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The accuracy of defining the energy efficiency of drive systems exemplified by comparison with hydrostatic drives with proportional motor speed control

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Abstract
The aim of the article is to look at the possibility of accurately determining the energy efficiency of drive systems. The results of experimentally determined efficiencies and the efficiencies determined from simulations of two hydrostatic systems with throttling control and fed by a constant capacity pump were compared. The research apparatus was very precisely designed, made and automated. The measuring instruments that were used are characterized by their high measuring accuracy. The issues related to the determination of the energy losses and the energy efficiency of the hydraulic motor or drive system, which should be determined as dependent on the physical quantities independent of these losses, were also discussed. A Paszota diagram of the power increase in the direction opposite to the direction of the power flow, replacing the Sankey diagram of the power decrease in the direction of the power flow in the hydraulic motor or in the drive system, was analyzed. The results showed that a Paszota diagram opens up a new perspective on research on the power of energy losses and energy efficiency of hydraulic motors and drive systems.

Nomenclature

- a – coefficient of the 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, the force currently required of a linear motor;
- F_{Mi} – force indicated on the piston of the hydraulic linear motor (cylinder);
- F_{Mm} – hydraulic linear motor's mechanical losses;
- F_{SP} – force of the spring in the overflow valve;
- k_1 – coefficient of relative volumetric losses per one shaft revolution of a fixed capacity pump;
- k_2 – coefficient of the relative decrease in the pump's 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 the pump, at $\Delta p_{Pi} = 0$;
- $k_{4.2}$ – coefficient of relative increase of the pump's mechanical losses, at an increase in the pressure in the pump's working chambers;
- k_5 – coefficient of relative pressure losses (flow resistance) 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 resistance) in the line joining the throttle control unit of the hydraulic motor, at theoretical pump delivery Q_{Pt};
- $k_{6.2}$ – coefficient of relative pressure losses (flow resistance) in the hydraulic motor's outlet line, at theoretical pump delivery Q_{Pt};

$k_{7.1}$ – coefficient of relative mechanical losses in the hydraulic motor – cylinder, at a force $F_M = 0$;
$k_{7.2}$ – coefficient of relative increase of the mechanical losses in motor – cylinder, for increase of the force F_M;
k_8 – coefficient of relative pressure losses (flow resistance) in internal ducts of hydraulic motor, at theoretical pump delivery Q_{Pt};
k_9 – coefficient of relative volumetric losses in the hydraulic motor;
k_{10} – coefficient of relative minimum pressure decrease in a 2-way flow control valve, which still ensures flow regulation, or coefficient of the relative pressure decrease in a 3-way flow control valve;
k_{11} – coefficient of relative pressure decrease Δp_{DE} in a directional control valve (servo-valve, proportional valve) demanded by a maximum throttling section $f_{DE\max}$ when receiving a flow intensity equal to the theoretical pump delivery Q_{Pt};
$\overline{M}_M$ – hydraulic motor's 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's proportional valve inlet;
p_2 – pressure in the outlet conduit from the 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 the proportional valve to the oil reservoir;
p_n – nominal (rated) working pressure of the 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 of the overflow valve;
p_{SP0} – opening pressure of the overflow valve for ($Q_0 = 0$);
p_{SPS} – operating pressure of the overflow valve controlled by the receiver's 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 the 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 – pressure decrease in the hydraulic linear motor (cylinder);
Δp_{Mi} – pressure drop indicated between the inlet and outlet chamber of the cylinder;
Δp_P – pressure increase in the pump;
Q_0 – intensity of the 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's 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 speed of a hydraulic linear motor – $\overline{\omega}_M = v_M / v_{Mn}$.

Introduction

A control system with a proportional directional throttling control valve or a directional control servo valve, controlling a cylinder (linear hydraulic motor) is used in a ship's steering gear drive, controllable pitch propeller control, a variable capacity pump control system for hydraulic deck equipment motors or fixed pitch propellers in small ships (for example ferries), in deck cranes and ship ramps (Piatek, 2004). The scope of the basic research in drives and hydrostatic controls includes the study of the energy efficiency of components and systems, including a detailed analysis of the sources of individual losses (Czyński, 2005).

The energy efficiency, which is one of the most important features that characterize the system, is defined as the ratio of the useful power P_{Mu} of the hydraulic motor to, corresponding to this value P_{Mu}, the power P_{Pc} consumed by the pump on its shaft from the motor that drives it (electric, diesel). In the case of improper selection of the type of system, this may result in an increase in the losses as well as the temperature of the liquid, and hence, in the viscosity of the liquid, which in turn causes a decrease in the efficiency of individual elements and affects the movement characteristics of the system. Therefore, energy efficiency can be a decisive factor in the applicability of a system in a specific case. However, detailed energy efficiency analysis often leads to structural improvements in various elements of the system. However, improving the quality of hydrostatic systems cannot be carried out solely by improving their elements (Paszota, 2016a).

The hydrostatic system, along with the interrelationships that occur in it, and the interdependence of phenomena occurring in various elements during the system's operation, cannot be treated as only a set of elements from which it is built. A comprehensive approach to the system reveals both the elements of the system in which the need to improve specific features is most evident, as well as making, under specific operating conditions, the selection of characteristic parameters of particular elements of the system that guarantee optimal results for the system as a whole (Paszota, 2016b).

