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COMPARISON OF THE STRUCTURAL AND TOTAL ENERGY EFFICIENCY OF SELECTED HYDRAULIC SYSTEMS WITH PROPORTIONAL CONTROL LINEAR MOTOR

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Abstract

In the paper are presented the diagrams of the structural energy efficiency of system with the throttling control assembly and total energy efficiency of the system with constant or variable capacity pump cooperating an overflow valve with the throttling control of the linear hydraulic motor. Diagrams of total energy efficiency of three hydraulic systems working at the same parameters of speed and load of hydraulic linear motor, which were different due to structure and ability of energy saving were presented and compared. This publication also presents analyses and compares the areas of the power fields of energy losses occurring in the elements of three compared hydraulic systems with different structures of the hydraulic linear motor speed control on example on Load Sensing system. The graphical interpretation of the power of losses in the hydrostatic drive and control system elements lets to compare the same power fields of energy losses with other power fields of another structure. This enables to understand what energy losses are the biggest and in which elements of compared hydraulic systems. The best possibility to use in system, as a supply source of the hydraulic cylinder speed series throttling control assembly, is a set consisting of a variable capacity pump cooperating with a Load Sensing (LS) regulator, which totally eliminates the structural volumetric losses in a system. Power ΔP_{stv} of structural volumetric losses is equal to zero, because the current pump capacity Q_P is adjusted, by the LS regulator, to the current flow intensity Q_M set by the throttling assembly.

Keywords: *energy efficiency, power of losses, hydrostatic system, throttling steering, pump, hydraulic linear motor, proportional directional valve*

1. Introduction

In the design of hydrostatic drive systems, the aim is to reduce the loss, reduce energy consumption and thus improve the efficiency. Reduced power losses being converted into heat in the hydraulic systems make less demand for electrical energy or energy fuel for internal combustion engines powering the pump. In order to achieve energy savings in hydrostatic drive systems, it is necessary to select an appropriate control technology.

By joining, the hydrostatic drive design should start from the identification of specific criteria, which limit the design decisions in terms of operating systems, the control method, working parameters and cost. If one accepts the criterion related to energy efficiency for hydrostatic drive with proportional control of the hydraulic motor or cylinder is particularly important is the possibility of using alternatives. The hydrostatic drive systems frequently can reduce the loss of pressure, if the pressure and flow rate will be adjusted to the actual needs of the receiver. Matching to the load pressure of the motor is also possible the use of control valves.

One way of saving is to use a system with a constant capacity pump with an overflow valve controlled pressure to the inlet chamber of the cylinder. Another way is to use the system with load sensing system. It is a system with feedback from the external load acting on the cylinder and automatically adjusting energy demand.

2. Diagrams of compared structures with proportional controlled cylinder

Proportional control of a cylinder consists in throttling the liquid stream at its both inlet and outlet.

The basic proportional control system is a system fed by the constant capacity pump. The overflow valve SP (Fig. 1a) determines the system nominal pressure. The pressure decrease in the cylinder compensates the load on the cylinder. The pump in the $p = \text{const}$ system must generate, before the overflow valve, pressure not lower than pressure required by the cylinder.

The variable pressure ($p = \text{var}$) structure is represented by a system with constant capacity pump cooperating with an overflow valve controlled by the cylinder inlet pressure (Fig. 1b). The variable pressure ($p = \text{var}$) structure with the overflow valve controlled by the current directional valve outflow to cylinder pressure allows to adjust the pump discharge conduit pressure to the current cylinder load, which limits the pressure loss in the working liquid outflow slot from the directional valve to the tank. Additionally, the system maintains constant piston speed irrespective of the load. This is an effect of maintaining practically constant pressure drop Δp_{DE1} in the proportional directional valve-throttling slot.

