = \frac{1 + a[(1-k_1)(1-k_2) - \overline{Q}_M]}{1 + a[(1-k_1)(1-k_2) - \overline{Q}_M] + k_3(1-k_1)^2(1-k_2)^2} \quad [13], \tag{12}$$

• volume efficiency η_{Pv} of a pump:

$$\eta_{Pv} = 1 - k_1 \quad [13]. \tag{13}$$

• overall efficiency η_P of a pump:

$$\eta_P = \frac{X(1-k_1)}{k_{4.1} + (1+k_{4.2})[X + k_3(1-k_1)^2(1-k_2)^2]} \quad [13], \tag{14}$$

where:

$$X = 1 + a[(1-k_1)(1-k_2) - \overline{Q}_M] \quad [13],$$

• total efficiency η_C of wires:

$$\eta_C = \frac{1 + a[(1-k_1)(1-k_2) - \overline{Q}_M] - k_5\overline{Q}_M}{1 + a[(1-k_1)(1-k_2) - \overline{Q}_M]} \frac{k_{7.1} + (1+k_{7.2})\overline{M}_M}{k_{7.1} + (1+k_{7.2})\overline{M}_M + k_6\overline{Q}_M} \quad [13], \tag{15}$$

• structural pressure efficiency η_{stp}:

$$\eta_{stp} = \frac{k_{7.1} + (1+k_{7.2})\overline{M}_M + k_6\overline{Q}_M}{1 + a[(1-k_1)(1-k_2) - \overline{Q}_M] - k_5\overline{Q}_M} \quad [13], \tag{16}$$

• structural volume efficiency η_{stv}:

$$\eta_{stv} = \frac{\overline{Q}_M}{(1-k_1)(1-k_2)} \quad [13], \tag{17}$$

• structural efficiency η_{st}:

$$\eta_{st} = \eta_{stp}\eta_{stv} = \frac{k_{7.1} + (1+k_{7.2})\overline{M}_M + k_6\overline{Q}_M}{1 + a[(1-k_1)(1-k_2) - \overline{Q}_M] - k_5\overline{Q}_M} \frac{\overline{Q}_M}{(1-k_1)(1-k_2)} \quad [13], \tag{18}$$

• overall efficiency η_M of the cylinder, equal to its mechanical efficiency η_{Mm}:

$$\eta_M = \eta_{Mm} = \frac{\overline{M}_M}{k_{7.1} + \left(1+k_{7.2}\right)\overline{M}_M} \quad [13], \tag{19}$$

• overall efficiency η of the system:

$$\eta = \frac{P_{Mu}}{P_{Pc}} = \eta_P\eta_C\eta_{st}\eta_M = \frac{\overline{Q}_M}{1-k_2} \frac{\overline{M}_M}{k_{4.1} + (1+k_{4.2})[X + k_3(1-k_1)^2(1-k_2)^2]} \quad [13], \tag{20}$$

where:

$$X = 1 + a\left[\left(1-k_1\right)\left(1-k_2\right) - \overline{Q}_M\right] \quad [13], \tag{21}$$

FORMS FOR DETERMINING THE COEFFICIENTS "k_i" AND "a" AT THE REFERENCE VISCOSITY "v"

Knowledge of the energy efficiency course of a positive displacement machine as an independent whole is not enough to assess its energetic behavior in a hydrostatic drive system with a selected structure of hydraulic motor speed control. The energetic behavior of the positive displacement machine in the system results from the conditions created by the applied structure of the speed change of the hydraulic motor shaft, so it is different in different conditions. It is also required to describe it in the form of the function of current parameters of the operating system as well as the function of the speed coefficient $\overline{\omega}_M$ and load coefficient $\overline{M}_M$ of the hydraulic motor in the system.

It is necessary to replace the previously used researches of energy efficiency of displacement machines with researches of energy losses in machines. The results of such researches make it possible to assess the efficiency of the pump or hydraulic motor as an independent whole, as well as to evaluate the efficiency of the machine used in the drive system. Such possibilities are offered by energy loss tests in the pump and in the hydraulic motor combined with the assessment of the coefficients of energy losses occurring in them. These coefficients were called "k_i coefficients".

Knowing the coefficients k_i of energy losses occurring in the hydrostatic system element (in the pump, in the hydraulic motor, but also in the conduits and in the motor speed control group) allows to build mathematical models of losses and energy efficiency of the element working in the system and energy efficiency of the system as a whole composed of elements. Mathematical model of losses and energy efficiency of an element in the system must take into account the conditions resulting from the applied structure of the system, from the level of assumed nominal pressure, from the rotational speed of the engine driving the pump shaft, from the viscosity change of the applied working fluid (hydraulic oil) [11].

The coefficients k_i describe the relative value of individual losses in the element. They allow to assess the proportions of losses and to assess the value of the energy efficiency of the element (volume, pressure, mechanical) resulting from losses occurring at the nominal pressure p_n of the system in which the element is used [4].

As a result, thanks to the knowledge of the coefficients k_i of individual losses, it is possible to determine losses and energy efficiency of components operating in the drive system as well as the efficiency of the total system with a determined structure of the motor speed control as a function of the speed coefficient $\overline{\omega}_M$ and load coefficient $\overline{M}_M$ of the hydraulic motor.

In order to compare losses and energy efficiency of two examined structures: constant pressure system p=cte and variable pressure system p=var, consisting of the cylinder, conduits, proportional directional control valve, overflow valve SP (p=cte and p=var systems) and controlled overflow valve SPS (only p=var system) and pumps, measurements were made using a laboratory computer and using the National Instruments LabView 6.0 program for this purpose. The results of the measurements were processed in Excel program. The computer with measuring transducers was connected using a PCI 1713 Advantech measurement card. In this way, 4 signals from pressure transducers were recorded, the piston rod signal was signaled using a linear displacement transducer, on the basis of which its v_M speed was determined, and the signal of the F_M force loading the piston rod [14].

Each measurement cycle was started from oil heating to the temperature of υ=43°C, at which its kinematic viscosity is $\nu=35mm^2s^{-1}$.

At the beginning, the test cylinder performed reciprocating movements without load. Then, after starting the pump supplying the load cylinder, the load was systematically increased. When the measurement results were repeatable, i.e. stable, researches were started.

Thanks to the automated work of the systems, the hydraulic cylinders moved in both directions without interruption, while during the working movement the load cylinder loaded the tested cylinder. In the return movement, both cylinders moved independently. In this way, there are no air bubbles in the oil filling the cylinder chambers, which would adversely affect the measurement results.

In order to compare two hydraulic systems, the same series of speed v_M and piston rod loads F_M were assumed when testing them. During the measurements, the tested cylinder was loaded with the force of F_M equal to 0kN, 5kN, 10kN, 15kN, 20kN, 25kN, 30kN. On the other hand, the cylinder's piston moved at a speed v_M equal to 0.025m/s, 0.05m/s, 0.075m/s, 0.10m/s, 0.15m/s, 0.20m/s, 0.25m/s, 0.30m/s, 0.35m/s, 0.40m/s [14].

Thanks to the knowledge of the coefficients k_i of individual losses, it is possible to determine the losses and energy efficiency of components operating in the hydraulic system.

The following is a set of definitional formulas allowing to determine the coefficients k_i of energy losses in the elements of the hydraulic system and the value of the "a" coefficient of the overflow valve at the reference viscosity $\upsilon_n=35mm^2s^{-1}$.