The energy efficiency of hydrostatic transmissions, in particular with regard to throttling control of the hydraulic motor, as well as the efficiency of hydraulic servo systems may in fact be higher than the most frequent values given in the literature (Skorek, 2013). The ability to calculate the real overall efficiency of the system as a function of the many parameters that define it is becoming a useful tool for the comprehensive assessment of the quality of the designed system. The possibility of such an evaluation is also important due to the application of hydrostatic control and regulation systems in various machines and devices, as well as due to the increasing power of hydrostatic drives in the era of ever rising energy generation costs (Skorek, 2010).

In a system with too low an efficiency, the load, mainly on the pumps, increases, which leads to an increased risk of failure and the need to repair or replace it, as well as to a shorter service life (Quan, Quan & Zhang, 2014). Too low an efficiency of the system, usually resulting from intensive throttling of the liquid stream, is also a source of rapid deterioration of the exploitation characteristics, especially the lubricating properties of the hydraulic oil, which is the result of a too high operating temperature and thus a too low oil viscosity – the energy carrier in the hydrostatic transmission (Skorek, 2013).

More about the hydrostatic drive of machines and the state of the technology can be read in the literature (Osiecki, 1998; Kollek & Stasiak, 2012; Pietrzak & Okularczyk, 2012; Stefański & Zawarczyński, 2012; Siemieniako, 2013; Quan, Quan & Zhang, 2014).

Constant and variable pressure system with series throttling control of the speed of the hydraulic linear motor (cylinder)

The most common system for throttling control of a linear hydraulic motor is the system (Figure 1) in which the directional proportional control valve is supplied by a constant capacity pump in cooperation with an overflow valve that stabilizes the constant supply pressure p = cte, to be equal to the nominal pressure. With the decrease of the load F_M of the motor, and especially with the reduction of its speed v_M, the energy efficiency η of a constant pressure system with series throttle control decreases sharply (Figure 6) (Skorek, 2010).

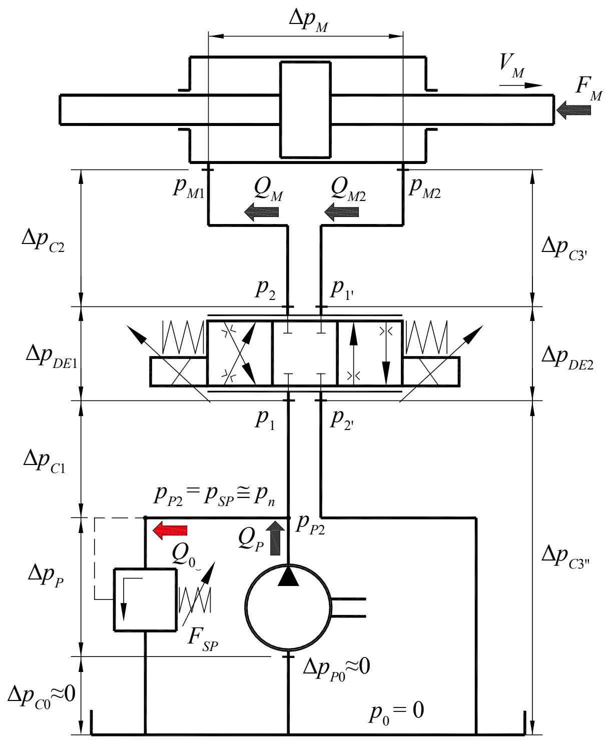

Figure 1. Diagram of the tested system at a constant pressure – *p* = cte structure

There are several ways to reduce the energy losses in the elements of a system with proportional control (in the pump, in the throttling control unit and in the hydraulic motor, especially in a linear motor), and thus increase the energy efficiency of a system with a throttling control valve.

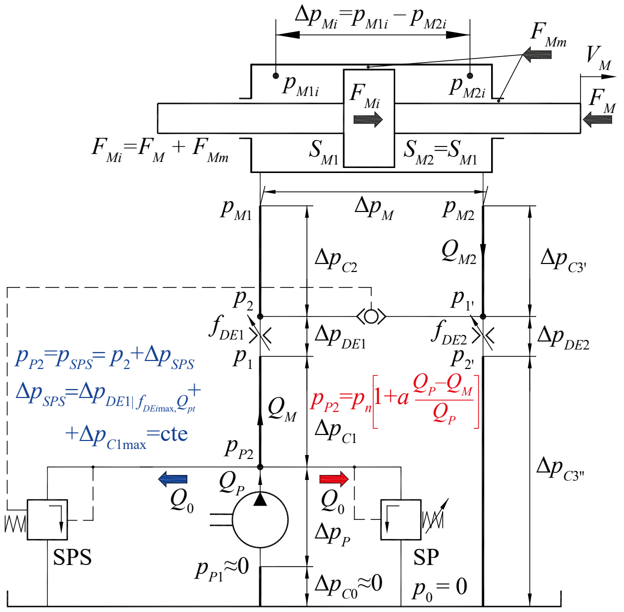

Figure 2. Diagram of the tested system with the proportional valve fed by a constant capacity pump cooperating with a controlled overflow valve in a variable pressure system – p = var (Skorek, 2010)

The hydraulic system of the drive and the proportional control of the linear hydraulic motor can be, for example, supplied by a constant capacity pump cooperating with a pressure overflow valve controlled, by the pressure p_2, at the outlet from the proportional directional control valve to the receiver. The variable pressure system p = var (Figure 2) enables a reduction of the losses in the pump, as well as in the control unit and in the hydraulic linear motor (Skorek, 2010).

Paszota diagram of the power increase in a motor or in a drive system opposite to the direction of the flow of power

Paszota (Paszota, 2016b) reduced the energy tests of the pump and hydraulic motor to independent elements of each hydrostatic drive only, in order to determine the coefficients k_i of the pressure, volume and mechanical losses occurring in these machines; the coefficients were determined at the reference viscosity $\nu_n = 35\ \mathrm{mm^2 s^{-1}}$.

