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INFLUENCE OF HYDRAULIC OIL VISCOSITY ON THE ENERGY EFFICIENCY OF HYDROSTATIC DRIVE
Wpływ lepkości oleju hydraulicznego na sprawność energetyczną napędu hydrostatycznego

Abstract: *The article presents the influence of oil viscosity on the overall efficiency of the compared systems with proportional cylinder control. There are energy losses in the elements of the hydraulic system, which are, among other things, a function of the viscosity of the working fluid used, as well as energy losses which are practically independent of the viscosity. In order to assess the possibilities of saving energy during the operation of the hydrostatic drive system, it is necessary to understand and describe the losses occurring in the system. Determining the energy efficiency of the system can be performed simulation with the help of a computer program using an appropriate mathematical model. The efficiency determined in this way can be used in the process of designing and operating the system.*
Keywords: energy efficiency, proportional control, hydrostatic transmission, load sensing, pump, cylinder

Streszczenie: *W artykule przedstawiono wpływ lepkości oleju na sprawność całkowitą porównywanych układów ze sterowaniem proporcjonalnym siłownika. W elementach układu hydraulicznego występują straty energii będące m.in. funkcją lepkości zastosowanego płynu roboczego oraz straty energii praktycznie niezależne od lepkości. W celu oceny możliwości oszczędzania energii podczas pracy hydrostatycznego układu napędowego konieczne jest zrozumienie i opisanie strat występujących w układzie. Wyznaczenie sprawności energetycznej systemu można przeprowadzić symulacją za pomocą programu komputerowego z wykorzystaniem odpowiedniego modelu matematycznego. Tak wyznaczona sprawność może być wykorzystana w procesie projektowania i eksploatacji systemu.*
Słowa kluczowe: sprawność energetyczna, sterowanie proporcjonalne, przekładnia hydrostatyczna, load sensing, pompa, cylinder

1. Introduction

Knowing the energy efficiency of the hydrostatic transmission is important both in nominal conditions and in the entire range of changes in operating conditions, i.e. speed, load of the hydraulic motor and hydraulic oil viscosity, especially with the parameters most frequently or the longest occurring during operation.

Currently, only some manufacturers test the energy efficiency of the machines that make up the system with a selected oil viscosity. Thus, the designer or user of the hydraulic drive system can only have the data and the characteristics he can obtain. There is no tool enabling a complete energy analysis of the hydrostatic transmission as a whole, and a whole consisting of any selected machines in the form of a computer program based on mathematical models [12÷15].

The efficiency of the gearbox should be presented as a dependence on the speed and load of the hydraulic motor, with the possibility of assessing the impact of the level of volumetric, pressure and mechanical losses, which differ in individual types of machines, as well as the impact of pressure losses in the system pipes. All these losses are also a function of the current hydraulic cylinder operating parameters and the viscosity of the oil used, changing during the operation of the system [12÷15].

The possibilities of simulation studies require the use of an appropriate model of losses and energy efficiency of a constant or variable capacity pump, and then a model of the efficiency of a system with such a pump. In order for the models to be reliable, it is necessary to compare them with the results of carefully performed tests.

In order to assess the possibilities of saving energy during the operation of the hydrostatic drive system, it is necessary to understand and describe the losses occurring in the system. Determining the energy efficiency of the system can be performed simulation with the help of a computer program using an appropriate mathematical model. The efficiency determined in this way can be used in the process of designing and operating the system. The mathematical model of a specific system should be verified in the laboratory [12÷15].

The measure of the quality and usefulness of a mathematical description is its compliance with the results of an experiment carried out on a technical scale.

Experimental tests of the basic systems of hydrostatic transmissions with throttling control of the motor speed were carried out.

A research stand has been designed.

The article describes the experimental tests carried out on the stand shown in fig. 1, the results of which were developed with the use of an appropriate program. Based on the results of laboratory tests, the values of the coefficients k_i of energy losses in the system elements and the pump speed drop were determined.

The analysis of the efficiency of individual elements of the examined structures and the comparison of the efficiency determined by simulation with those obtained in the laboratory were possible thanks to the study by prof. Z. Paszota of mathematical models of losses and energy efficiency of the systems under consideration. On the basis of these models, related to the considered structures, simulation programs for determining the energy efficiency of the systems were developed. In addition to obtaining an image of the efficiency of the systems, which is the effect of the operating parameters of the hydraulic motor and the operating conditions of the systems, it is possible to compare and assess the impact of the three structures on the energy losses occurring in them.

The aim of this article is to determine the influence of the following factors on the system efficiency (assuming a variable oil viscosity of ν_n = 10, 35 and 120 mm^{2}s^{-1}):

- power supply structures of the system (p = cte, p = var, load sensing),
- load and speed of the hydraulic motor.

The article deals with, inter alia, the comparison of the energy efficiency of three structures: constant pressure p = cte, variable pressure p = var and load sensing consisting of an actuator, conduits, proportional valve, valves: overflow SP (p = cte and p = var) and overflow of the controlled SPS (p = var) and the corresponding pump.

The measurements were performed with a laboratory computer using the LabView 6.0 program by National Instruments. The measurement results were processed in Excel. The computer with the measuring transducers was connected with the PCI 1713 Advantech measuring card. In this way, 4 signals from pressure transducers were recorded, the piston rod position signal by means of a linear displacement transducer, on the basis of which its velocity v_M was determined, and the signal of the force F_M loading the piston rod.

2. Systems with a proportionally controlled hydraulic cylinder

The current research on the influence of oil viscosity on losses and energy efficiency concerns three systems with a proportional distributor:

- powered by a constant capacity pump with an overflow valve − constant pressure structure (p=cte),
- powered by a constant capacity pump using a pressure − controlled overflow valve from the supply line of the hydraulic cylinder − variable pressure structure (p=var),
- powered by a variable capacity pump equipped with a load sensing controller − a variable pressure structure with the lowest losses and highest energy efficiency (Q_P=var) [19,20].

Figure 1 shows the view of the laboratory stand from the side of the hydraulic cylinders: double-rod tested (on the left) and loading (on the right).