A. THE PUMP

- **Coefficients "k_i"**
  - **k_1 – a coefficient of relative volumetric losses Q_{Pv} per one shaft revolution of fixed capacity pump**

$$k_1 = \frac{Q_{Pv|\Delta p_{Pi}=p_n}}{n_{P|\Delta p_{Pi}=p_n}} \frac{1}{q_{Pt}} \quad [12], \qquad (22)$$

Therefore, based on the results obtained during laboratory researches, the coefficient "k_1" is:

$$k_1 = \frac{q_{Pt} - q_{P|\Delta p_{Pi}=p_n}}{q_{Pt}} = \frac{32{,}20 - 30{,}35}{32{,}20} = 0{,}0574 \quad [14], \qquad (23)$$

- **k_2 – a coefficient of relative decrease Δn_P in pump rotational speed (Δn_P speed decrease is related to the rotational speed "n" characteristic as a function of the torque "M" of the driving motor – e.g. electric motor):**

$$k_2 = \frac{n_{P0} - n_{P|\Delta p_{Pi}=p_n}}{n_{P0}} = \frac{1500 - 1494}{1500} = 0{,}004 \quad [14], \qquad (24)$$

- **k_3 – a coefficient of relative pressure losses (flow resistance) in internal pump ducts, at theoretical pump delivery Q_{Pt}:**

$$k_3 = \frac{\Delta p_{Pp|Q_P=Q_{Pt}}}{p_n} \quad [9]. \qquad (25)$$

According to the above formula, the coefficient k_3 of the tested pump is:

$$k_3 = \frac{0{,}38}{160} = 0{,}0024 \quad [14]. \qquad (26)$$

- **$k_{4.1}$ – a coefficient of relative mechanical losses $M_{Pm\,|\,\Delta p_{Pi}=0}$ in the unloaded constant capacity pump, at $\Delta p_{Pi}=0$ (so at $M_{Pi}=0$) to the nominal moment M_{Pn}:**

$$k_{4.1} = \frac{2\Pi M_{Pm|\Delta p_{Pi}=0}}{q_{Pt}\, p_n} \quad [9]. \qquad (27)$$

According to the above formula, the coefficient $k_{4.1}$ of the tested pump is:

$$k_{4.1} = \frac{3{,}21}{81{,}99} = 0{,}0391 \quad [14]. \qquad (28)$$

- **$k_{4.2}$ – a coefficient of relative increase of mechanical pump losses $\Delta M_{Pm\,|\,\Delta p_{Pi}=p_n}$, at increase in pressure in pump working chambers:**

$$k_{4.2} = \frac{\Delta M_{Pm|\Delta p_{Pi}=p_n}}{M_{Pn}} = \frac{2\Pi \Delta M_{Pm|\Delta p_{Pi}=p_n}}{q_{Pt}\, p_n} \quad [9]. \qquad (29)$$

According to the above formula, the coefficient $k_{4.2}$ of the tested pump type A7.VS0.58.DR is:

$$k_{4.2} = \frac{2\Pi \cdot 1{,}23}{32{,}20 \cdot 10^{-6} \cdot 160 \cdot 10^{5}} = 0{,}015 \quad [14]. \qquad (30)$$

- **Characteristics of basic pump types – approximate values of coefficients „k_i"**

It is proposed to use, in the industrial energy calculations, approximate values of coefficients "k_i" of basic types of pumps.

For example, the approximate values of the "k_i" coefficients of axial piston pumps are shown below (Tab. 1):

Tab. 1. The approximate values of the "k_i" coefficients of axial piston pumps [1]

p_n=10 MPa	p_n=25 MPa
k_1 = 0,01	k_1 = 0,02
k_3 = 0,01	k_3 = 0,01
$k_{4.1}$ = 0,01	$k_{4.1}$ = 0,05
$k_{4.2}$ = 0,01	$k_{4.2}$ = 0,02

B. THE CONDUIT CONNECTING THE PUMP TO THE THROTTLE CONTROL UNIT

- **k_5 – a coefficient of relative pressure losses (flow resistances) Δp_{C1} in the line joining the pump with throttle control unit, at theoretical pump delivery Q_{Pt}:**

$$k_5 = \frac{\Delta p_{C1|Q_{Pt}}}{p_n} \quad [9]. \tag{31}$$

Thus, the value of the coefficient k_5 is:

$$k_5 = \frac{3{,}46}{160} = 0{,}0216 \quad [14]. \tag{32}$$

The coefficient k_5 represents the sum of the flow resistance (pressure loss) in the conduit and in the elements installed on this conduit (filters, check valves, connectors ...).

C. THE CONDUIT CONNECTING THE THROTTLE CONTROL UNIT WITH THE HYDRAULIC CYLINDER

- **$k_{6.1}$ – a coefficient of relative pressure losses (flow resistances) Δp_{C2} in the line joining the throttle control unit with hydraulic motor, at theoretical pump delivery Q_{Pt}:**

$$k_{6.1} = \frac{\Delta p_{C2|Q_{Pt}}}{p_n} \quad [9]. \tag{33}$$

Thus, the value of the $k_{6.1}$ coefficient is:

$$k_{6.1} = \frac{2{,}64}{160} = 0{,}0165 \quad [14]. \tag{34}$$

The coefficient $k_{6.1}$ represents, as above, the sum of the flow resistance (pressure loss) in the conduit and in the various elements that the conduit contains.

D. THE OUTLET LINE FROM THE HYDRAULIC CYLINDER

- **$k_{6.2}$ - a coefficient of relative pressure losses (flow resistances) Δp_{C3} in hydraulic motor outlet line, at theoretical pump delivery Q_{Pt}:**

$$k_{6.2} = \frac{\Delta p_{C3|Q_{Pt}}}{p_n} \quad [9]. \tag{35}$$

Thus, the value of the coefficient $k_{6.2}$ is:

$$k_{6.2} = \frac{2{,}75}{160} = 0{,}0172 \quad [14]. \tag{36}$$

The coefficient $k_{6.2}$ includes, as before, the sum of flow resistance (pressure loss) in the conduit and in the components installed on it.

E. A HYDRAULIC ROTATIONAL MOTOR

- **Coefficients "k_i"**
- **$k_{7.1}$ – a coefficient of mechanical losses $M_{Mm\,|\,M_M=0}$ in an unloaded hydraulic rotational motor with constant absorbency per revolution:**

$$k_{7.1} = \frac{M_{Mm|M_M=0}}{M_{Mn}} = \frac{2\,\Pi M_{Mm|M_M=0}}{q_{Mt}\,p_n} \quad [10]. \tag{37}$$

- **$k_{7.2}$ – a coefficient of mechanical loss increase $M_{Mm\,|\,M_M=M_{Mn}}$ in a hydraulic rotational motor with constant absorbency per revolution:**

$$k_{7.2} = \frac{M\Delta_{Mm|M_{Mi}=M_{Mn}}}{M_{Mn}} = \frac{2\Pi\Delta M_{Mm|M_M=M_{Mn}}}{q_{Mt}\,p_n} \quad [10]. \tag{38}$$

- **k_8 – a coefficient of pressure losses Δp_{Mp} in internal channels (and in the distributor, if any) of a hydraulic rotational motor:**

$$k_8 = \frac{\Delta p_{Mp|Q_M=Q_{Pt}}}{p_n} \quad [10]. \tag{39}$$

- **k_9 – a coefficient of volume losses Q_{Mf} in a hydraulic rotational motor:**

$$k_9 = \frac{Q_{Mf|\Delta p_{Mi}=p_n}}{Q_{Pt}} \quad [10]. \tag{40}$$

- **Characteristics of basic types of hydraulic rotational motors – approximate values of coefficients "k_i"**

As in the case of the pumps, it is proposed to introduce, for the energy calculations of hydraulic systems, approximate values of "k_i" coefficients of losses in basic types of motors. For example, the approximate values of the coefficients "k_i" of low-speed radial piston motors (with an internal cam – eccentric) are presented below (Tab. 2):

Tab. 1. The approximate values of the coefficients "k_i" of low-speed radial piston motors [1]

p_n =10 MPa	p_n =25 MPa
$k_{7.1}$ =0.04	$k_{7.1}$ =0.02
$k_{7.2}$ =0.02	$k_{7.2}$ =0.02
k_8 =0.02	k_8 =0.01
k_9 =0.02	k_9 =0.03

F. THE LINEAR HYDRAULIC MOTOR – CYLINDER

A measurement of friction forces F_{Mm} in the tested cylinder, operating at a constant speed, were made using the indirect method, as there is no device enabling the measurement to be carried out using the direct method. This made it necessary to use high quality transducers for measuring force and pressure.