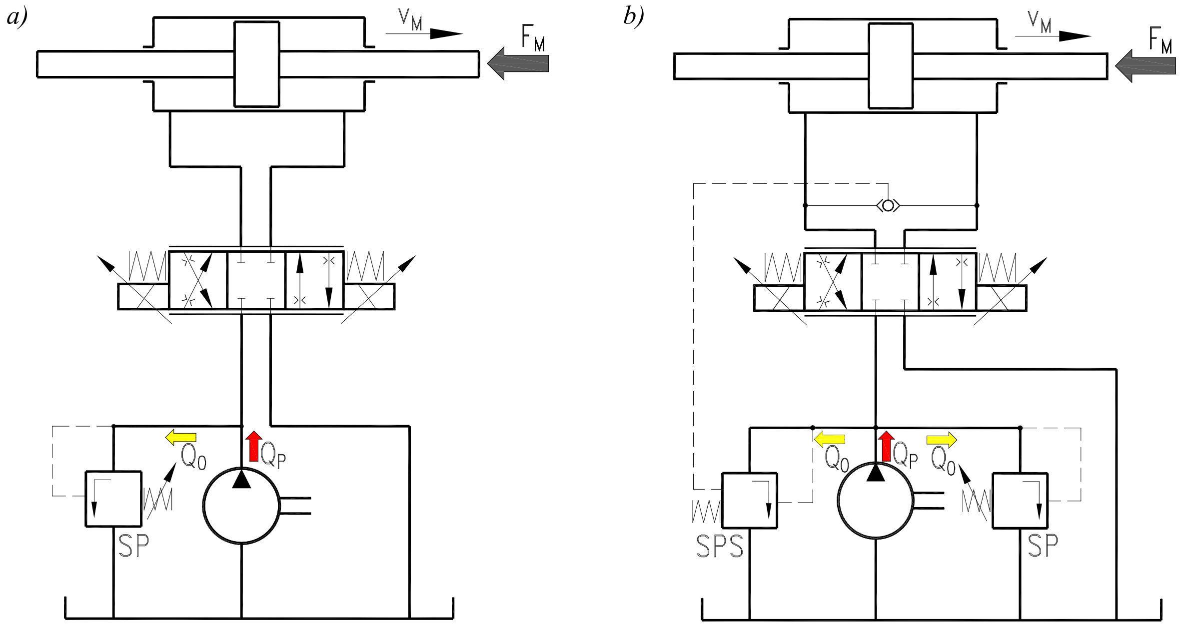

Fig. 1. System with proportional directional valve fed by a constant capacity pump with the use of an overflow valve – p = const structure (on the left) and system with proportional directional valve fed by a constant capacity pump with the use of a hydraulic cylinder supply conduit pressure controlled overflow valve – p = var structure (on the right)

There are another opportunities to reduce energy losses in the elements of the proportional control (pump, a unit with throttling control and hydraulic motor, especially linear cylinder), and thus the possibility of increasing the energy efficiency of the system with the valve throttling.

The use of a variable capacity pump equipped with Load Sensing control system with proportional control (Fig. 2) makes it possible to simultaneously eliminate structural volumetric losses, serious structural reduction of pressure losses, reduce mechanical losses in the linear hydraulic cylinder, and a reduction in mechanical losses and volume in the pump.

The use of a variable displacement pump equipped with a $p = \text{var}$ regulator associated with the high cost of the pump and the regulator should take place after the economic analysis, that the additional investment costs compared with gains that can be achieved during operation of the device.

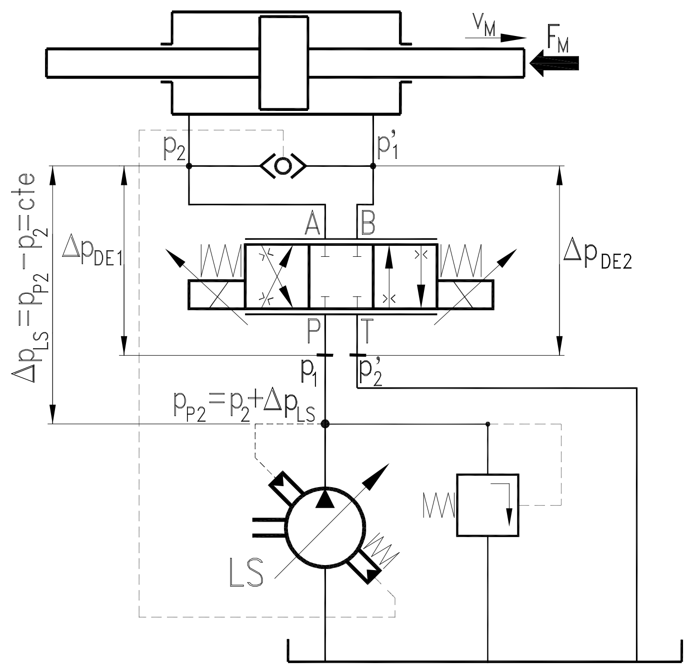

Fig. 2. Individual system with the linear cylinder speed series throttling control fed by a variable capacity pump cooperating with Load Sensing regulator in the variable pressure conditions p_{P2} = var; the throttling control assembly in the form of servo-valve or proportional directional valve

3. Structural energy efficiency of the constant and variable pressure systems

In Fig. 3 there is presented the structural energy efficiency η_{st} that is the energy efficiency of the throttling control unit. The structural energy efficiency is a product of a structural pressure energy efficiency η_{stp} (connected with the proportional valve) and a structural volumetric energy efficiency η_{stv} (connected with the overflow valve):

$$\eta_{st} = \eta_{stp} \cdot \eta_{stv} \,. \tag{1}$$

Figure 3 presents the graph of the structural energy efficiency η_{st} at the chosen coefficients of the hydraulic linear motor's speed $\bar{\omega}_M$.