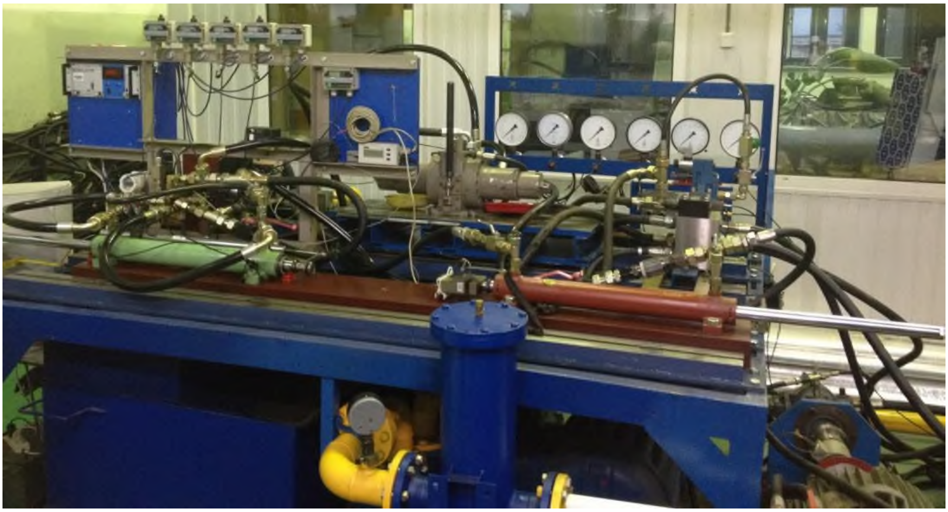

Fig. 1. View of the laboratory stand

Figures 2, 3, 4 show diagrams of the systems compared.

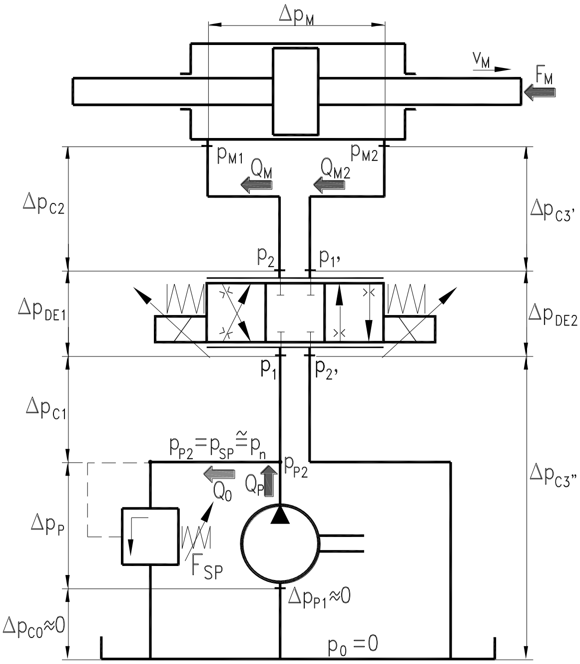

Fig. 2. Diagram of the tested system with proportional control of an actuator powered by a constant capacity pump cooperating with an overflow valve in a constant pressure system − structure p = cte [10]

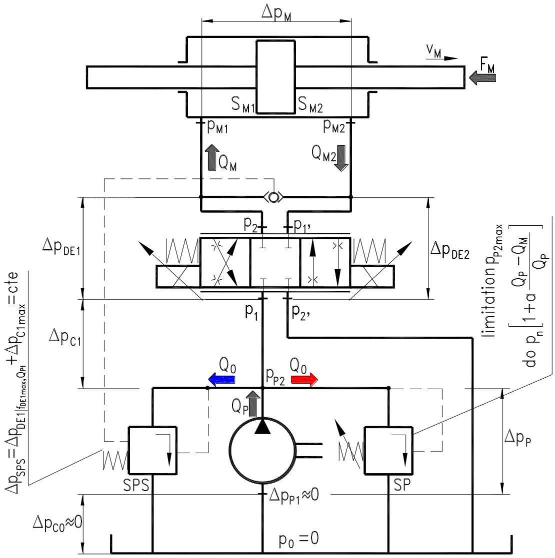

Fig. 3. Diagram of the tested system with proportional control of an actuator powered by a constant capacity pump cooperating with an overflow valve controlled in a variable pressure system − p = var [11]

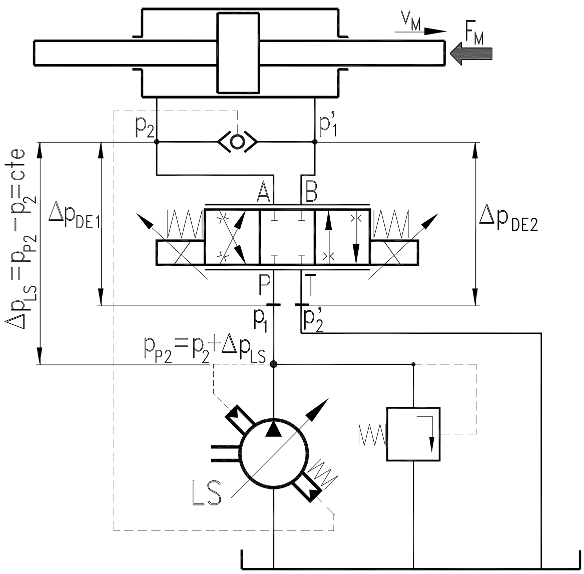

Fig. 4. Diagram of the tested system with proportional control of an actuator powered by a variable capacity pump cooperating with the load sensing regulator in a variable pressure system − Q_p = var

The hydraulic system of proportional linear hydraulic motor drive and control can be powered by a constant capacity pump cooperating with the overflow valve stabilizing the supply pressure of the proportional valve at the nominal pressure level (fig. 2), or by a pump cooperating with the overflow valve controlled by pressure on the inflow to the receiver. The p = var variable pressure system (fig. 3) enables the reduction of losses in the pump, in the control unit and in the linear hydraulic motor [14].

The variable − pressure structure p = var represents a system with a constant capacity pump cooperating with an overflow valve controlled by the supply pressure of the actuator (fig. 3). It is a solution favorable from the point of view of energy efficiency of both the actuator itself, the pump and the entire control system. The p = var variable-pressure structure with the SPS − controlled overflow valve with the current outflow pressure from the throttling manifold to the actuator allows the pressure level in the pump discharge line to be adjusted to the current actuator load, so that it also limits the pressure loss in the outlet slot of the distributor's working liquid to the tank. Additionally, this system maintains a constant speed of the piston, independent of the load. This is the effect of maintaining a practically constant pressure drop Δp_{DE1} (fig. 3) in the choking gap of the proportional distributor [13].

The hydraulic system can also be equipped with a variable displacement pump equipped with a Load Sensing regulator (fig. 3), which adjusts the pump capacity to the flow rate controlled by the throttling valve. In a system with Load Sensing control, the pump discharge pressure p_{P2} (fig. 3) is continuously adjusted to the momentary pressure exerted by the hydraulic motor. This pressure is set at a level slightly higher than the pressure in the control line. The difference $p_{P2} - p_2$, determined by the spring tension in the LS regulator (fig. 4), should ensure a minimum pressure drop corresponding to the proper operation of the throttling manifold in the event of the highest pressure loss in the pump discharge line.

