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CHARACTERISTICS OF LOSS POWER IN COMPARED HYDRAULIC SYSTEMS

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Abstract

In this article, two hydrostatic systems with a throttling steering fed by a constant capacity pump were compared. 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 a speed and a load of hydraulic cylinder, which were different due to structure and ability of an energy saving, were presented and were compared. 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 the required unchanged usable pump-driven hydraulic cylinder. Presents the impact on the output (useful) power of the 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 occurs in the cylinder, the 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.

Keywords: *power, energy losses, hydraulic system, pump, throttling valve, cylinder*

1. Introduction

The article presents characteristics of loss power and power developed in elements of two different hydraulic systems with throttling control of linear speed of cylinder. The analysis was performed comparing at the selected parameters of operation of the hydraulic cylinder power lines of energy losses in the elements of these structures.

The study was concerned 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 common system of proportional throttling control of the hydraulic cylinder is a linear system (Fig. 1), wherein the proportional directional control valve is supplied with a constant capacity pump cooperating with the overflow valve stabilizing a constant level of the supply pressure – p = cte. The pressure drop in the cylinder balances the load acting on the hydraulic cylinder. The proportional throttling valve generates two pressure drops on 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 consist pump and 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 $\bar{M}_M$ and the speed coefficient $\bar{\omega}_M$. With decreasing engine load, especially with engine speed dropping, the efficiency η decreases rapidly [1].

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 $p = \text{var}$ (Fig. 2) allows reduction of losses in the pump, in the control unit and in the hydraulic cylinder [1, 3].

In the variable pressure system $p = \text{var}$, 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 papers [3-5].

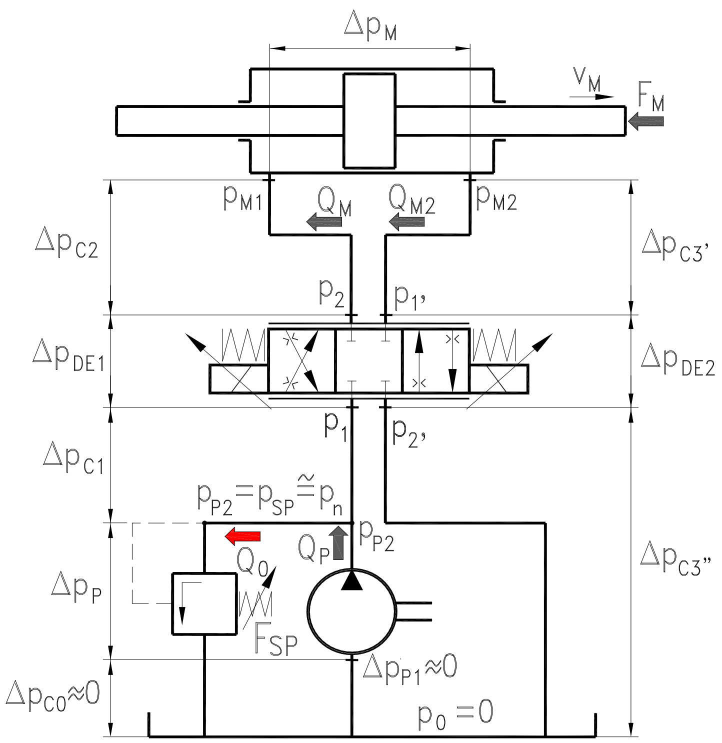

Fig. 1. Diagram of the test system fed at constant pressure – structure p = cte

The variable pressure structure $p = \text{var}$ 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