The coefficients k_i are used in the mathematical descriptions of the dependence of individual losses on the physical quantities that directly define them. The characteristics of pressure, volume, mechanical and overall pump energy efficiency, as well as the hydraulic motor, are calculated simultaneously by determining the efficiency of the overall hydrostatic drive system in which the pump and hydraulic motor are used. By knowing the coefficients k_i of the losses in the elements of the hydrostatic system it is possible to obtain, by the numerical method, the efficiency dependences η_{Pp}, η_{Pv}, η_{Pm} and η_P of the pump, the efficiencies η_{Mp}, η_{Mv}, η_{Mm} and η_M of the hydraulic motor, the efficiency η_C of the conduits, the structural efficiency h_{st} of the throttling control unit of the hydraulic motor's speed (if one is used) and the overall efficiency η of the hydrostatic drive system as a function of the speed coefficient $\overline{\omega}_M$ and load coefficient $\overline{M}_M$ of the motor in the range $(0 \leq \overline{\omega}_M < \overline{\omega}_{M\,\max}, 0 \leq \overline{M}_M < \overline{M}_{M\,\max})$ of the system's operation field, for a selected ratio ν/ν_n; viscosity ν of the hydraulic oil to the reference viscosity ν_n (Paszota, 2016a).

In a hydraulic motor or drive system, the size of the power flow increases as a result of the need to balance the energy losses in the direction opposite to the direction of the power flow. The energy losses and energy efficiency of the motor or drive system should be presented as functions of the physical quantities, independent of losses; the motor speed and load are such values. However, the image of the power stream in the motor or in the drive system is still presented in the literature in the form of the traditional Sankey diagram of the power decrease in the direction of the power flow (Paszota, 2016a; 2016b).

The Sankey diagram (Figure 3), with reference to the energy balance of the drive system, shows that the useful (output) power P_u of the motor (drive system) results from the difference in the consumed (input) power P_c and the power ΔP of the losses (Paszota, 2016b):

$$P_u = P_c - \Delta P \tag{1}$$

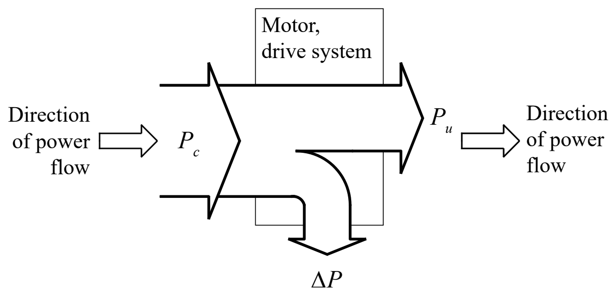

Figure 3. Sankey diagram of the decrease of the power in the motor or drive system in the direction of power flow (Paszota, 2016b); P_c is the consumed (input) power expressed in watts [W], P_u is the useful (output) power [W], ΔP is the power loss [W]

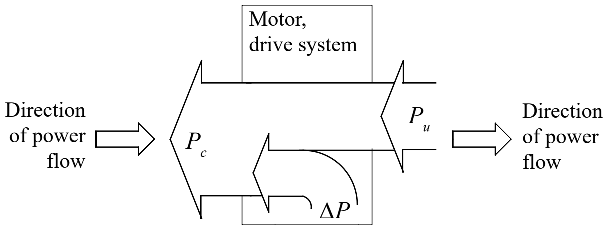

Figure 4. Paszota diagram of the increase of power in the motor or in drive system opposite to the direction of power flow, replacing the Sankey diagram presented in Figure 3 (Paszota, 2016b)

According to the Paszota diagram (Figure 4), in a motor or in a drive system, the power increases in order to overcome the energy losses in the direction opposite to the direction of power flow.

The useful (output) power P_u of the motor or drive system operating in the range ($0 \le \overline{\omega}_M < \overline{\omega}_{M\,\max}$, $0 \le \overline{M}_M < \overline{M}_{M\,\max}$) does not depend on the power ΔP of the losses in the motor or in the drive system and results from the instantaneous values of the speed coefficient $\overline{\omega}_M$ and the load coefficient $\overline{M}_M$ required by the driven device (Paszota, 2016b):

$$P_u = f(\overline{\omega}_M, \overline{M}_M) \tag{2}$$

The power losses ΔP in the motor or in the drive system depend on the structure of the drive system and on the quality of its components and, in a different way, from the instantaneous values of the motor's speed coefficient $\overline{\omega}_M$ and the load coefficient $\overline{M}_M$ of the motor in the range ($0 \le \overline{\omega}_M < \overline{\omega}_{M\,\max}$, $0 \le \overline{M}_M < \overline{M}_{M\,\max}$) (Paszota, 2016b):

$$\Delta P = f(\overline{\omega}_M, \overline{M}_M) \tag{3}$$

The power consumed (input) P_c by the motor or drive system results from the sum of the useful (output) power P_u and the power ΔP of the losses (Paszota, 2016b):

$$P_c = P_u + \Delta P = f(\overline{\omega}_M, \overline{M}_M) \tag{4}$$

The outcome of equations (3) and (4) is the description of the energy efficiency η of the motor or drive system as dependent on the $\overline{\omega}_M$ and $\overline{M}_M$ coefficients (Paszota, 2016b):

$$\eta = \frac{P_u}{P_c} = \frac{P_u}{P_u + \Delta P} = \frac{1}{1 + \dfrac{\Delta P}{P_u}} = f\left(\overline{\omega}_M, \overline{M}_M\right) \tag{5}$$

The Paszota diagram (Figure 4) of the power increase in the motor or in the drive system in the opposite direction to the power flow allows the influence of the useful (output) power P_u to be shown, i.e. the influence of the speed and load of the shaft or piston rod on the power ΔP of the losses in the motor or in the drive system and, as a result, the increase in the power of the stream in the opposite direction to the direction of the power flow and the power consumed P_c. The Paszota diagram is different to the Sankey diagram (Figure 3) with the power decrease in the motor or in the drive system in the direction of the power flow.