The use of a variable displacement pump equipped with a load sensing controller in a proportional control system allows for the simultaneous elimination of structural volumetric losses, a significant reduction in structural pressure losses, reduction of mechanical losses in a linear hydraulic motor - actuator, as well as reduction of mechanical and volumetric losses in the pump. In addition to eliminating volumetric losses in the throttle control unit, the pressure losses in the throttle manifold are also greatly reduced by generating the required minimum pressure drop in this manifold in the entire range of the engine load change, taking into account the flow resistance in the conduit connecting the pump with the manifold. Thus, it is a system with the highest structural efficiency among systems with throttling control of the speed of a linear motor [15].

3. Influence of oil viscosity on the characteristics of elements of the tested hydraulic systems

The temperature change range ϑ of the system operation, i.e. the minimum temperature ϑ_{min} and maximum ϑ_{max} of the oil, is a function of the oil class selected according to ISO standards, defined by the kinematic viscosity ν_{40}, in mm^2s^{-1} at the temperature of 40°C. The range of operating temperature changes is also a function of the permissible viscosity limits, defined by component manufacturers, and related to the proper operation of these components, for example:

$$\nu_{min} = 10 \div 13\ mm^2s^{-1} \qquad i \qquad \nu_{max} = 115 \div 300\ mm^2s^{-1}.$$

A.S.T.M. (American Society for Testing Materials) developed viscosity-temperature diagrams that allow sufficiently accurate determination of the viscosity of mineral oils as a function of their temperature [5].

The system uses Total Azola 46 hydraulic oil (specific mass ρ = 873,3 kgm^{-3}) with kinematic viscosity changing as a function of temperature as follows:

temperature $\vartheta \approx 10°C$ – viscosity $\nu = 300\ mm^2s^{-1}$
$\vartheta \approx 15°C$ – viscosity $\nu = 120\ mm^2s^{-1}$
$\vartheta \approx 43°C$ – viscosity $\nu = 35\ mm^2s^{-1}$
$\vartheta \approx 80°C$ – viscosity $\nu = 10\ mm^2s^{-1}$

$\nu = 10\ mm^2s^{-1}$ is the acceptable minimum viscosity limit,
$\nu = 300\ mm^2s^{-1}$ is the maximum permissible viscosity limit,
$\nu = 35\ mm^2s^{-1}$ is the viscosity recommended by the manufacturers.

The evaluation of the energy behavior of various types 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 shaft of the rotary motor or the piston rod of the linear motor (e.g. a hydraulic cylinder), coefficients changing in the field of operation ($0 \le \overline{\omega}_M < \overline{\omega}_{Mmax}$, $0 \le \overline{M}_M < \overline{M}_{Mmax}$). The coefficients of losses k_i are calculated with the reference viscosity υ_n of the hydraulic oil [19,20].

Figures 5, 6 and 4 show the efficiency courses of the systems tested in the simulation. Each curve represents the relationship between the overall efficiency of the constant pressure system (red) and the variable pressure system (blue) and is defined as the ratio of useful power to consumed power. The efficiencies are shown as dependence of the load coefficient $\overline{M}_M$, for different speed coefficients $\overline{\omega}_M$ of the cylinder piston rod [19,20].

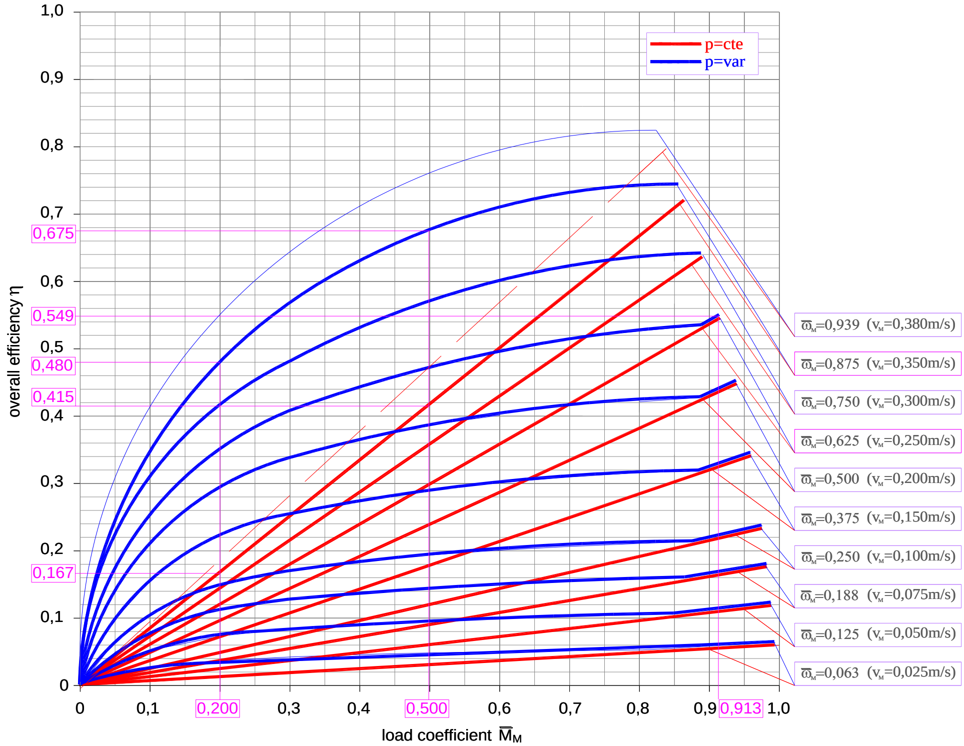

Fig. 5. Dependence of the overall efficiency η of the constant pressure system (p = cte) and the variable pressure system (p = var) on the load coefficient $\overline{M}_M$ at different cylinder speed coefficients $\overline{\omega}_M$; efficiency η of the systems determined by simulation based on the coefficients k_i of losses determined in a laboratory viscosity $\nu_n = 10$ mm²/s ($\vartheta \approx 80°C$)