- **Coefficients "k_i"**
  - **$k_{7.1}$ – a coefficient of mechanical losses in the unloaded cylinder:**

$$k_{7.1} = \frac{F_{Mm|F_M=0}}{F_{Mn}} = \frac{F_{Mm|F_M=0}}{S_{M1}\,p_n} \quad [13]. \qquad (41)$$

  - **$k_{7.2}$ – a coefficient of mechanical loss increase in the cylinder:**

$$k_{7.2} = \frac{F\Delta_{Mm|F_M=F_{Mn}}}{F_M} = \frac{F\Delta_{Mm|F_M=F_{Mn}}}{S_{M1}\,p_n} [13]. \qquad (42)$$

In the case of a cylinder, the coefficient $k_{7.2}$ can be positive or negative; the latter case may result from the application of a throttle control at the outflow from the cylinder (an example of controlling the cylinder with a proportional directional control valve or servo valve).

  - **k_8 – a coefficient of pressure loss (flow resistance) in the internal channels of the cylinder:**

$$k_8 = 0. \qquad (43)$$

The picture of the loss and energy efficiency of the linear hydraulic motor – the cylinder – is more complicated compared to the case of a rotary hydraulic motor.

It was decided, therefore, to consider the energy efficiency problem of systems with the cylinder only for the case of the cylinder with pressure losses (flow resistance) Δp_{Mp} in the inlet and outlet channel, as well as with internal leakages Q_{Mf} negligible.

This is the case of the majority of cylinders, in which the pressure and volumetric energy losses are to be neglected in comparison with mechanical losses, and in particular in comparison with the power developed by the cylinder.

  - **k_9 – a coefficient of volume losses Q_{Mf} in the cylinder:**

$$k_9 = 0. \qquad (44)$$

- **Characteristics of selected varieties and sizes of hydraulic cylinders with lip sealing rings – approximate values of coefficients $k_{7.1}$ and $k_{7.2}$ of mechanical losses in the cylinder (Tab. 3)**

Conditions:
• a pressure p_{M2} in the cylinder outlet near the zero – $p_{M2}\approx 0$,
• the piston rod is not loaded with transversal forces.
Variations of cylinders:
K – double-acting, with one-sided piston rod,
L – double-acting, with one-sided piston rod, with differential connection,
R – double-acting, with double-sided piston rod.

H. THE PROPORTIONAL DIRECTIONAL CONTROL VALVE

  - **k_{11} – a coefficient of relative pressure decrease Δp_{DE} in directional control valve (servovalve, proportional valve) demanded by a maximum throttling section f_{DEmax} for receiving flow intensity equal theoretical pump delivery Q_{Pt}:**

$$k_{11} = \frac{p\Delta_{DE|f_{DEmax},Q_{Pt}}}{p_n} = \frac{\left(p\Delta_{DE1} + p\Delta_{DE2}\right)_{|f_{DEmax},Q_{Pt}}}{p_n} \quad [13]. \qquad (45)$$

The coefficient k_{11} corresponds to the sum of the pressure drops $\Delta p_{DE}=\Delta p_{DE1}+\Delta p_{DE2}$ in the two slots of the throttle distributor. This coefficient for the proportional directional control valve type 4WRA 10 E60-21 / G24 N9K4 can be determined using equation (45). So, in this case:

$$k_{11} = \frac{0{,}592\,\text{MPa} + 0{,}592\ \text{MPa}}{16\,\text{MPa}} = \frac{1{,}184\,\text{MPa}}{16\,\text{MPa}} = 0{,}074 \quad [14]. \qquad (46)$$

The choice of the size of the throttling valve is made in such a way that its nominal flow rate, while the hydraulic system is working, corresponds more or less to the maximum electric current (close to 100%). The application of this principle

Tab. 1. The approximate values of coefficients $k_{7.1}$ and $k_{7.2}$ of mechanical losses in the cylinder[4]

Push or pull movement (example):

Pulling motion									
D diameter of the piston [mm]	ρ – ratio of active surfaces of the piston	type K				type R			
		p_n = 10 MPa		p_n = 25 MPa		p_n = 10 MPa		p_n = 25 MPa	
		$k_{7.1}$	$k_{7.2}$	$k_{7.1}$	$k_{7.2}$	$k_{7.1}$	$k_{7.2}$	$k_{7.1}$	$k_{7.2}$
32÷50	1,4	0.05	0.07	0.03	0.05	0.06	0.07	0.03	0.06
	2	0.11	0.10	0.05	0.07	0.14	0.10	0.06	0.08
125÷200	1,4	0.04	0.05	0.02	0.03	0.04	0.05	0.02	0.03
	2	0.05	0.06	0.02	0.04	0.08	0.03	0.03	0.05
400÷630	1,4	0.02	0.02	0.01	0.02	0.02	0.02	0.01	0.02
	2	0.02	0.04	0.01	0.03	0.03	0.03	0.01	0.03

makes it possible to optimally use the size controlled by the throttling valve and their mild change during control.

If the nominal intensity of the throttling valve used is equal to the theoretical efficiency of the pump operating in the system, $Q_{DEn} = Q_{Pt}$, then a specific numerical value of the k_{11} coefficient can be determined with the selected nominal pressure p_n of the system. For example, the size of the servo valve coefficient k_{11}, characterized by a nominal pressure drop $\Delta p_{DEn} = 7MPa$ and operating in a system with a nominal pressure $p_n = 20MPa$, will be equal to:

$$k_{11} = \frac{p\Delta_{DE|f_{DEmax},Q_{Pt}}}{p_n} = \frac{7MPa}{20MPa} = 0.35 \text{ [9].} \qquad \textbf{(47)}$$

As approximate values of the coefficient k_{11}, we can cite [4]: $k_{11} = 0.05 \div 0.50$ in the case of a servo valve, $k_{11} = 0.03 \div 0.06$ in the case of a proportional directional control valve.

J. AN OVERFLOW VALVE

A coefficient "a" of the pressure increase p_{SP} stabilized by an overflow valve (pressure limiter) in the constant capacity pump:

$$a = \frac{p_{SP|Q_0=Q_{Pt}} - p_{SP0}}{p_{SP0}} = \frac{p_{SP|Q_0=Q_{Pt}} - p_n}{p_n} \text{ [9],} \qquad \textbf{(48)}$$

where:
$P_{SP|Q_0 = Q_{Pt}}$ – a pressure stabilized by the overflow valve at the flow rate Q_0 in the valve equal to theoretical pump capacity Q_{Pt}– $Q_0 = Q_{Pt}$, p_{SP0} [Pa] – the opening pressure of the overflow valve equal to the nominal pressure p_n of the system – $p_{SP0} = p_n$, obtained at the flow rate Q_0 in the valve equal to 0 – $Q_0 = 0$. Range of change of the coefficient "a":
$a = 0 \div 0.20$.