The field of the hydrostatic drive system on the example of two studied systems

Figure 5 presents the ranges of working fields investigated in the literature (Skorek, 2010) for hydrostatic systems with p = cte and p = var determined by the range of the change of the speed coefficient $\overline{\omega}_M$ and the load coefficient $\overline{M}_M$ of the linear hydraulic motor – cylinder.

The research stand was very accurately designed, made and automated. The applied measuring instruments were characterized by high measurement accuracy.

Maximum values of $\overline{\omega}_{M\,\max}$ and $\overline{M}_{M\,\max}$ (speed and load coefficients of the cylinder), resulting from the maximum capabilities of the drive system and the losses occurring in it, determined the motor's working field and the limits of the effective output power P_u.

The limits of the system's working field, in which there would be no volumetric, pressure and mechanical losses, were marked with a horizontal line 1 and a vertical line 2. In fact, the working fields were smaller and limited by curves 3, 4 (structure p = cte) and 5 and 6 (structure p = var). Curves 3 and 5 (Figure 5) indicate the limits of the maximum load F_M ($\overline{M}_M$) of the cylinder (Figures 1 and 2), in which mechanical losses occur (pressure losses in the cylinder channels were treated as being negligible – $k_8 = 0$, volumetric losses in the hydraulic cylinder were treated as being negligible too – $k_9 = 0$), but there were pressure losses in the system's conduits. These losses increase when there is an increase in the speed of the cylinder. As a result, the system's working area, limited by lines 3, 4 and 5 and 6, was smaller than the boundary marked with lines 1 and 2. The structure of the system and the volumetric losses in the pump dictate the limit marked with lines 4 and 6 – in the case of p = cte line 4 corresponds to a constant closing pressure of the overflow valve (regardless of the value of the coefficient $\overline{M}_M$), while line 6 in the p = var system corresponds to the increasing capacity of the pump, with the decreasing coefficient $\overline{M}_M$.

The value (lines 3 and 5 in Figure 5) depends on the structure of the system, the current value of the speed coefficient $\bar{\omega}_M$ and on the value of the coefficients k_i of the mechanical and pressure losses in the elements of the system.

Each operating point of a device powered by the p = cte and p = var system, described by the speed coefficient $\bar{\omega}_M$ and the load coefficient $\bar{M}_M$, in a common field limited by lines 4 and 5 (point 7 in Figure 5), can be achieved and determines the conditions under which the system works; this is independent of the losses in the drive system and at the same time dictates the losses.

The working field of the constant pressure system (red in Figure 5), in the zone of the large values of the cylinder load coefficient $\bar{M}_M$, is greater than the working range of the variable pressure system, in practice the latter (blue in Figure 5) is enlarged by the upper zone of the p = cte field, because it is related to the transition of the p = var system into the work area as a p = cte system (Skorek, 2010).

In a constant pressure system, throughout the entire range of the load coefficient $\bar{M}_M$ of the hydraulic motor (cylinder), the pump operates at a constant pressure close to the nominal pressure, and therefore the working field is limited by the red vertical line 4 in Figure 5. The variable pressure system (in Figure 5 in blue) can enlarge its surface with an additional field shifted to the right of the working area of the constant pressure system (marked in red). The range of the variable pressure system shifts to the right when the pump begins to operate at a lower load coefficient $\bar{M}_M$ and, therefore, at a lower pressure because the cylinder load coefficient $\bar{M}_M$ affects the pump's pressure level and the pump cooperates with the controlled overflow valve. With lower loads on