Figure 5 shows a summary diagram of the overall efficiency η of a constant pressure (p = cte) and a variable pressure (p = var) system with one selected hydraulic oil viscosity ν_n equal to ν_n = 10 mm^{2}s^{-1} and with 10 cylinder speed coefficients $\overline{\omega}_M$. The overall efficiency η of both systems increases with the increase of the cylinder load coefficient $\overline{M}_M$. The overall efficiency η takes, for example, the same value in the case of p = cte and p = var systems equal to η = 0,549, with the cylinder load coefficient $\overline{M}_M$ = 0,913 and the cylinder speed coefficient $\overline{\omega}_M$ = 0,625 (v_M = 0,250 m/s) and with oil viscosity ν_n of hydraulic pressure equal to ν_n = 10 mm^{2}s^{-1}. On the other hand, the efficiency η of the system p = cte and p = var at the same viscosity ν_n of oil and, for example, with a load coefficient $\overline{M}_M$ equal to $\overline{M}_M$ = 0,500 and the cylinder speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M$ = 0,875 (v_M = 0,350 m/s), has different values, the system p = cte achieves value of the efficiency η equal to η = 0,415, while the variable pressure system p = var achieves value of the efficiency η equal to η = 0,675. With the same cylinder speed coefficient $\overline{\omega}_M$ and the load coefficient $\overline{M}_M$ = 0,200, the overall efficiency η of the system p = cte is then η = 0,167 (fig. 5). In turn, the overall efficiency η of the system p = var, with the same load coefficient $\overline{M}_M$ and the speed coefficient $\overline{\omega}_M$ of the cylinder, is then η = 0,480 with the viscosity ν_n of hydraulic oil equal to ν_n = 10 mm^{2}s^{-1}. The highest, 2,9 times, efficiency gain η in the p = var system in relation to the p = cte system is obtained in the load coefficient range $\overline{M}_M$ = 0,200.

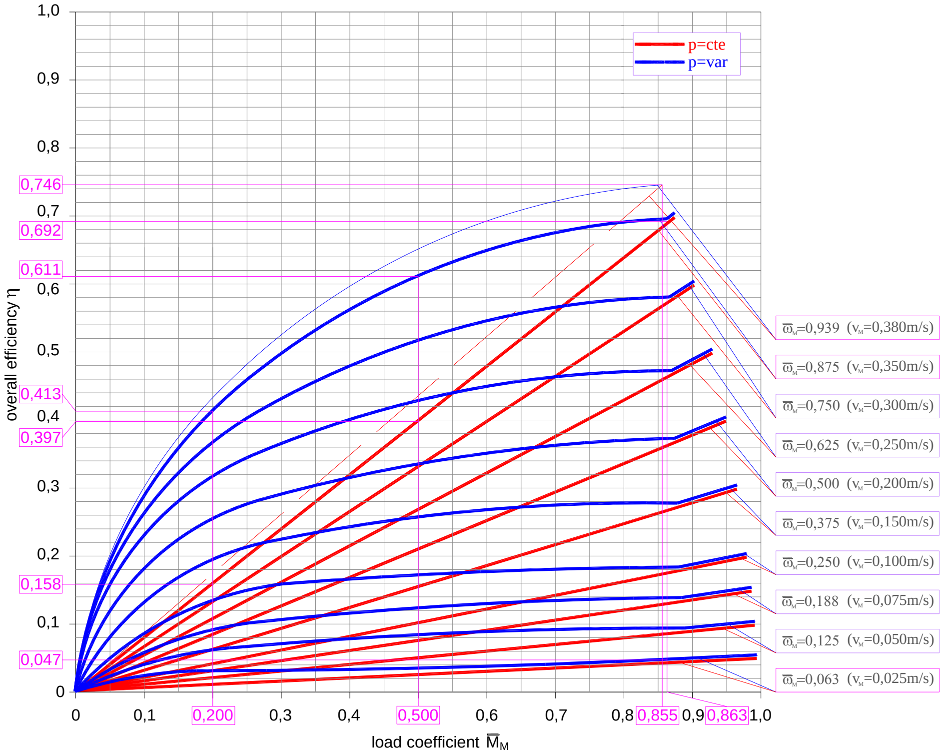

Fig. 6. Dependence of the overall efficiency η of the constant pressure system (p = cte) and the variable pressure system (p = var) on the load coefficient $\overline{M}_M$ at different coefficients $\overline{\omega}_M$ of the cylinder speed; efficiency η of systems determined by simulation based on coefficients k_i of losses determined in a laboratory viscosity ν_n = 35 mm^2/s (ϑ ≈ 43°C) [19,20]

Figure 6 shows the curves of the overall efficiency of the constant pressure system p = cte and the variable – pressure system p = var determined by simulation. Figure 6 also shows the efficiency curves η of the systems with thin dashed lines for the case of the maximum use of the pump capacity by the system, i.e. in a situation where the Q_M of the flow directed to the cylinder through the proportional control valve approaches the pump capacity Q_P. In this case, it is possible to obtain the maximum energy efficiency η of both systems equal to

η = 0,746 at $\overline{M}_M$ = 0,855 (F_M = 25650N) and $\overline{\omega}_M$ = 0,939 (v_M = 0,380m/s). The use of the total pump capacity Q_P would be possible if the SP overflow valve used in the p = cte and p = var systems was an ideal valve, i.e. one that enables operation at $Q_0 = Q_P - Q_M$ approaching zero ($Q_0 \rightarrow 0$) [19,20].

By using a variable pressure system p = var, a lot of energy is saved, especially with a lower load coefficient $\overline{M}_M$ and a higher cylinder speed coefficient $\overline{\omega}_M$. On Fig. 6 can be shown an excellent increase in the energy efficiency η of the variable pressure system in relation to the constant pressure system, especially in the range of average values of the load coefficient $\overline{M}_M$ and the upper values of the cylinder speed coefficient $\overline{\omega}_M$. As the speed ratio of the cylinder increases, simultaneously more and more stream Q_M is drawn from the pump directed to the cylinder and the smaller stream Q_0 flows through the overflow valve SP (SPS) into the tank. Accordingly, the overall efficiency η of the system increases. This is due to the fact that the structural volumetric efficiency η_{stv} (of the throttle control unit) increases. For example, the efficiency η of the system p = cte, with the same coefficient $\overline{M}_M$ = 0,500 (F_M = 15000N) of the cylinder load and its speed coefficient $\overline{\omega}_M$ = 0,875 (v_M = 0,350 m/s), takes the value η = 0,397. On the other hand, the efficiency η of the system p = var, with the same load coefficients and cylinder speed, is η = 0,611 [19,20].

With the cylinder load coefficient $\overline{M}_M$, which is equal $\overline{M}_M$ = 0,863 (F_M = 25890N), the efficiency η of both systems for its speed coefficient $\overline{\omega}_M$, equal to $\overline{\omega}_M$ = 0,063 (v_M = 0,025 m/s) is only about $\eta \approx 0{,}047$. On the other hand, the efficiency η of both systems, with the same load coefficient $\overline{M}_M$ equal to $\overline{M}_M$ = 0,863 (F_M = 25890N) and with a common speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M$ = 0,875 (v_M = 0,350 m/s), reaches the highest value of approximately $\eta \approx 0{,}692$.