In order to determine the coefficient "a" of the pressure increase controlled by the overflow valve, installed on a test stand, researches were carried out. Thus, the coefficient "a" of the tested overflow valve type DBW 10 A3 – 52 \ 315XU 6E G2 4N9K4 from Rexroth is:

$$a = \frac{p_{SP|Q_0=Q_{Pt}} - p_{SP0}}{p_{SP0}} = \frac{16{,}38 - 16{,}0}{16} = 0{,}023 \text{ [14].} \qquad \textbf{(49)}$$

EXAMPLES OF LABORATORY VERIFICATION OF ENERGY SAVING IN THE SYSTEM WITH A CONSTANT CAPACITY PUMP IN A VARIABLE PRESSURE SYSTEM p=var, IN COMPARISON WITH A CONSTANT PRESSURE SYSTEM p=cte

The test results, shown in Figures 3–9, allow for a comparison of the energy loss power values, expressed in Watts [W], in the p=cte and p=var system elements.

The most significant reduction of energy losses, when p=cte system is replaced by a p=var system, is obtained in the case of the structural pressure loss power ΔP_{stp} (Fig. 3) in the proportional directional control valve. With the cylinder load coefficient $\overline{M}_M = 0$ and speed coefficient $\overline{\omega}_M = 0{,}875$, the loss power is reduced from approx. 9800 W to approx. 1800W, i.e. by 7,5times. The pressure loss power ΔP_{stp} in both systems equalizes in the maximum cylinder load area (maximum $\overline{M}_M$ values), i.e. in the area where the p=var system begins to operate as a p=cte system. The pressure structural loss power ΔP_{stp} in both systems is then relatively small – below 2300W.

The volumetric structural loss power ΔP_{stv} (Fig. 4), occurring in the overflow valve (p=cte system) or in the controlled overflow valve and the overflow valve (p=var system), decreases also when the p=cte system is replaced by the p=var system. However, the power reduction is not as significant as in the case of pressure structural loss power ΔP_{stp}.

With the cylinder coefficient $\overline{M}_M = 0$ and speed coefficient $\overline{\omega}_M = 0{,}063$, the volumetric loss power ΔP_{stv} is reduced from ca. 12000W to approx. 2400W, ie. 5 fold. The volumetric loss power ΔP_{stv} in both systems equalizes in the cylinder maximum load area (maximum $\overline{M}_M$ values, i.e. in the area of the p=var system operating as a p=cte system). However, the same volumetric loss power ΔP_{stv} in both systems is at its maximum – it reaches 12000W at $\overline{\omega}_M = 0{,}063$ [15].

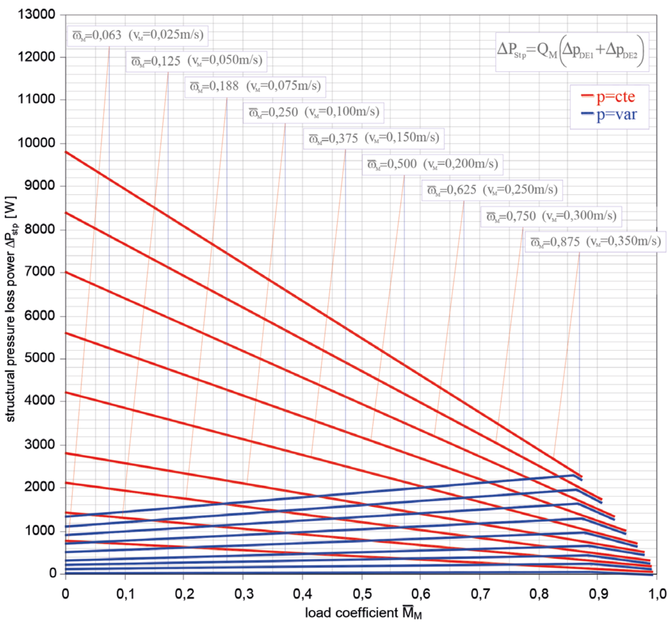

Fig. 3. Relation between the structural pressure loss power ΔP_{stp} (in the directional proportional control valve), in the constant pressure system (p=cte) and variable pressure system (p=var), and the load coefficient $\overline{M}_M$ with different speed coefficients $\overline{\omega}_M$ of the cylinder [15]

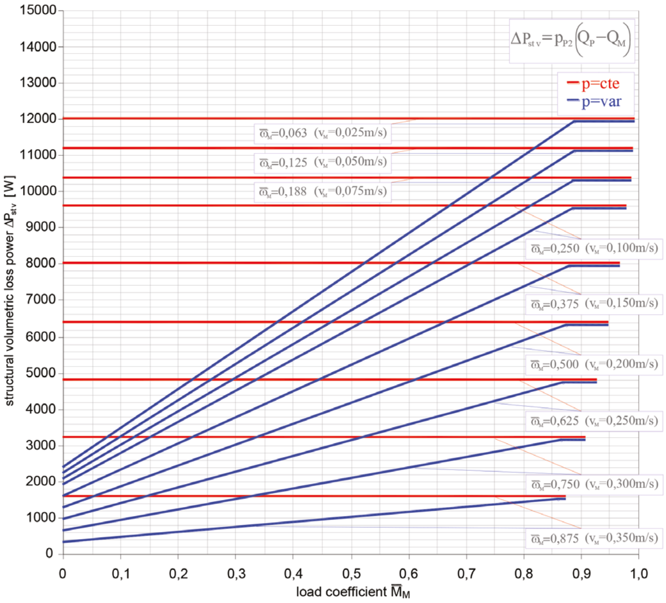

Fig. 4. Relation between the structural volumetric loss power ΔP_{stv} (in the overflow valve and in controlled overflow valve), in the constant pressure system (p=cte) and variable pressure system (p=var), and the load coefficient $\overline{M}_M$ with different speed coefficients $\overline{\omega}_M$ of the cylinder [15]

The structural loss power ΔP_{st} is the sum of the structural pressure loss power ΔP_{stp} in the proportional distributor (proportional directional control valve) and the structural volume loss power ΔP_{stv} in the overflow valve or in the control overflow valve:

$$\Delta P_{st} = \Delta P_{stp} + \Delta P_{stv} \tag{50}$$

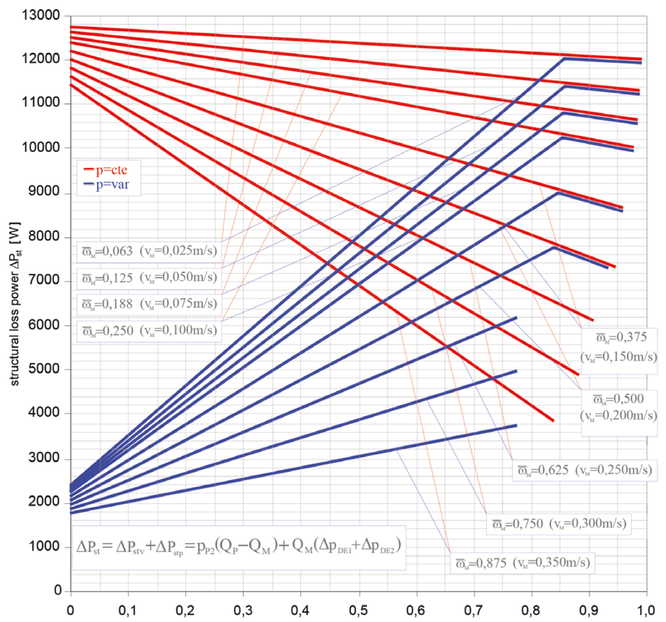

Fig. 5. Relation between the structural loss power ΔP_{st} in the throttle control unit (sum of structural pressure loss power ΔP_{stp} in the proportional directional control valve and the structural volume loss power ΔP_{stv} in the overflow valve and in the control overflow valve) in constant pressure system (p=cte) and variable pressure system (p=var) from the load coefficient $\overline{M}_M$ at different speed coefficients $\overline{\omega}_M$ in hydraulic cylinder

Fig. 5 shows the diagram of structural loss power ΔP_{st} in constant pressure system (p=cte) and variable pressure system (p=var).