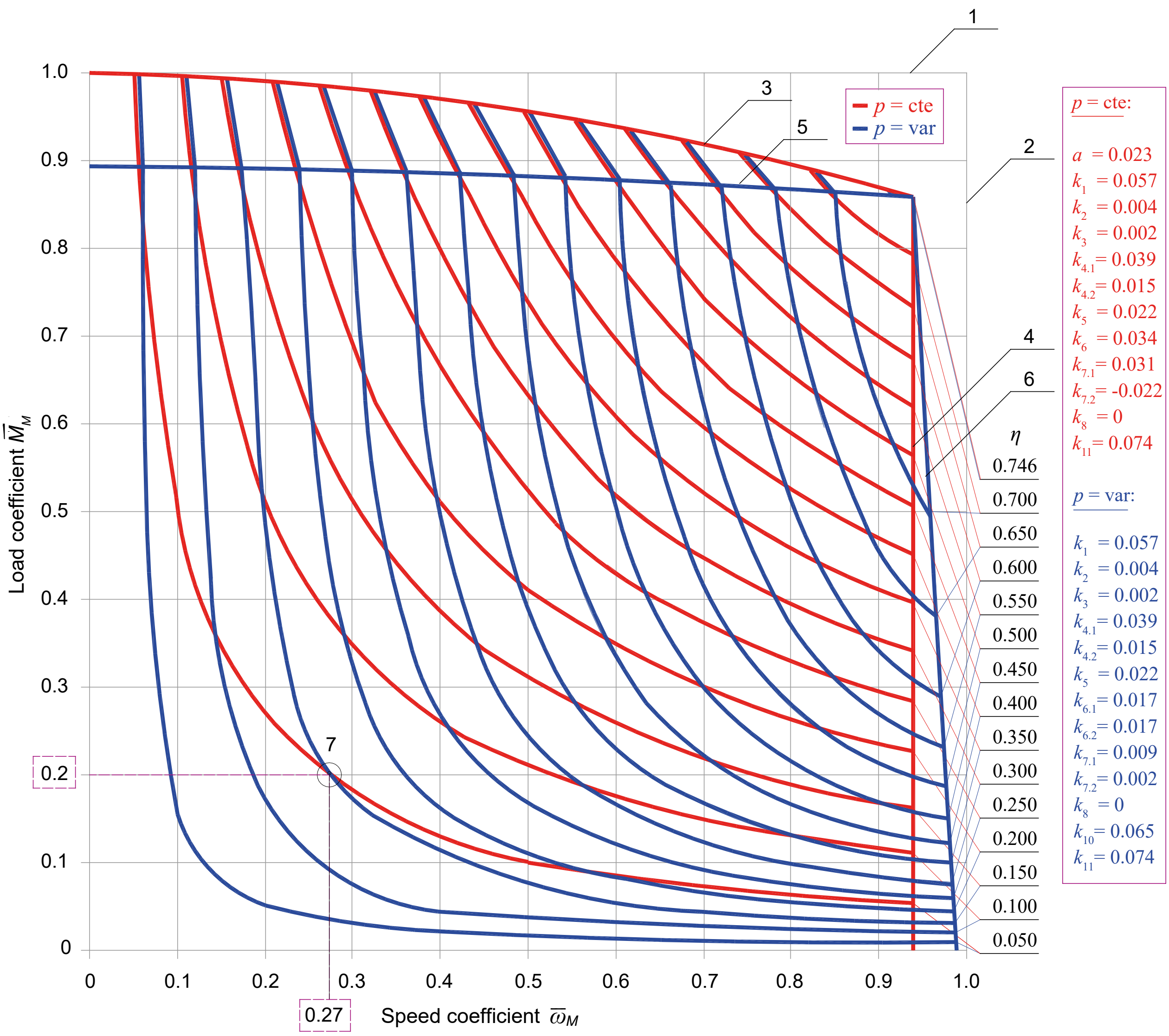

Figure 5. Fields of work and lines of constant overall efficiency η of the hydrostatic systems under investigation: a constant pressure system p = cte and a variable pressure system p = var; viscosity υ_n = 35 mm^2/s (Skorek, 2010)

the hydraulic motor (cylinder), the maximum motor speed can therefore increase as the pump runs at higher efficiency.

Figure 5 also shows the lines η = cte of the constant overall efficiency of the hydrostatic systems: the constant pressure system p = cte (red color) and the variable pressure system p = var (blue). Comparing the systems in terms of progress η = cte, their efficiency shows the influence of the speed coefficient $\overline{\omega}_M$ and the load coefficient $\overline{M}_M$ of the cylinder on the change in the overall efficiency η of the systems. For example, in point 7 (Figure 5), with a speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M$= 0.270 and load coefficient $\overline{M}_M$ equal to $\overline{M}_M$= 0.200, the overall efficiency η of the constant pressure system is η = 0.050, while the variable pressure system achieves its overall efficiency at the same point η = 0.150 and thus 3 times higher.

Equations (2)–(5) not only allow the overall efficiency of the power consumed by the hydraulic motor or drive system to be determined, but also describe the mathematical dependence of the instantaneous useful power P_u, the power ΔP of the losses and, as a result, the consumed power P_c and the instantaneous energy efficiency values η of the motor or drive system for the speed coefficient $\overline{\omega}_M$ and the load coefficient $\overline{M}_M$ of the motor shaft or the cylinder piston (Paszota, 2016b).

Energy efficiency of hydrostatic systems determined by simulation based on laboratory investigations of the coefficients k_i of the losses

Assessing the energy behavior of various forms and sizes of motors or drive systems requires a mathematical simulation description and comparison of their energy efficiency as a dependence on the speed $\overline{\omega}_M$ and load $\overline{M}_M$ coefficients of the rotational motor shaft or linear motor piston rod (hydraulic cylinder), where the coefficients change in the range ($0 \le \overline{\omega}_M < \overline{\omega}_{M\,max}$, $0 \le \overline{M}_M < \overline{M}_{M\,max}$). The coefficients