From the point of view of overall efficiency η of the system, the greatest gain occurs around the cylinder load coefficient $\overline{M}_M$ equal to approximately $\overline{M}_M \approx 0{,}200$ ($F_M \approx 6000$N), with its speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M$ = 0,875 (v_M = 0,350m/s). The efficiency of the system p = cte is then η = 0,158, and the efficiency of the system p = var – η = 0,413, i.e. it is about 2,6 times higher than the efficiency of the constant pressure system.

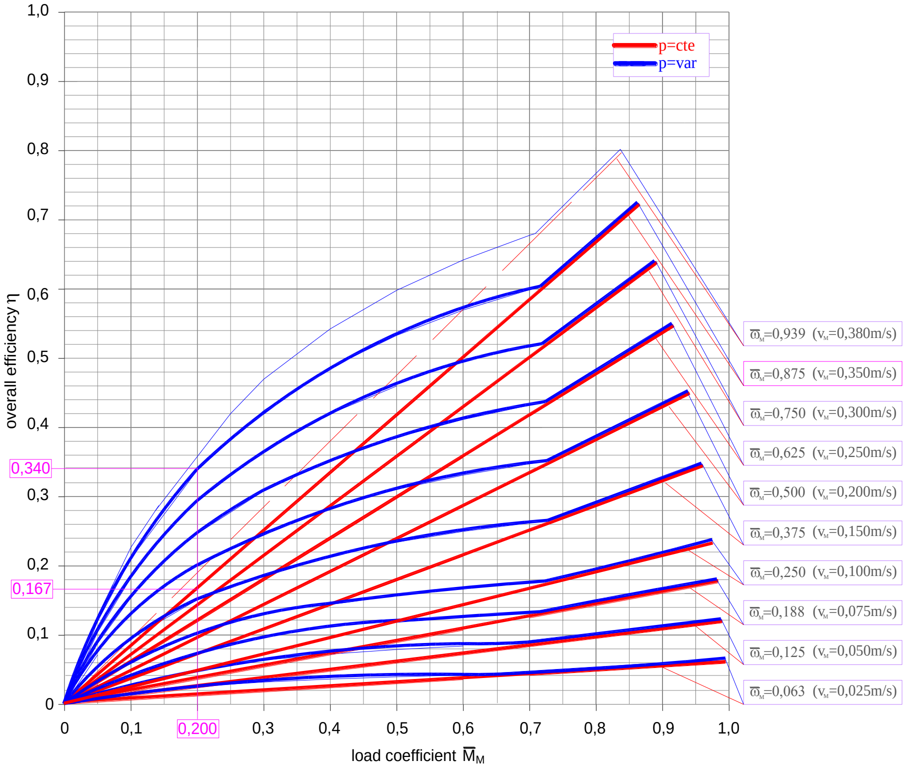

Fig. 7. Dependence of the overall efficiency η of the constant pressure system (p = cte) and the variable pressure system (p = var) on the load coefficient $\overline{M}_M$ at different coefficients $\overline{\omega}_M$ of the cylinder speed; efficiency η of systems determined by simulation based on coefficients k_i of losses determined in a laboratory viscosity ν_n = 120 mm^2/s ($\vartheta \approx$ 15°C)

Figure 7 shows a summary diagram of the overall efficiency η of a constant pressure (p = cte) and a variable pressure (p = var) system with one selected hydraulic oil viscosity ν_n of 120 mm^2/s and with ten cylinder speed coefficients $\overline{\omega}_M$.

From the point of view of overall efficiency η of the system, the greatest gain occurs around the cylinder load coefficient $\overline{M}_M$ equal to approximately $\overline{M}_M \approx 0{,}200$, with its speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M = 0{,}875$ ($v_M = 0{,}350$m / s). The efficiency η of the system p = cte is then η = 0,167, and the efficiency of the system p = var - η = 0,340, i.e. it is approximately 2 times higher than the efficiency of the constant pressure system.

Briefly summarizing, it can be stated that when comparing only two hydrostatic drive systems, in terms of viscosity influence on the overall efficiency of the system, it can be noticed that the higher the oil viscosity, the lower the efficiency gain between the systems compared. On the other hand, the lower the viscosity of the hydraulic oil, the overall efficiency η of the energy - saving system in relation to the efficiency η of the less energy - efficient system increased faster in the range of lower load coefficients $\overline{M}_M$, especially in the range of higher cylinder speed coefficients $\overline{\omega}_M$, which was illustrated in figures 5, 6 and 7 above.

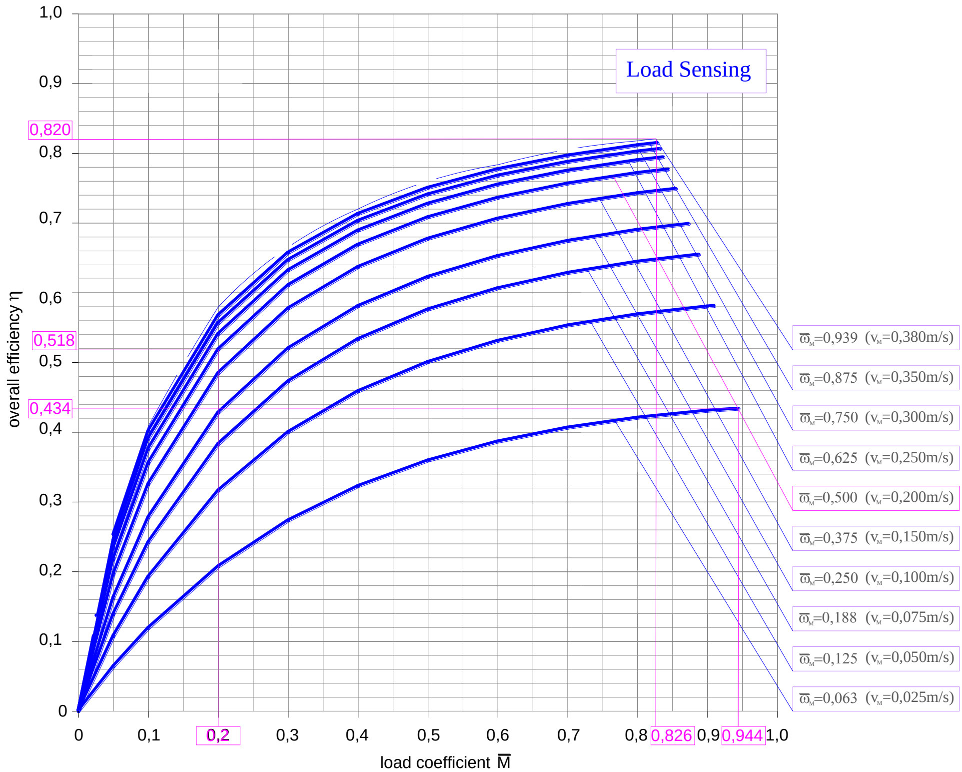

Fig. 8. Dependence of the overall efficiency η in the load sensing system on the load coefficient $\overline{M}_M$ at different coefficients $\overline{\omega}_M$ of the cylinder speed and viscosity ν_n of hydraulic oil equal to ν_n = 10 mm²/s ($\vartheta \approx 80°C$)