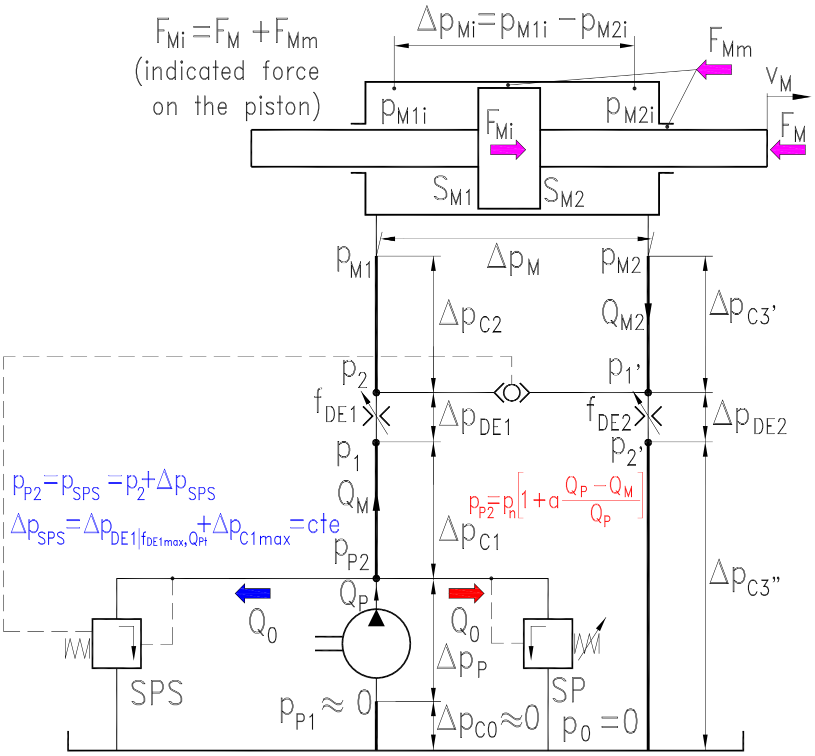

Fig. 2. Schematic diagram of a proportional valve system supplied by a constant capacity pump working with an controlled overflow valve in a variable-pressure system – p = var

pressure system $p = \text{var}$ 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) [1].

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.

2. Structural loss power in two compared hydraulic systems

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:

$$\Delta P_{st} = \Delta P_{stp} + \Delta P_{stv} \, . \tag{1}$$

Figure 3 shows the diagram of structural loss power ΔP_{st} in constant pressure system ($p = \text{cte}$) and variable pressure system ($p = \text{var}$).

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

With the load coefficient $\bar{M}_M = 0$ and the speed coefficient $\bar{\omega}_M = 0.063$ ($v_M = 0.025$ m/s) of the cylinder, the loss power ΔP_{st} of the constant pressure system $p = \text{cte}$ achieves the greatest value of