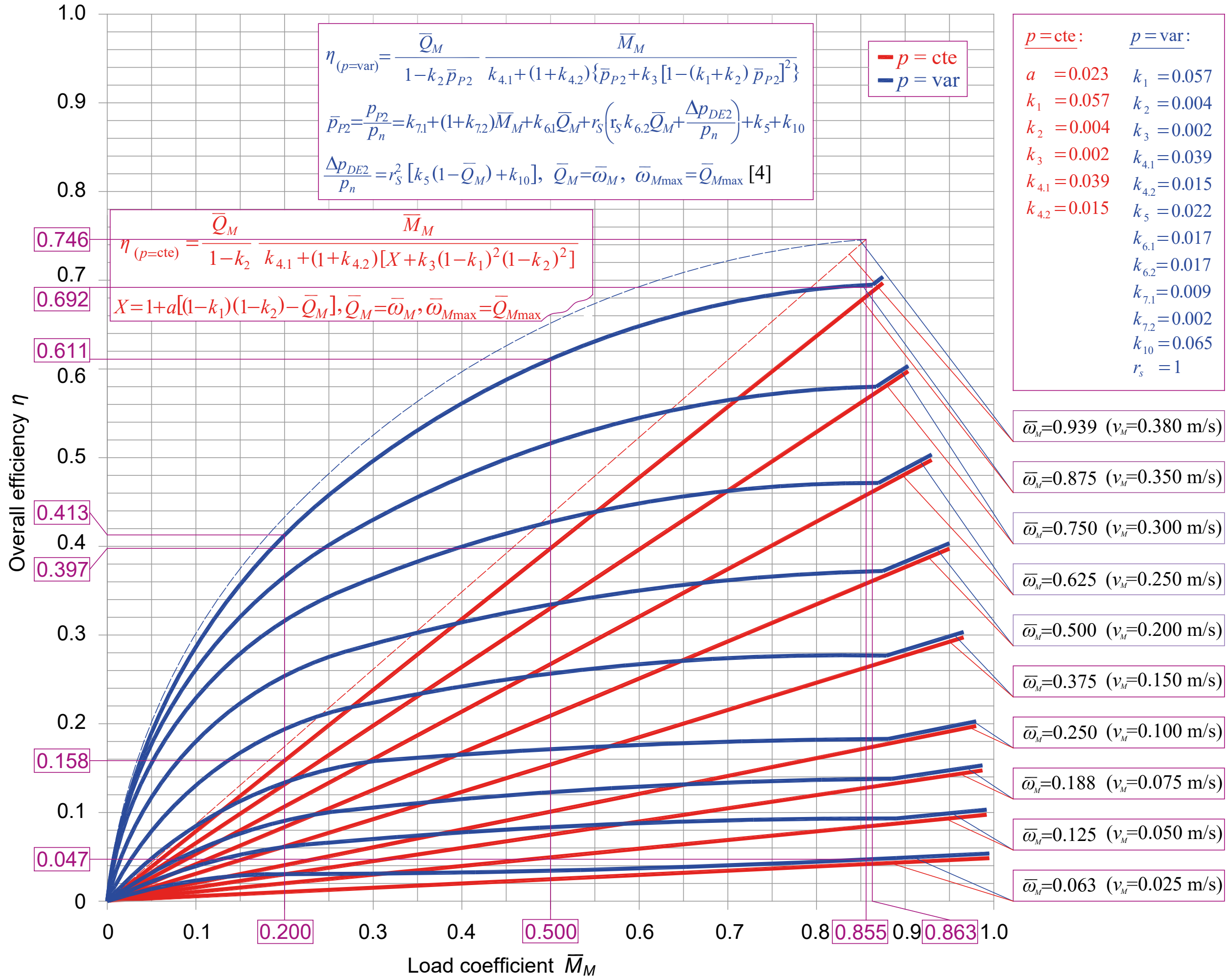

Figure 6. The dependence of the overall efficiency η for the constant pressure system (p = cte) and the variable pressure system (p = var) on the load coefficient $\overline{M}_M$ at different speed coefficients $\overline{\omega}_M$ of the cylinder; overall efficiency η of the systems defined by simulation based on the laboratory investigations of the coefficients k_i of the losses; viscosity υ_n = 35 mm^2/s (Skorek, 2010)