Figure 8 shows a collective diagram of the overall efficiency η of a system with a variable capacity pump equipped with a load sensing controller, made with one hydraulic oil viscosity ν_n, which is equal ν_n = 10 mm^2s^{-1} and with 10 different values of the cylinder speed coefficient $\overline{\omega}_M$. Efficiency η then takes, for the same viscosity ν_n, exemplary values (fig. 8): η = 0,434 with a large cylinder load coefficient $\overline{M}_M$ = 0,944 and with a small cylinder speed coefficient $\overline{\omega}_M$ = 0,063 (which corresponds to the cylinder speed v_M =

0,025 m/s); efficiency η = 0,820 with a high load coefficient $\overline{M}_M$ of $\overline{M}_M$ = 0,826 and a high speed coefficient $\overline{\omega}_M$ which is equal $\overline{\omega}_M$ = 0,939 (v_M = 0,380 m/s).

The highest overall efficiency gain η of the l–s system is observed at low values of the cylinder load coefficient $\overline{M}_M$ in the range from $\overline{M}_M$ = 0,050 to the value of $\overline{M}_M$ = 0,200 and at oil viscosity ν_n equal to ν_n = 10 mm^{2}s^{-1} (fig. 8). Most often it is in this area that you work. For example, with a cylinder load coefficient $\overline{M}_M$ which is equal $\overline{M}_M$ = 0,200 and a cylinder speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M$ = 0,500 (which corresponds to the cylinder speed v_M = 0,200 m/s), the overall efficiency η of the load sensing system is then η = 0,518.

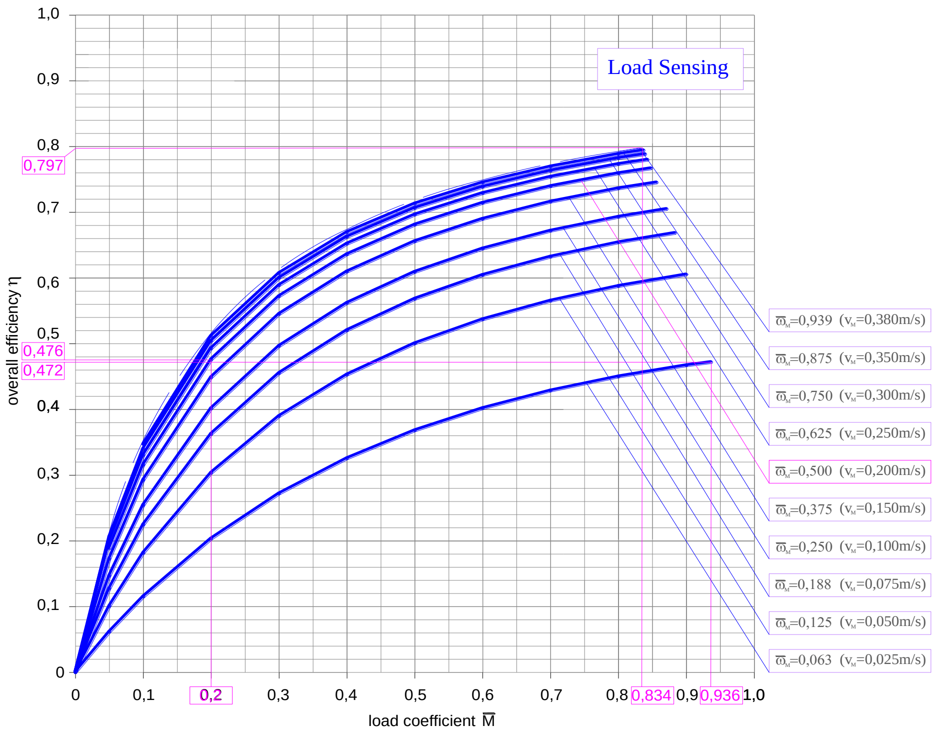

Fig. 9. Dependence of the overall efficiency η in the load sensing system on the load coefficient $\overline{M}_M$ at different coefficients $\overline{\omega}_M$ of the cylinder speed and viscosity ν_n of hydraulic oil equal to ν_n = 35 mm^2/s ($\vartheta \approx 43$°C)

Figure 9 shows a collective diagram of the overall efficiency η of a system with a variable capacity pump equipped with a load sensing controller, performed with one hydraulic oil viscosity ν_n equal to ν_n = 35 mm^2/s and with 10 different values of the cylinder speed coefficient $\overline{\omega}_M$. Efficiency η takes, for one viscosity ν_n equal to ν_n = 35 mm^2/s, sample values (fig. 9): η = 0,472 with a big cylinder load coefficient $\overline{M}_M$ equal to $\overline{M}_M$ = 0,936 and with a small cylinder speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M$ = 0,063 (v_M = 0,025 m/s); efficiency η = 0,797 with a big load coefficient $\overline{M}_M$, which is equal $\overline{M}_M$ = 0,834, and a big speed coefficient $\overline{\omega}_M$, which is equal $\overline{\omega}_M$ = 0,939 (v_M = 0,380 m/s).