The structural energy efficiency η_{st} of the constant pressure system p = const assumes, at the cylinder load coefficient of the hydraulic linear motor which equals $\bar{M}_M$ = 0.10 and the speed coefficient which equals $\bar{\omega}_M$ = 0.875 (v_M = 0.350 m/s), the value η_{st} = 0.10. However, the structural energy efficiency η_{st} of the p = var system, at the same coefficients of the cylinder load and speed of the cylinder assumes η_{st} = 0.44. In turn the structural energy efficiency η_{st} of the p = const system assumes, at the cylinder load coefficient $\bar{M}_M$ of the hydraulic linear motor which equals $\bar{M}_M$ = 0.80 and the speed coefficient $\bar{\omega}_M$ which equals $\bar{\omega}_M$ = 0.875 (v_M = 0.350 m/s), the value η_{st} = 0.82. However, the structural energy efficiency η_{st} of a p = var system assumes η_{st} = 0.87, at the same coefficients of the cylinder load and the speed of the cylinder.

To sum up, considerable increase of the structural energy efficiency η_{st} of the p = var system is noticeable at the bigger cylinder speed coefficients $\bar{\omega}_M$ and smaller cylinder load coefficients $\bar{M}_M$. However, at the biggest cylinder load coefficients $\bar{M}_M$ the structural energy efficiency the two of compared systems is equal.

On the basis of the quoted examples can be stated, that by means of application of the variable pressure system p = var, we obtain a considerable increase of the energy efficiency η_{st} at smaller cylinder loads. However, at smaller values of cylinder speed coefficient $\bar{\omega}_M$, the profit connected with using the p = var system is little, mainly because of the volumetric losses, connected with withdrawing the excess of hydraulic oil to the reservoir.

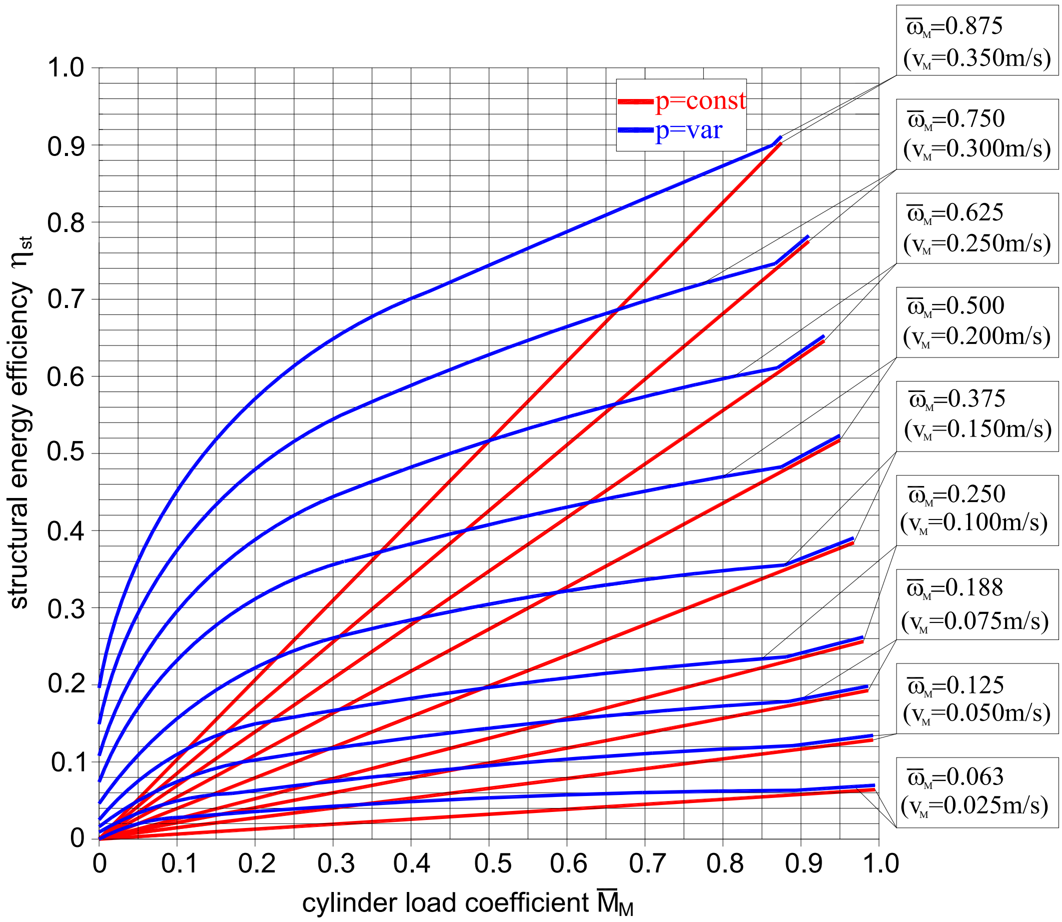

Fig. 3. Dependence of the structural energy efficiency η_{st} of the constant pressure system (p = const) and the variable pressure system (p = var) from the cylinder load coefficient $\bar{M}_M$ at the different cylinder speed coefficients $\bar{\omega}_M$

4. Comparing the energy efficiency of the tested systems

Figure 4 and 5 present the complete energy efficiency η of the constant pressure system (p = const), the variable pressure system (p = var) and the system with the variable capacity pump (Q_P = var) in function of the load coefficient $\bar{M}_M$ at different cylinder speed coefficients $\bar{\omega}_M$.