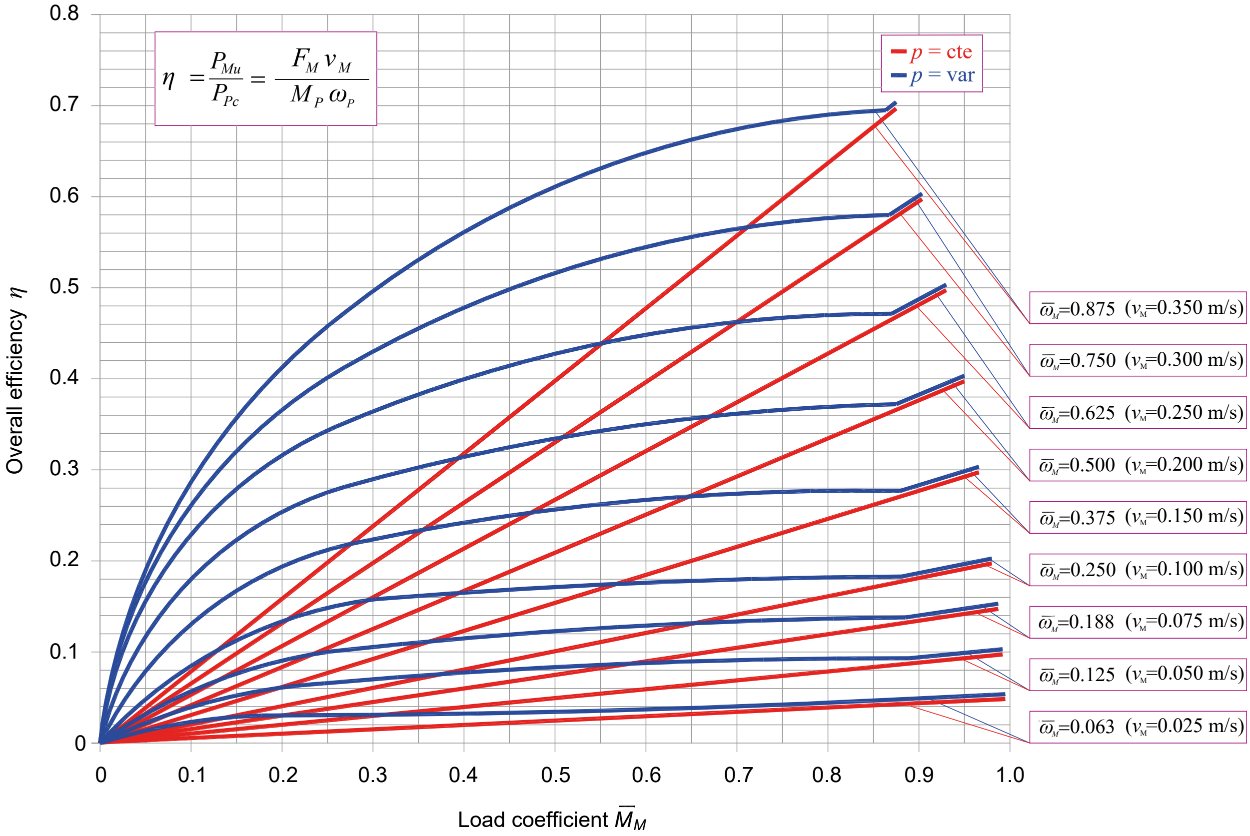

Figure 7. The dependence of the overall efficiency η of the constant pressure system (p = cte) and the variable pressure system (p = var) on the load coefficient $\overline{M}_M$ at different speed coefficients $\overline{\omega}_M$ of the cylinder; the overall efficiency η of the systems determined experimentally as ratios $P_{Mu} = F_M \cdot v_M$ to $P_{Pc} = M_P \cdot \omega_P$; viscosity υ_n = 35 mm^2/s (Skorek, 2010)

k_i of the losses were calculated with the reference viscosity, υ_n = 35 mm^2/s, of the hydraulic oil.

Figures 6 and 7 show the diagrams of the efficiency of the systems calculated by both simulation and experimentally and that they were very similar. Each curve represents the dependence of the overall efficiency for the constant pressure system (red) and the variable pressure system (blue) and is defined as the ratio of useful power to the consumed power. Efficiency has been shown to depend on the load coefficient $\overline{M}_M$, for different speed coefficients $\overline{\omega}_M$ of the cylinder's piston rod.

Figure 6 shows the overall efficiency η of the constant-pressure system p = cte (Figure 1) and variable pressure system p = var (Figure 2) determined through simulation. Figure 6 also shows the thin dashed lines of the overall efficiency η of the systems for the maximum use of the pump's efficiency system, i.e. in a situation in which the intensity Q_M of the stream directed to the cylinder through the proportional directional control valve approaches the pump's capacity Q_P. In this case, it is possible to achieve the maximum energy efficiency η of both systems equal to η = 0.746 at $\overline{M}_M$ = 0.855 (F_M = 25,650 N) and $\overline{\omega}_M$ = 0.939 (v_M = 0.380 m/s). The use of the flow intensity Q_P of the pump would be possible if the overflow valve SP (Figures 1 and 2) applied in the systems p = cte and p = var, was an ideal valve.

Due to the variable pressure system p = var, energy savings are possible, especially at a lower load coefficient $\overline{M}_M$ and higher cylinder speed coefficient $\overline{\omega}_M$. In Figure 6, an excellent increase of the overall energy efficiency of the variable pressure system in relation to the constant pressure system can be noticed, especially in the range of the average values of the load coefficient $\overline{M}_M$ and the upper values of the cylinder speed coefficient $\overline{\omega}_M$. When the cylinder's speed coefficient $\overline{\omega}_M$ was increased, the flow Q_M was increased to the cylinder at the same time, and the smaller flow Q_0 flows through the SP (SPS) overflow valve into the tank. Therefore, the overall efficiency η of the system was growing. This is due to the fact that the structural volumetric efficiency η_{stv} (of the throttle control unit) was increasing. For example, the overall efficiency η of the p = cte system, with the same coefficient $\overline{M}_M$ = 0.500 (F_M = 15,000 N) of the cylinder load and its speed coefficient equal to $\overline{\omega}_M$ = 0.875 (v_M = 0.350 m/s), assumes the value η = 0.397. However, the overall efficiency η of the p = var system, with the same load coefficients and cylinder speed coefficient, was $\eta = 0.611$.

For the cylinder load $\overline{M}_M$ the coefficient was equal to $\overline{M}_M = 0.863$ (F_M = 25,890 N), the efficiency η of both systems, for the speed coefficient $\overline{\omega}_M$ was equal to $\overline{\omega}_M = 0.063$ (v_M = 0.025 m/s) which was only about $\eta \approx 0.047$. In turn, the efficiency η of both systems, with the same load coefficient $\overline{M}_M$ equal to $\overline{M}_M = 0.863$ (F_M = 25,890 N) and at a common speed $\overline{\omega}_M$ coefficient equal to $\overline{\omega}_M = 0.875$ (v_M = 0.350 m/s), reached the approximate value of $\eta \approx 0.692$ (Skorek, 2010).

From the point of view of the overall efficiency η of the system, the greatest gain was the value of the cylinder load coefficient $\overline{M}_M$ of approximately $\overline{M}_M \approx 0.200$ ($F_M \approx$ 6000 N), with speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M = 0.875$ (v_M = 0.350 m/s). The overall efficiency η of the p = cte system was then $\eta = 0.158$, and the overall efficiency of the system p = var – $\eta = 0.413$, which is around 2.6 times higher than the efficiency of the constant pressure system. In this zone, the medium load zone begins.

Accuracy of the simulation method for determining the efficiency of the systems

In order to verify the mathematical models proposed in the simulation method for determining the energy efficiency of the motor's proportional control system and to evaluate the accuracy of this method, it was necessary to compare the energy efficiency results η of the constant pressure system p = cte and the variable pressure system p = var, determined simultaneously with the results of the direct and accurate laboratory tests (Figure 7). Therefore, these results were compared at selected values of the speed coefficient $\overline{\omega}_M$ and the load coefficient $\overline{M}_M$ of the cylinder.