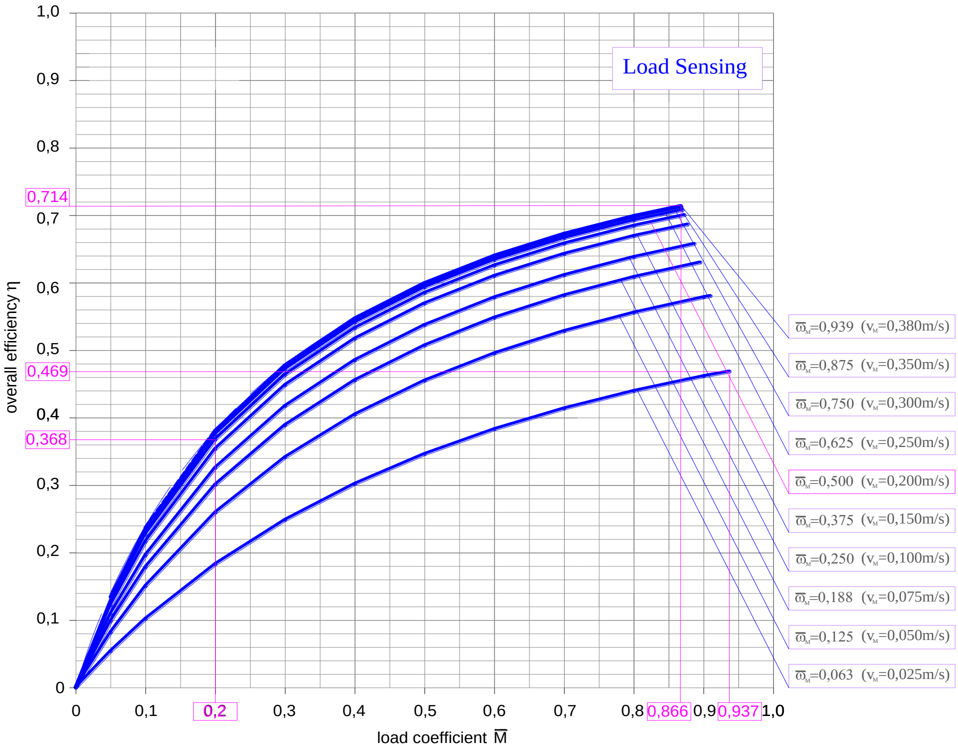

Fig. 10. Dependence of the overall efficiency η in the load sensing system on the load coefficient $\overline{M}_M$ at different coefficients $\overline{\omega}_M$ of the cylinder speed and viscosity ν_n of hydraulic oil equal to ν_n = 120 mm^2/s ($\vartheta \approx$ 15°C)

Figure 10 shows a collective diagram of the overall efficiency η of a system with a variable capacity pump equipped with a load sensing controller, performed with one hydraulic oil viscosity ν_n equal to ν_n = 120 mm^2/s and with 10 different values of the cylinder speed coefficient $\overline{\omega}_M$. Efficiency η takes, for one viscosity ν_n equal to ν_n = 120 mm^2/s, sample values (fig. 10): η = 0,368 with a big cylinder load coefficient $\overline{M}_M$ equal to $\overline{M}_M$ = 0,937 and with a small cylinder speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M$ = 0,063 (v_M = 0,025 m/s); efficiency η = 0,714 with a big load coefficient $\overline{M}_M$ equal to $\overline{M}_M$ = 0,866 and a big speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M$ = 0,939 (v_M = 0,380 m/s).

Comparing the characteristics of the overall efficiency η of the load sensing system with proportional control of the cylinder, presented in figures 5, 6 and 7, regarding the influence of hydraulic oil viscosity on the overall efficiency η of the system, it can be stated: that the efficiency η tends to increase strongly in the range of low cylinder load coefficients $\overline{M}_M$ at low hydraulic oil viscosities ν_n equal to ν_n = 10 mm^{2}s^{-1} and ν_n equal to ν_n = 35 mm^{2}s^{-1}. On the other hand, with a high oil viscosity ν_n equal to ν_n = 120 mm^{2}s^{-1}, the overall efficiency η of the l–s system increases more smoothly with the increase of the load coefficient $\overline{M}_M$ and the cylinder speed coefficient $\overline{\omega}_M$.

Therefore, summarizing briefly, it can be stated that the efficiency η of the l–s system, regardless of the viscosity ν_n of the hydraulic oil, increases rapidly at low cylinder speed coefficients $\overline{\omega}_M$ and at low cylinder load coefficients $\overline{M}_M$ (figures 8, 9 and 10).

4. Summary

The hydrostatic transmission, as one of the basic hydraulic systems, is an object that has long been known and often used. At the current stage of development of hydrostatic drive systems, efforts are made to improve their efficiency and lifetime, increase the control accuracy, and thus increase the accuracy of technological tasks performed and the automation of selected working movements [1÷27].

One of such solutions, ensuring high efficiency of hydrostatic drive systems of machines and vehicles, are load sensing systems, which in relation to throttled – controlled systems are characterized by higher efficiency in a significant part of the working range of hydraulic pumps. Hydrostatic drive systems with load compensation ensure the maintenance of constant operating parameters of the hydraulic system, regardless of the size and nature of its load, and also allow for precise control of the cylinders of the system.

Load sensing systems prove themselves as very efficient drive systems used in various types of machines and devices, both on land and at sea. They ensure, for example, in working machines, maintaining a constant speed of the machine tool during specific technological tasks, regardless of the nature and value of the load, and also determine the precision of the tasks performed [10,24,27].

The evaluation of the energy behavior of a hydraulic motor is an evaluation of its overall efficiency.

The structure of the hydraulic system has a major influence on the efficiency of the hydraulic system. Its influence is most often considered with the assumption of an ideal pump and motor and the assumption that energy losses actually occurring in the pump and the motor will further proportionally reduce the overall efficiency of the system.

The picture of the mutual influence of losses in all elements of the hydraulic system turns out to be much more complex. The instantaneous energy efficiency of the pump is, for example, among other factors, primarily due to the control structure of the hydraulic motor used.

The use, in the proportional control system, of supplying the throttling manifold (servo valve or proportional distributor) with a constant capacity pump cooperating with the overflow valve controlled by the cylinder supply pressure allows to minimize structural pressure losses in the system. It also allows to reduce the pressure in the drain line of the cylinder and the mechanical losses in the cylinder [19,20]. Models of structural efficiency η_{st} of the throttling control unit, overall η of the system, as well as models of the range of change of the speed coefficient $\overline{\omega}_M$ and the motor load coefficient $\overline{M}_M$ (maximum values – $\overline{\omega}_{M\max}$ and $\overline{M}_{M\max}$) enable their determination as a function of the coefficients $\overline{\omega}_M$ and $\overline{M}_M$. The coefficients k_i of energy losses used in the models in the system elements are determined with reference to:

- theoretical capacity of Q_{Pt},
- nominal pressure p_n of gear operation,
- recommended hydraulic oil viscosity ν_n = 35 mm^2s^{-1}.

Descriptions of the dependence of the coefficients k_i on the viscosity υ of the hydraulic oil used allow the assessment of the influence of viscosity on the efficiency of the transmission.

There are energy losses in the elements of the hydraulic system, which are, inter alia, a function of the viscosity of the working fluid used, as well as energy losses which are practically independent of the viscosity.

In a proportional control system, in which the pump with constant capacity cooperates with the overflow valve and supplies the throttling valve at almost constant pressure, the change in the viscosity of the hydraulic oil used in the range of $10\ mm^2s^{-1} < \nu < 300\ mm^2s^{-1}$ has practically no effect on the efficiency curves η = cte of the system determined as a function of the speed coefficient $\overline{\omega}_M$ and the load coefficient $\overline{M}_M$ of the hydraulic motor (figs. 5, 6 and 7).

The change in oil viscosity υ has a significant impact on the operating range ($\overline{\omega}_M$, $\overline{M}_M$) of the hydraulic motor in the system, and consequently - on the maximum achievable energy efficiency η of the system.