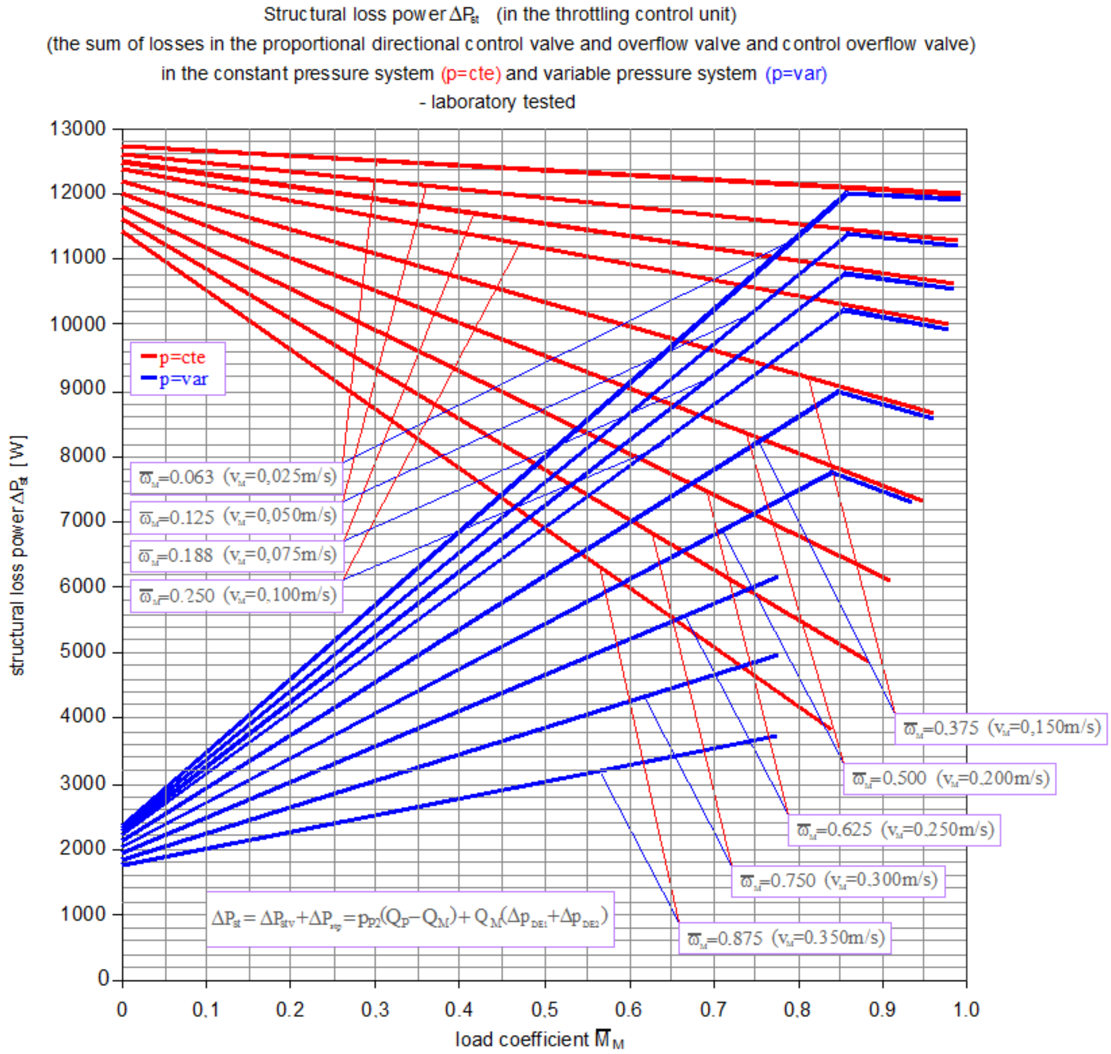

Fig. 3. A dependence ΔP_{st} of structural loss power in the throttle control unit (sum of structural pressure loss power ΔP_{stp} in the proportional directional control valve and the structural volume loss power ΔP_{stv} in the overflow valve and in the control overflow valve) in constant pressure 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

ΔP_{st} = 12700 W. 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} = 12000 W. 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} = 3815 W. 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.

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 = 0.025 m/s) of the cylinder, the structural loss power decreases from ΔP_{st} = 12700 W (p = cte) to about ΔP_{st} = 2400 W (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), i.e. 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} = 12000 W.

In the p = var system, when the hydraulic cylinder is operating at a high-speed coefficient of

$\bar{\omega}_M = 0.875$ ($v_M = 0.350$ m/s), the structural loss power ΔP_{st} decreases markedly, changing from $\Delta P_{st} = 1780$ W at $\bar{M}_M = 0$ to $\Delta P_{st} = 3800$ W at $\bar{M}_M = 0.775$.

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. 3). 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.

3. Dependence of 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

Shown in Fig. 4 test results allow 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 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 $\bar{\omega}_M = 0.875$ ($v_M = 0.350$ m/s).