The structural loss power ΔP_{st} in the p=cte system, with the determined values of the speed coefficient $\overline{\omega}_M$ of the cylinder decreases, as the load coefficient $\overline{M}_M$ increases.

With the load coefficient $\overline{M}_M$=0 and the speed coefficient $\overline{\omega}_M$=0,063 (v_M=0,025m/s) of the cylinder, the loss power ΔP_{st} of the constant pressure system p=cte achieves the greatest value of ΔP_{st} =12700W. At the same speed value and with a maximum load coefficient of $\overline{M}_M$=0,988, the structural loss power ΔP_{st} in the p=cte system drops to ΔP_{st}=12000W. On the other hand, with the maximum values of speed and load of the cylinder, ΔP_{st} of the system p=cte assumes the smallest value equal to ΔP_{st} = 3815W. This 3,3 times decrease of ΔP_{st} is mainly related to the decreasing pressure drop Δp_{DE} in the proportional directional control valve and to the decreasing flow intensity Q_0 facing to the reservoir through the overflow valve [14].

After replacing the constant pressure system p=cte with the variable pressure system p=var, the structural loss power ΔP_{st} is noticeable. This is due to the reduced pressure p_{P2} in the pump discharge line at lower load coefficients $\overline{M}_M$ of the cylinder.

With a load coefficient $\overline{M}_M$=0 and a speed coefficient $\overline{\omega}_M$=0,063 (v_M=0,025m/s) of the cylinder, the structural loss power decreases from ΔP_{st}=12700W (p=cte) to about ΔP_{st}=2400W (p=var) and therefore 5,3 times. The structural loss power ΔP_{st} in both systems equate in the zone of maximum cylinder load (maximum values), ie in the zone where system p=var works as p=cte. Then the structural loss power ΔP_{st} in both systems, at a minimum speed coefficient of $\overline{\omega}_M$=0,063, is high and is ΔP_{st}=12000W.

In the p=var system, when the hydraulic cylinder is operating at a high speed coefficient of $\overline{\omega}_M$=0,875 (v_M=0,350m/s), the structural loss power ΔP_{st} decreases markedly, changing from ΔP_{st}=1780W at $\overline{M}_M$=0 to ΔP_{st}=3800W at $\overline{M}_M$=0,775 [15].

In summary, the advantage of replacing the constant pressure structure p=cte with the p=var structure is most evident in the representation of the structural loss power ΔP_{st} in the studied systems in the aggregate diagram of these losses (Fig. 5). It follows that ΔP_{st} of the p=cte structure decreases both with increasing speed and with increasing load of the cylinder. In p=var system, ΔP_{st} increases with increasing load, and decreases with increasing speed of hydraulic cylinder.

RELATION BETWEEN LOSS POWER IN HYDRAULIC COMPONENTS AND POWER REQUIRED BY THE CONSTANT CAPACITY PUMP FROM USEFUL POWER OF CYLINDER IN p=cte AND p=var STRUCTURES

Test results shown in Fig. 6 allow for comparison depending on the amount of the loss power ΔP (expressed in watts [W]) occurring in the elements and the consumed power P_{Pc} by

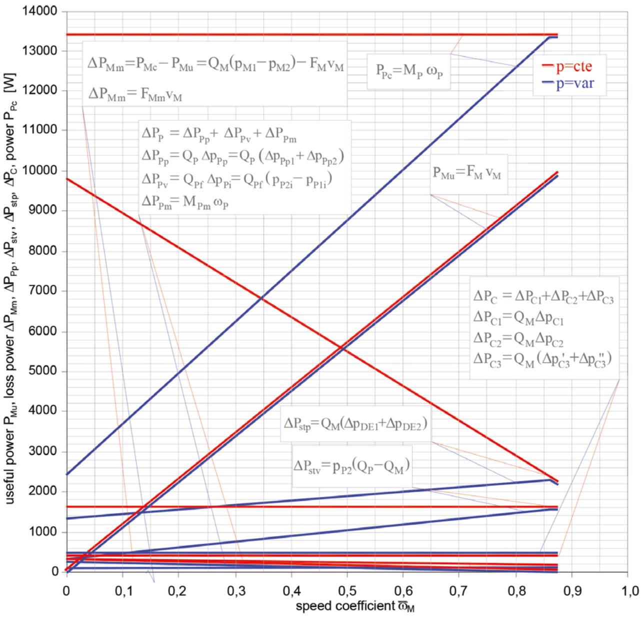

Fig. 6. The loss power ΔP of the system components and the power P_{Pc} demanded by the pump in the constant pressure system (p=cte) and variable pressure system (p=var) from the load coefficient $\overline{M}_M$ at the hydraulic cylinder speed coefficient $\overline{\omega}_M$=0,875 (v_M=0,35m/s); The useful power P_{Mu} of the hydraulic cylinder is resulted from the product of the current load F_M ($\overline{M}_M$) and the actual speed v_M ($\overline{\omega}_M$) of the cylinder required by the driven device [14]

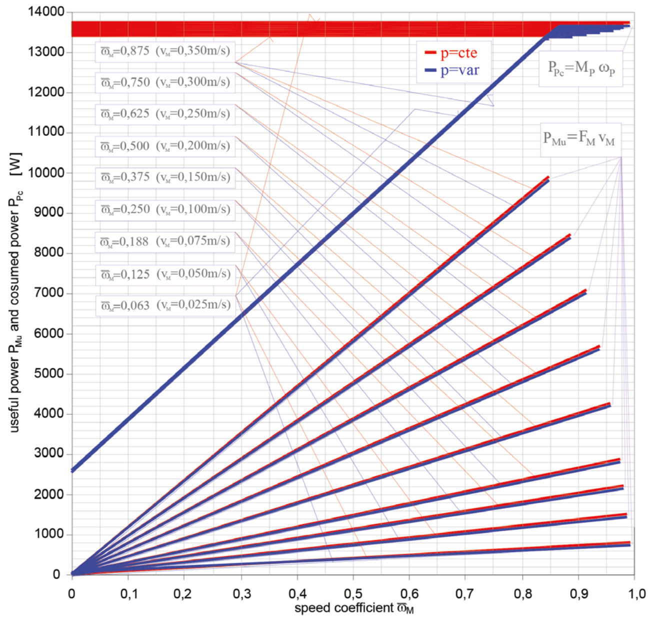

Fig. 7. Relation of power P_{Pc} demanded by the pump in the constant pressure system (p=cte) and variable pressure system (p=var) from the load coefficient $\overline{M}_M$ at the different speed coefficient $\overline{\omega}_M$ (So this is the dependence the consumed power P_{Pc} from the useful power P_{Mu} of the hydraulic cylinder) [14]

the pump from the useful power P_{Mu} of cylinder controlled in a constant pressure system p=cte and variable pressure system p=var at the speed coefficient of the cylinder $\overline{\omega}_M$=0,875 (v_M=0,350m/s).