The obtained results allow the conclusion to be drawn that the simulation method for determining the overall energy efficiency of a hydrostatic drive system, with proportional control of a hydraulic motor supplied by a constant capacity pump in a constant pressure system p = cte and variable pressure system p = var, has very high accuracy compared to the laboratory investigations.

The absolute error of the simulation model of the mathematical overall energy efficiency of the hydrostatic drive with proportional control of the hydraulic motor (hydraulic cylinder) supplied by a constant-capacity pump in the constant pressure system p = cte (as the difference between experimental and simulation results) was in the order of −0.0036 to +0.0009, which is equivalent to 0.4%. The absolute error of determining the overall efficiency of the system working in the variable pressure system p = var, as the difference between the experimental and simulation results, ranged from −0.0014 to +0.0108, which is in the order of 1%.

The results of laboratory verification tests confirmed with high accuracy the theoretical and mathematical descriptions of the simulations of the energy losses in elements of the hydraulic servo systems or the systems with a proportional directional control valve operating in the supply system, p = cte and p = var.

Conclusions

1. The losses and energy efficiency of the motor or drive system should be presented as functions of physical quantities that are independent of losses. Such quantities are motor speed and its load ranging in the field of work ($0 \leq \overline{\omega}_M < \overline{\omega}_{M\max}$, $0 \leq \overline{M}_M < \overline{M}_{M\max}$) (Paszota, 2016a; 2016b).
2. Assessing the energy behavior of various forms and sizes of motors or drive systems requires a mathematical description and a comparison of their energy efficiency as a dependence on the speed $\overline{\omega}_M$ and load $\overline{M}_M$ coefficients for the rotational motor shaft or linear motor piston rod (hydraulic cylinder), where the coefficients ranged in the field ($0 \leq \overline{\omega}_M < \overline{\omega}_{M\max}$, $0 \leq \overline{M}_M < \overline{M}_{M\max}$). The maximum values of the speed coefficient $\overline{\omega}_{M\max}$ and the load coefficient $\overline{M}_{M\max}$ of the hydraulic motor, resulting from the maximum capabilities of the drive system and the losses occurring in it, determine the range of the hydraulic motor's working field (Paszota, 2016a; 2016b).
3. The analysis of the p = cte and p = var hydrostatic drive systems with proportional control fed by a constant capacity pump, demonstrated that these systems, in a certain range of operating parameters, could achieve high energy efficiency values. However, the character of the changes in the constant lines of efficiency of both systems as a function of the cylinder's speed coefficient $\overline{\omega}_M$ and load coefficient $\overline{M}_M$ was different. With lower values of the load and speed of the cylinder, a drastic reduction in its energy efficiency occurred in the p = cte system. However, in the case of the p = var system, with the same parameters $\overline{\omega}_M$ and $\overline{M}_M$, the efficiency reduction was not so rapid. The biggest benefits and energy

gains from using the p = var system in comparison to the p = cte system were in the range of the average values of cylinder loads. For example, with a speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M = 0.270$ and a load coefficient $\overline{M}_M$ equal to $\overline{M}_M = 0.200$, the overall efficiency η of the constant pressure system was $\eta = 0.050$, while the variable pressure system achieved an efficiency at the same point of work of $\eta = 0.150$ and thus was 3 times higher.

4. The tested structures (p = cte and p = var) of the hydrostatic drives with throttling control of linear speed, fed by a constant capacity pump, could achieve, with a maximum load of $F_{M\max}$ ($\overline{M}_{M\max} = 0.863$) and a simultaneous maximum speed $v_{M\max}$ ($\overline{\omega}_{M\max} = 0.875$) of the hydraulic motor, the same maximum overall efficiency $\eta_{\max}$ equal $\eta_{\max} = 0.692$ of the system. The variable pressure system (p = var) then became a constant pressure system (p = cte), so the operating conditions of both systems became the same and at the same time the structural losses in the throttle control unit could be practically eliminated (Skorek, 2010).
5. The energy efficiency tests with speed series throttling control of the hydraulic motor (systems with a throttling control valve) confirmed that their energy efficiency can be much higher than the values that have been given again and again in the literature on the subject.
6. The results of the laboratory verification tests (direct laboratory tests of energy efficiency of the systems) confirmed the high accuracy of the mathematical descriptions of the simulations of the energy losses in elements of systems with a proportional control valve working in the supply system p = cte and p = var.
7. An advantage of the simulation method for evaluating the energy efficiency of drive systems is also a description of the simulation of the system work field determined by the course of the maximum speed coefficient $\overline{\omega}_{M\max}$ and the maximum load coefficient $\overline{M}_{M\max}$ of the hydraulic motor and simultaneous filling of the working field with the net of the efficiency curves η = cte, which provides the opportunity to evaluate the efficiency η at each point of this field.
8. In the simulation method for determining the energy efficiency of hydrostatic drive systems, the characteristics of the pumps and hydraulic motors were limited to only determining the coefficients k_i of the energy losses in these elements.
9. The presented test results are the first example of the practical application of simulating the working field and the energy efficiency of the system that is dependent on the speed coefficient $\overline{\omega}_M$ and load coefficient $\overline{M}_M$.
10. The article is also the first example of research into the energy efficiency of drive systems within the drive test method according to the Paszota diagram for the increase of the power in the motor or drive system opposite to the direction of power flow, replacing the Sankey diagram.

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