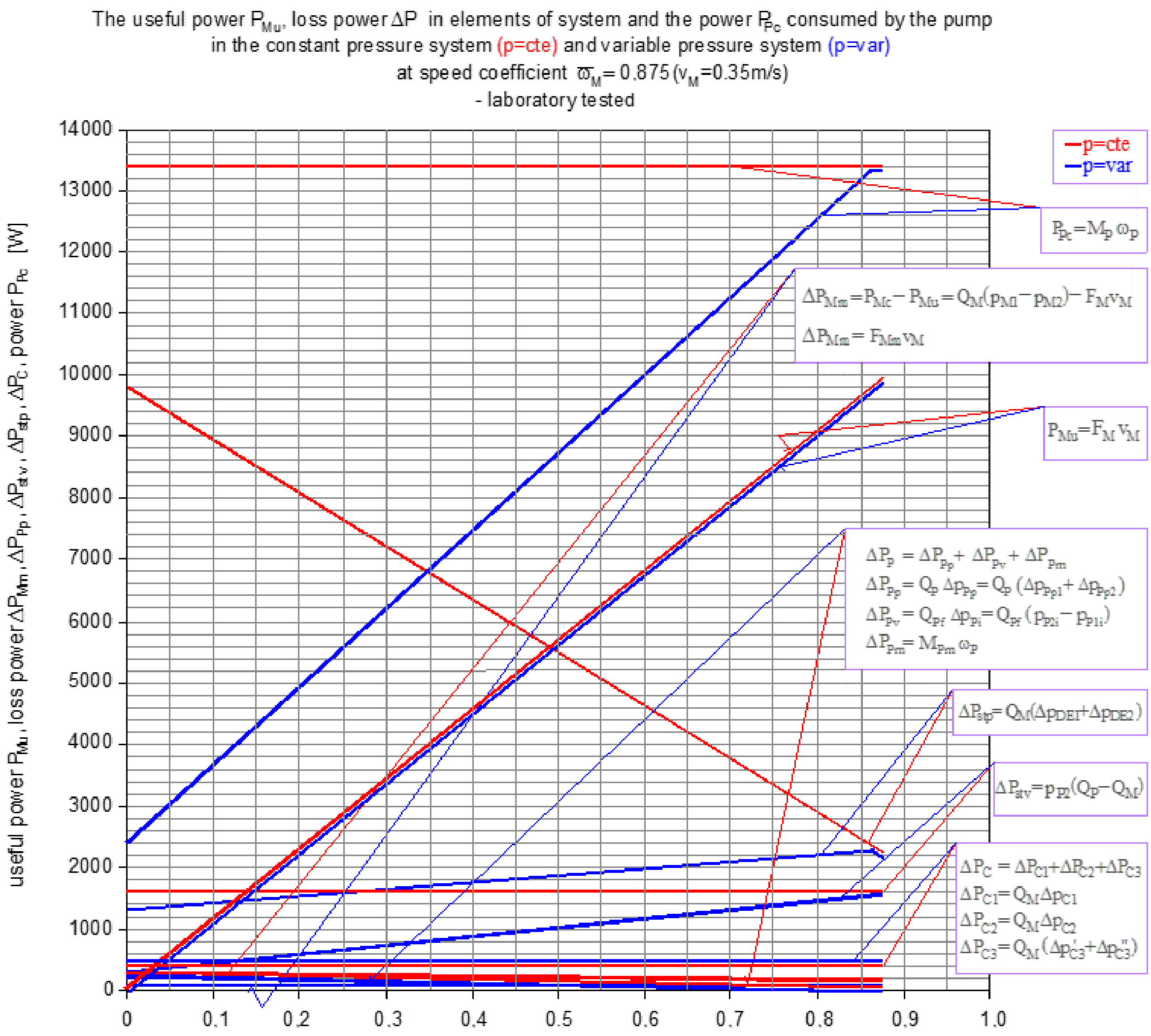

Fig. 4. The loss power ΔP of the system components and the power P_{Pc} demanded by the pump in the constant pressure system (p = cte) and variable pressure system (p = var) from the load coefficient $\bar{M}_M$ at the hydraulic cylinder speed coefficient $\bar{\omega}_M = 0.875$ ($v_M = 0.35$ m/s). The useful power P_{Mu} of the hydraulic cylinder is resulted from the product of the current load F_M ($\bar{M}_M$) and the actual speed v_M ($\bar{\omega}_M$) of the cylinder required by the driven device

From the graph in Fig. 4 results 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 13380 W. 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 3200 W at $\bar{M}_M = 0$ to 13380 W at $\bar{M}_M = 0.875$. The useful power P_{Mu} of the cylinder increases over the whole load coefficient range, is equal to zero at $\bar{M}_M = 0$ and 9900 W at $\bar{M}_M = 0.875$.