The graph in Fig. 6 shows that the charts of the consumed power P_{Pc} of the pump (at the same useful power P_{Mu} of the cylinder) are different for the two investigated systems. In the constant pressure system, the consumed power P_{Pc} is constant throughout range of change of the load coefficient and is 13380W. On the other hand, in the case of a variable pressure system, the power P_{Pc} varies, depending on the load of the cylinder, in the range of 3200W at $\overline{M}_M$=0 to 13380W at $\overline{M}_M$=0,875. The useful power P_{Mu} of the cylinder increases over the whole load coefficient range, is equal to zero at $\overline{M}_M$=0 and 9900W at $\overline{M}_M$=0,875.

Fig. 7 shows the dependence of the power P_{Pc} demanded by the pump from the output useful power P_{Mu} in the constant pressure system (p=cte) and variable pressure system (p=var). The power P_{Pc} required by the pump and the power P_{Mu} of the cylinder are shown here as a function of the load coefficient $\overline{M}_M$ at different cylinder speed coefficients $\overline{\omega}_M$.

At the smallest speed v_M of the cylinder (v_M=0,025m/s), the power P_{Pc} required by the pump is greatest in the constant pressure system p=cte. This is related to the operation of the overflow valve SP. With increasing speed v_M of the cylinder, the pressure p_{P2} decreases as the overflow valve sets the lower pressure p_{SP}. Consequently, the power P_{Pc} required by the pump decreases.

On the other hand, the useful power of the cylinder, which is the product of the speed v_M of the cylinder and its force loading F_M, is independent of the system. The speed and load of the cylinder are independent of the control structure. Consequently, all loss power ΔP that occur in p=cte and p=var systems are a function of the useful power P_{Mu} and the quality of these components (ie loss power in these components). The loss power ΔP, on the other hand, depends on the current useful power P_{Mu} and above all on the current load F_M and the current speed v_M of the cylinder.

The useful power must be supplied by the investigated systems with the same load F_M and speed v_M and is the same. The useful power P_{MU} will increase as load and speed increase.

The power P_{Pc} demanded by the pump results from the useful power P_{Mu} of the cylinder and all the loss power ΔP occurring in the system.

In conclusion, the power P_{Pc} demanded by the pump depends on the useful power P_{Mu}, the structure of the circuit, and the loss power ΔP that are present in the system components.

COMPARISON OF THE ENERGY EFFICIENCY OF VARIOUS SYSTEM VERSIONS

Investigations of the efficiency of elements and systems, taking into account detail analysis of the sources of particular energy losses, may be included in the basic scope of research into the hydrostatic drive and control systems.

The energy efficiency, one of the most important system characteristics, is defined as the ratio of, by the driven device currently demanded, useful power P_{Mu} of the hydraulic motor to the power P_{Pc}, corresponding to P_{Mu}, obtained by the pump on its shaft from the (electric or combustion) motor driving the pump. In case of improper choice of the system type, the

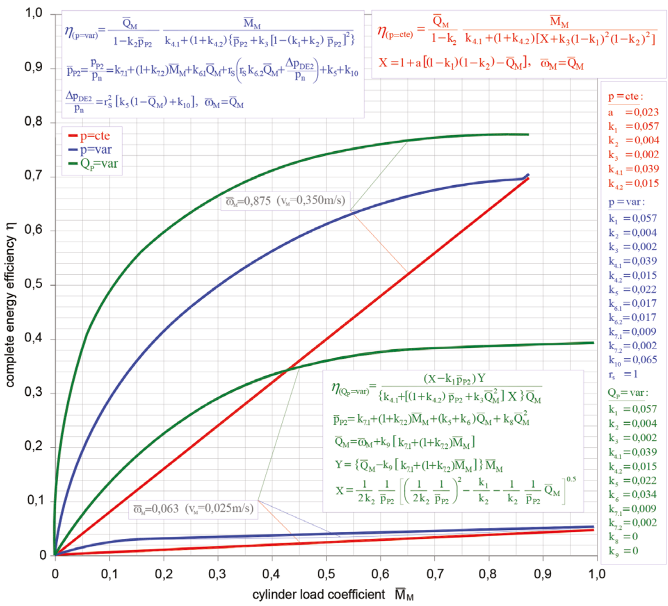

Fig. 8. Relation of the overall efficiency η of a constant pressure (p=cte) and a variable pressure (p=var) system with proportional control and a system with volumetric control by a variable capacity pump (Q_P=var) to the load coefficient $\overline{M}_M$ at different values of the cylinder speed coefficient $\overline{\omega}_M$ (efficiency determined by simulation from experimentally obtained k_i coefficient ; the v_M = 0,350m/s ($\overline{\omega}_M$=0,875) speed was the highest cylinder speed used during the tests) [15]

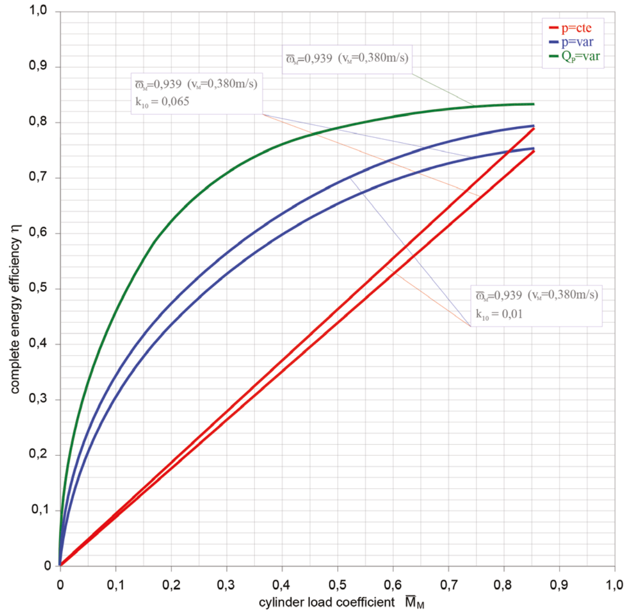

Fig. 9. Relation of the overall efficiency η of a constant pressure (p=cte) and a variable pressure (p=var) system with proportional control, with the used proportional directional valve coefficient k_{10}=0,065 and with possible use of a bigger proportional directional valve with k_{10}=0,010, as well as a system with volumetric control by a variable capacity pump (Q_P=var,), to the load coefficient $\overline{M}_M$ at the value of the cylinder speed coefficient $\overline{\omega}_M$=0,939 (v_M = 0,380m/s) resulting from the maximum pump capacity Q_{Pmax} . Maximum η_{max} values of the three considered systems are closer to one another [15]

consequence may be increased hydraulic oil temperature, i.e. decreased oil viscosity and, in turn, lower efficiency of the system elements, and also an impact on the system run characteristics. Therefore, the energy efficiency may be a decisive factor for usability of a system in a specific case. Its detailed analysis quite often leads to design improvements of the system elements. However, improving the quality of hydrostatic systems does not consist exclusively in the improvements of their elements [10].

Fig. 8 and 9 present the overall efficiency η of a constant pressure (p=cte) and a variable pressure (p=var) system with proportional control and a system with volumetric control by a variable capacity pump (Q_P=var) as a function of the load coefficient $\overline{M}_M$ at different values of the cylinder speed coefficient $\overline{\omega}_M$.

In the case of a system with volumetric control by a variable capacity pump (Q_P=var), increasing the cylinder load coefficient $\overline{M}_M$ causes rapid increase of the system overall efficiency η (Fig. 8). However, efficiency of structures with the series throttling control fed by a constant capacity pump is, with small value of the $\overline{\omega}_M$ coefficient, distinctly lower than the volumetric control efficiency with the same value of $\overline{\omega}_M$, because the structural losses are high [15].