The lowest oil viscosity causes the maximum reduction in the range of change of the engine speed coefficient $\overline{\omega}_M$. This is the result of maximum volumetric losses in the pump and the hydraulic motor.

At the lowest oil viscosity, on the other hand, the highest values of the motor load coefficient $\overline{M}_M$ are obtained, as a result of the lowest resistances of the laminar flow in the system pipes.

The highest oil viscosity maximizes the range of changes in the speed coefficient $\overline{\omega}_M$ of the hydraulic motor and minimizes the range of changes in its load coefficient $\overline{M}_M$.

Both structures (p = cte and p = var) of throttling control of the series speed of a linear hydraulic motor, powered by a constant displacement pump, can reach, in the period of maximum load F_{Mmax} and simultaneous maximum cylinder speed v_{Mmax}, the same maximum overall efficiency η_{max} of the system. It is close to the value of the maximum energy efficiency η_{max} of the system with volumetric speed control of the hydraulic motor (variable displacement pump). The variable pressure system (p = var) then becomes a constant pressure system (p = cte), so the operating conditions of both systems become the same and at the same time the structural losses in the throttle control unit can be practically eliminated [19,20].

The main conclusion resulting from the examples provided is as follows: the maximum achievable energy efficiency values are the same in systems with different structures. The remarkable increase in efficiency η of the p = var system is visible at higher speed coefficient $\overline{\omega}_M$ and at lower cylinder load coefficients $\overline{M}_M$. On the other hand, at the highest cylinder load coefficients $\overline{M}_M$, the efficiency η of the two compared structures are equal to each other. Due to the use of the variable pressure system p = var, a significant increase in efficiency η is obtained at lower loads on the cylinder. At small values of the cylinder speed coefficient $\overline{\omega}_M$, the profit related to the application of the p = var system is small, mainly due to the volumetric losses related to the discharge of excess liquid to the tank.

From the point of view of overall efficiency η of the system, the greatest gain occurs around the cylinder load coefficient $\overline{M}_M$ equal to approximately $\overline{M}_M \approx 0{,}200$, with its speed coefficient $\overline{\omega}_M$ equal to $\overline{\omega}_M = 0{,}875$ (v_M = 0,350 m/s). The efficiency η of the system p = cte is then $\eta = 0{,}167$, and the efficiency of the system p = var − $\eta = 0{,}340$, i.e. it is approximately 2 times higher than the efficiency of the constant pressure system.

Briefly summarizing, it can be stated that when comparing only two hydrostatic drive systems, in terms of viscosity influence on the overall efficiency of the system, it can be noticed that the higher the oil viscosity, the lower the efficiency gain between the systems compared. On the other hand, the lower the viscosity of the hydraulic oil, the overall efficiency η of the energy - saving system in relation to the efficiency η of the less energy - efficient system increased faster in the range of lower load coefficients $\overline{M}_M$, especially in the range of higher cylinder speed coefficients $\overline{\omega}_M$, which was illustrated in figures 8, 9 and 10 above.

Comparing the characteristics of the overall efficiency η of the load sensing system with proportional control of the cylinder, presented in figures 8, 9 and 10, regarding the influence of hydraulic oil viscosity on the overall efficiency η of the system, it can be stated: that the efficiency η tends to increase strongly in the range of low cylinder load coefficients $\overline{M}_M$ at low hydraulic oil viscosities ν_n equal to $\nu_n = 10$ mm^{2}s^{-1} and ν_n equal to $\nu_n = 35$ mm^{2}s^{-1}. On the other hand, with a high oil viscosity ν_n equal to $\nu_n = 120$ mm^{2}s^{-1}, the overall efficiency η of the l−s system increases more smoothly with the increase of the load coefficient $\overline{M}_M$ and the cylinder speed coefficient $\overline{\omega}_M$.

Therefore, summarizing briefly, it can be stated that the efficiency η of the l−s system, regardless of the viscosity ν_n of the hydraulic oil, increases rapidly at low cylinder speed coefficients $\overline{\omega}_M$ and at low cylinder load coefficients $\overline{M}_M$ (figures 5, 6 and 7).

The reduction of the range ($\overline{\omega}_M$, $\overline{M}_M$) and the associated reduction of the maximum efficiency η_{max} of the system is particularly pronounced after exceeding the viscosity ν = 100 mm^{2}s^{-1}. This viscosity range should be avoided during system operation.

The range of low temperatures of the oil working in the hydrostatic system is unacceptable mainly due to poor suction conditions of the pump and the phenomenon of cavitation, and in the event of a complete interruption of the stream at the pump suction - due to the possibility of seizure of the pump due to lack of lubrication. The high temperature range of the oil, on the other hand, is not suitable for operation not only because of an excessive drop in viscosity and the associated increase in leakage, but also because of the accelerated degradation of the oil.

Based on the mathematical descriptions of energy losses in the system elements:

- in a constant displacement and variable displacement pump,
- in a linear hydraulic motor - an actuator,

and about the characteristics of the work:

- overflow valve,
- choke distributor,

it is possible to simulate the energy behavior of the system at each point in its field of operation described by the range ($\overline{\omega}_M$, $\overline{M}_M$) of changes in the coefficients: engine speed $\overline{\omega}_M$ and load $\overline{M}_M$.

The conducted analysis of two systems with proportional control supplied by a constant capacity pump allows to conclude that these systems, within a certain range of operating parameters, enable the achievement of efficiency close to that of the system with a variable capacity pump.

However, the nature of changes in the system efficiency constant lines as a function of load coefficients $\overline{M}_M$ and actuator speed $\overline{\omega}_M$ is different. At lower load $\overline{M}_M$ and speed $\overline{\omega}_M$ parameters of the actuator, the p = cte system drastically reduces its energy efficiency. However, in the case of the p = var system, with the same parameters $\overline{M}_M$ and $\overline{\omega}_M$, the decrease in efficiency is not so rapid.

The values of η_{max} were obtained for changing values of the load coefficients $\overline{M}_M$ and actuator speed $\overline{\omega}_M$. It should be emphasized here that only in the high-load zone of the actuator, at the pressure $p_{P2} \approx p_n$, and only with the power supply Q_M of the actuator close to the pump capacity $Q_P - Q_M \approx Q_P$, high efficiency of the system can be expected.

During the tests, a fundamental difference in the energy behavior of the examined structures was revealed, resulting from different pressures occurring in them during operation.

In order to compare the energy of various system structures, it becomes necessary to calculate and present power charts of losses occurring in the examined structures, because comparing the efficiency of the systems alone does not give a real picture of losses and may lead to erroneous conclusions.

5. References

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