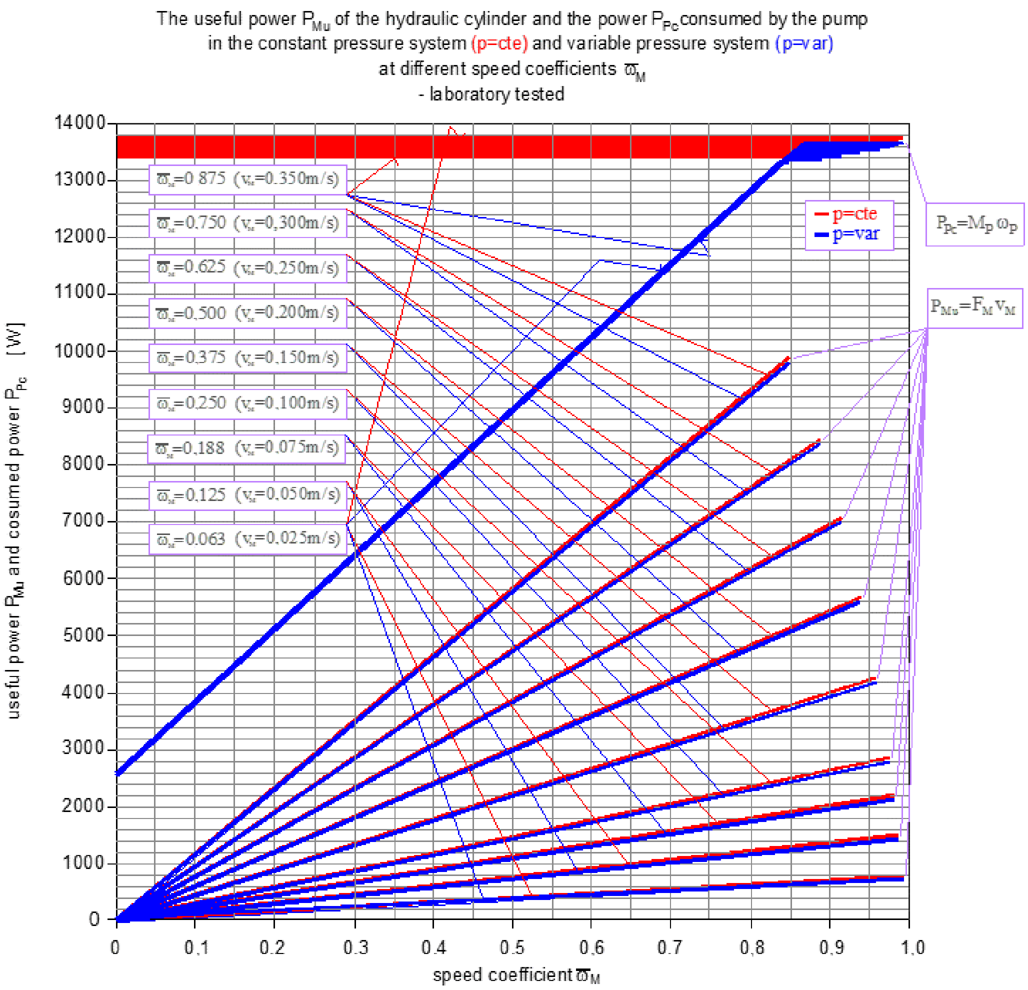

Fig. 5. A dependence of power P_{Pc} demanded by the pump in the constant pressure system (p = cte) and variable pressure system (p = var) from the load coefficient $\bar{M}_M$ at the different speed coefficient $\bar{\omega}_M$ (So this is the dependence the consumed power P_{Pc} from the useful power P_{Mu} of the hydraulic cylinder)

Figure 5 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 $\bar{M}_M$ at different cylinder speed coefficients $\bar{\omega}_M$.

At the smallest speed v_M of the cylinder (v_M = 0.025 m/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 (i.e. loss power in these components). The loss power ΔP, on the other hand, depend 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.

4. Summary and conclusions

This article compares the loss power of the two systems – p = cte and p = var, showing 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. 3).

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