Increase of the cylinder speed causes a proportional increase of efficiency of the p=cte and p=var systems, but with an increase of the cylinder speed v_M the relative increase of efficiency of the system fed by a variable capacity pump is smaller (Fig. 8).

It can be seen in Fig. 8 that a 14-fold increase of the cylinder speed in the investigated structures causes about 14-fold increase of their efficiency. As a comparison, a 14-fold increase of the cylinder speed in a Q_P=var structure causes about 2-fold increase of its efficiency (from η=0,39 at $\overline{\omega}_M$=0,063 and $\overline{M}_M$=0,875 to η=0,78 at $\overline{\omega}_M$=0,875 and $\overline{M}_M$=0,875).

Fig. 9 presents efficiency η of a constant pressure (p=cte) and a variable pressure (p=var) system with proportional control, with the used proportional directional valve coefficient k_{10}=0,065 and with possible use of a bigger proportional directional valve with k_{10}=0,010, as well as a system with volumetric control by a variable capacity pump (Q_P=var), as a function of the load coefficient $\overline{M}_M$ at the value of the cylinder speed coefficient $\overline{\omega}_M$=0,939 (v_M = 0,380m/s) resulting from the maximum pump capacity Q_{Pmax} [15].

In the maximum cylinder speed range, i.e. with the full use of the pump capacity, efficiency values of the p=cte and p=var systems with throttling control become close to the efficiency of the Q_P=var system with volumetric control.

SUMMARY AND CONCLUSIONS

There are possibilities to reduce energy losses in proportional control systems (in the pump, in the throttle control unit, especially in the cylinder), and thus to improve the energy efficiency of the throttling manifold. The considerations allow for comparison of the loss power resulting from the applied hydraulic control structure of the hydraulic cylinder and the power consumed by the pump from the electric motor that drives it, the power necessary to provide pump-driven hydraulic cylinder. Presents the impact on the output (useful) power consumed in the considered systems, and the impact on the power consumed of the loss power in the individual elements. Instantaneous useful power of the cylinder, which is determined by the product of force and speed of the cylinder rod is independent of all losses. There are mechanical loss power in the cylinder, loss power in the conduits, the structural volume and pressure loss power that are associated with the throttling control and loss power in the pump: pressure, volumetric and mechanical which have to be added to the useful power. As a result, the sum of the effective useful power and the loss power of all system is the instantaneous value of the power consumed by the pump from the electric motor that drives it.

This article presents a comparison of the loss power of the two systems – p=cte and p=var. Diagrams show how the power lines P_{Mu} of the cylinder are running, the power lines ΔP of the loss power in the components and the power line P_{Pc} taken by the pump from the motor that drives it. The energy gains associated with the introduction of a variable pressure p=var compared to the p=cte pressure system are also presented.

The influence of power P_{Mu} on the power P_{Pc} in the systems under consideration as well as the influence on the P_{Pc} of the loss power ΔP on the individual components are presented. The P_{Mu} momentary power of the cylinder, which is determined by the product of the force F_M and the speed v_M of the cylinder rod, is independent of all losses. For the useful power P_{Mu} comes the mechanical loss power ΔP_{Mm} in the cylinder, the loss power ΔP_C in the conduits, the structural volume loss power ΔP_{stv}, and the structural pressure loss power ΔP_{stp} associated with the throttling control and the losses in the pump: pressure loss power ΔP_{Pp}, volume loss power ΔP_{Pv} and mechanical loss power ΔP_{Pm}. As a result of the sum of the useful power P_{Mu} and all loss power ΔP in the system, the instantaneous power P_{Pc} value that the pump requires from the electric motor driving it is obtained. Changing the structure from p=cte to p=var, with the same useful power P_{Mu}, results in a significant decrease in structural loss power ΔP_{st} (Fig. 5).

The maximum achievable values of efficiency of systems with proportional (i.e. series throttling) control and of a system with volumetric control by a variable capacity pump are approximately similar. The compared systems were assembled from elements with the same k_i coefficients of energy losses.

By applying a variable pressure (p=var) system, a significant increase of the energy efficiency η can be achieved with smaller cylinder loads.

With small cylinder speed values, the effect of using a p=var system is little, mainly due to volumetric losses connected with draining the excess liquid to the tank.

Optimization of hydrostatic systems means, among other aspects, a possibility of foreseeing the behaviour of an energy system in various conditions of its operation, as a function of speed and load of the hydraulic motor, working liquid viscosity, losses in the elements and particularly as an effect of the system structure. The common acceptance and use of the objective, experimentally verified methods of determining the system energy efficiency, looking at the efficiency of entire combined system, can clarify many misunderstandings, e.g. those pertaining to the problem of maximum efficiency of specific structures.

The Author intends to carry out further investigations of systems with proportional control, aimed at determining the influence of the working liquid (oil) viscosity on the energy efficiency.

NOMENCLATURE

a – coefficient of pressure increase in the overflow valve or in the controlled overflow valve
cte – constant
f_{DE1} – throttling slot at the cylinder inlet
f_{DE2} – throttling slot at the cylinder outlet
F_M – hydraulic linear motor (cylinder) load, current force required of a linear motor
F_{Mi} – force indicated on the piston of the hydraulic linear motor (cylinder)
F_{Mm} – hydraulic linear motor mechanical losses
F_{SP} – force of spring in the overflow valve
k_1 – coefficient of relative volumetric losses per one shaft revolution of fixed capacity pump
k_2 – coefficient of relative decrease in pump rotational speed
k_3 – coefficient of relative pressure losses (flow resistance) in internal pump ducts, at theoretical pump delivery Q_{Pt}
$k_{4.1}$ – coefficient of relative mechanical losses in pump, at $\Delta p_{Pi}=0$
$k_{4.2}$ – coefficient of relative increase of mechanical pump losses, at increase in pressure in pump working chambers
k_5 – coefficient of relative pressure losses (flow resistances) in the line joining the pump with throttle control unit, at theoretical pump delivery Q_{Pt}
$k_{6.1}$ – coefficient of relative pressure losses (flow resistances) in the line joining the throttle control unit with hydraulic motor, at theoretical pump delivery Q_{Pt}
$k_{6.2}$ – coefficient of relative pressure losses (flow resistances) in hydraulic motor outlet line, at theoretical pump delivery Q_{Pt}
$k_{7.1}$ – coefficient of relative mechanical losses in hydraulic motor – cylinder, at a force $F_M=0$
$k_{7.2}$ – coefficient of relative increase of mechanical losses in motor – cylinder, at increase of force F_M
k_8 – coefficient of relative pressure losses (flow resistances) in internal ducts of hydraulic motor, at theoretical pump delivery Q_{Pt}
k_9 – coefficient of relative volumetric losses in hydraulic motor
k_{10} – coefficient of relative minimum pressure decrease in 2-way flow control valve, which still ensures the flow regulation, or coefficient of relative pressure decrease in 3-way flow control valve
k_{11} – coefficient of relative pressure decrease Δp_{DE} in directional control valve (servovalve, proportional valve) demanded by a maximum throttling section f_{DEmax} for receiving flow intensity equal theoretical pump delivery Q_{Pt}

$\overline{M}_M$ – hydraulic motor relative load coefficient $\overline{M}_M = F_M / F_{Mn}$
p_0 – the reference pressure in the oil reservoir
p_1 – pressure at the cylinder feed proportional valve inlet
p_2 – pressure in the outlet conduit from proportional valve to the cylinder
$p_{1'}$ – pressure in the inlet conduit to the proportional valve from the cylinder
$p_{2'}$ – pressure in the outlet conduit from proportional valve to the oil reservoir
p_n – nominal (rated) working pressure of hydrostatic transmission (hydraulic system)
p_{M1} – pressure in the inlet conduit to the cylinder
p_{M2} – pressure in the outlet conduit from the cylinder
p_{M1i} – pressure in the inlet chamber of the cylinder
p_{M2i} – pressure in the cylinder discharge chamber
p_{P1} – pressure in the pump inlet
p_{P2} – pump supplying pressure
p_{SP} – operating pressure overflow valve
p_{SP0} – opening pressure overflow valve for ($Q_0=0$)
p_{SPS} – operating pressure overflow valve controlled by the receiver inlet pressure
Δp_{C0} – pressure drop in the inlet conduit to the pump
Δp_{C1} – pressure drop in the inlet conduit to the control unit
Δp_{C2} – pressure drop in the line between the control unit and cylinder
$\Delta p_{C3'}$ – pressure drop in the outlet conduit from cylinder to the proportional valve
$\Delta p_{C3''}$ – pressure drop in the outlet conduit of the cylinder from the proportional valve

Δp_{DE1} – pressure drop in the proportional directional valve throttling slot f_{DE1} (at the cylinder inlet)
Δp_{DE2} – pressure drop in the f_{DE2} proportional valve throttling slot (at the cylinder outlet)

Δp_M – decrease of pressure (pressure drop) in hydraulic linear motor (cylinder)
Δp_{Mi} – pressure drop indicated between inlet and outlet chamber of the cylinder
Δp_P – increase of pressure in the pump
Δp_{Pp1} – pressure drop in the inlet channel pump (and the distributor, if any)
Δp_{Pp2} – pressure drop in the pump outlet duct (and the distributor, if there is one)
Q_0 – intensity of flow directed through the overflow valve to the oil reservoir
Q_M – hydraulic linear motor absorbing capacity, intensity of flow to hydraulic linear motor
Q_{M2} – intensity of flow from the hydraulic linear motor (cylinder)
Q_P – pump delivery
η – energy efficiency
S_{M1} – effective area of the hydraulic linear motor piston in its inlet chamber
S_{M2} – effective area of the hydraulic linear motor piston in its outlet chamber
SP – overflow valve
SPS – overflow valve controlled by the receiver inlet pressure
var – variable
v_M – hydraulic linear motor speed
$\overline{\omega}_M$ – hydraulic linear motor speed coefficient – ratio of instantaneous speed to the nominal one of a hydraulic linear motor – $\overline{\omega}_M = v_M / v_{Mn}$

REFERENCES

1. Paszota Z.: *Energy-saving structures of hydrostatic drives for ship deck machines.* Polish Maritime Research, Vol. 7, No 1, 2000, pp. 3–9.

2. Paszota Z.: *Energy efficiency of hydraulic cylinders in servo-mechanism. Conference proceedings.* International XIX Scientific Conference of Naval Architects and Marine Engineers "Marine Technology on the Theshold of XXI Century – Marine Technology 2000". May 4–6, 2000, Szczecin-Dziwnówek, Poland. Szczecin Technical University, Pol. Acad. Sci., Soc. Pol Naval Arch. Marine Eng. KORAB. 2000, pp. 139–147.

3. Paszota Z.: *Hydraulic proportional control of the cylinder with a variable pressure overflow valve – a model of the energy efficiency of the system.* Chapter in the monograph: "Research, design, manufacturing and operation of hydraulic systems", Edward Palczak editor, "Cylinder" Library (in Polish), Centrum Mechanizacji Górnictwa „Komag", Gliwice 2001, pp. 55–67.

4. Paszota Z.: *Energy – Saving Supply Structures of the Throttling Directional Control Valve in the System of Proportional Control of the Linear Hydraulic Motor.* [Proceedings] Viertes Deutch – Polnisches Seminar "Innovation und Fortschritt in der Fluidtechnik", Sopot, 20–21 September 2001, Technische Universitat Warszawa, Fakultat fur Mechatronik, Institut fur Automatik und Robotik, pp. 90–100.

5. Paszota Z.: *Energy aspects of hydrostatic drives.* Polish Maritime Research, Vol. 10, No 2 2003, pp. 18–20.

6. Paszota Z.: *An Energy Behaviour Comparison of two Kinds of Hydrostatic Drive of Ship Deck Machines.* Brodogradnja – Journal of Naval Architecture and Shipbuilding Industry, Brodarski Institut Zagreb, Vol. 52, No 3, 2004, pp. 213–222.

7. Paszota Z.: *Model of losses and energy efficiency of a hydraulic system with proportional control of the cylinder fed by a constant capacity pump in a variable pressure system.* Chapter in the monograph: "Research, design, manufacturing and operation of hydraulic systems", Edward Palczak editor, "Cylinder" Library (in Polish), Centrum Mechanizacji Górnictwa „Komag", Gliwice, 2005, pp. 145–162.

8. Paszota Z.: *Energy Saving in a Hydraulic Servomechanism System – Theory and Examples of Laboratory Verification*. Brodogradnja, Journal of Naval Architecture and Shipbuilding Industry, Zagreb, Vol. 58, No 2, 2007, pp. 146–157.

9. Paszota Z.: *Power of energy losses in the hydrostatic drive system elements – definitions, relations, change ranges, energy efficiencies. Part II – Conduits, throttling control assembly, pump*, Chapter in the monograph: "Research, design, manufacturing and operation of hydraulic systems", Andrzej Meder and Adam Klich editors, "Cylinder" Library (in Polish), Centrum Mechanizacji Górnictwa „Komag", Gliwice, 2007, pp. 121–139.

10. Paszota Z.: *Graphical presentation of the power of energy losses and power developed in the hydrostatic drive and control system elements. Part I – Systems with the series throttling control of the hydraulic rotational motor speed*, Chapter in the monograph: "Research, design, manufacturing and operation of hydraulic systems", Adam Klich, Edward Palczak and Andrzej Meder editors, "Cylinder" Library (in Polish), Centrum Mechanizacji Górnictwa „Komag", Gliwice, 2008, pp. 121–139.

11. Paszota Z.: *Losses and energy efficiency of hydraulic motors and drive systems (in Polish).* Drive and Control, No. 5, 2013, pp. 82–90.

12. Paszota Z.: *Losses and energy efficiency of drive motors and systems. Replacement of the Sankey diagram of power decrease in the direction of power flow by a diagram of power increase opposite to the direction of power flow opens a new perspective of research of drive motors and systems.* Polish Maritime Research, Vol.20, No 1 (77), 2013, pp. 3–10.

13. Paszota Z.: *Energy losses in hydrostatic drive. Monography*, LAP Lambert, Acadamic Publishing, 2016.

14. Skorek G.: *Energy characteristics of the hydraulic system with proportional control of cylinder, fed by a constant capacity pump in a constant pressure and variable pressure system (in Polish).* Doctor dissertation. Gdansk University of Technology, Faculty of Ocean Engineering and Ship Technology, 2010.

15. Skorek G.: *Energy efficiency of a hydrostatic drive with proportional control compared with volumetric control.* Polish Maritime Research, Vol. 20, No 3 (79), 2013, pp. 14–19.

16. Quan Z., Quan L., Zhang J.: *Review of energy efficient direct pump controlled cylinder electro-hydraulic technology.* Renewable and Sustainable Energy Reviews, Elsevier, Vol. 35, 2014, pp. 336–346.

CONTACT WITH THE AUTHOR

Grzegorz Skorek
e-mail: grzesko@am.gdynia.pl
Phone: +48-58-55-86-484
Department of Engineering Sciences
Faculty of Marine Engineering
Gdynia Maritime University
Morska 81–87, 81-225 Gdynia
Poland