In case of a system with volumetric control of the variable capacity pump (Q_P = var), enlargement of the cylinder load coefficient $\bar{M}_M$ causes violent increase of the complete energy efficiency η of the system (Fig. 4). However, the energy efficiency of the studied structures with throttling control supplied by the constant capacity pump is at small speed coefficient $\bar{\omega}_M$ clearly lower than energy efficiency with volumetric steering with the same $\bar{\omega}_M$, because the structural losses are so big. increasing the cylinder speed causes proportional growth of the energy efficiency of the p = const and p = var systems, however, at enlargement of the cylinder speed v_M, relative growth of the energy efficiency of the system supplied by the variable capacity pump is smaller (Fig. 4). In Fig. 4 there can be noticed, that 14-time increase of the cylinder speed in studied systems causes about 14-time growth of their energy efficiency. For comparison, 14-time growth of the cylinder speed in the QP=var system causes about 2-time growth of its energy efficiency (from $\eta = 0.39$ at $\bar{\omega}_M = 0.063$ and $\bar{M}_M = 0.875$ to $\eta = 0.78$ at $\bar{\omega}_M = 0.875$ and $\bar{M}_M = 0.875$).

Figure 5 presents graph of the energy efficiency η of the p = const and p = var systems at coefficient $k_{10} = 0.065$ of the proportional valve applied in research and in case of prospective application of a bigger valve – with $k_{10} = 0.010$ and the system with volumetric control of the variable capacity pump (Q_P = var) in function of cylinder load coefficient at speed coefficient $\bar{\omega}_M = 0.939$ ($v_M = 0.380$ m/s) resulted from maximum capacity $Q_{P\max}$ of the pump [2].

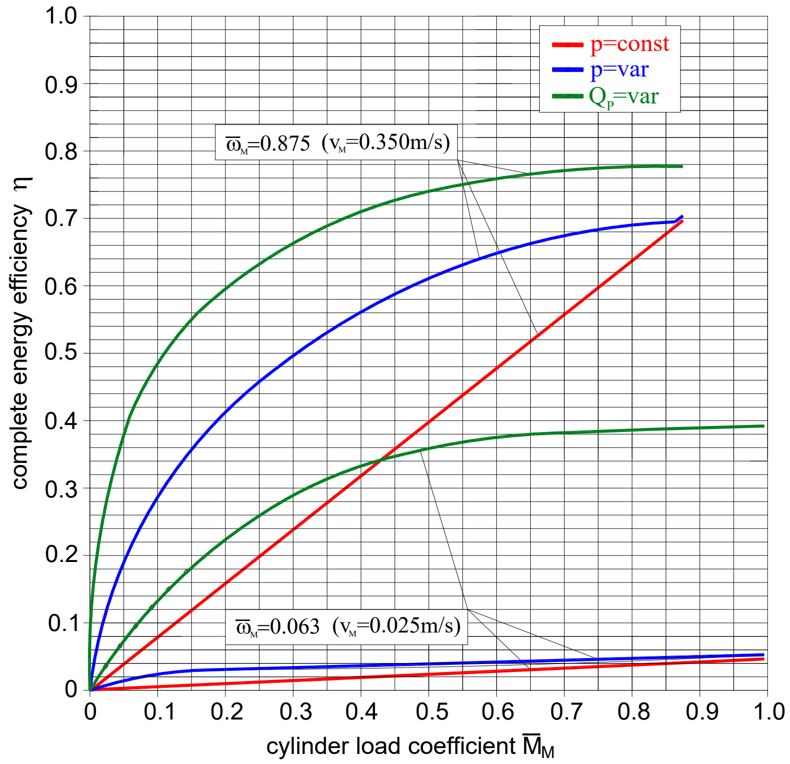

Fig. 4. Dependence of the complete energy efficiency η of the constant pressure system (p = const), the variable pressure system (p = var) and the volumetric control system with the variable capacity pump (Q_P = var) from the cylinder load coefficient $\bar{M}_M$ at the different speed coefficients $\bar{\omega}_M$ (the energy efficiency η of the system described by means of a computer simulation on the basis of laboratory assigned coefficients k_i of the losses in hydraulic elements; the cylinder speed v_M = 0.350 m/s ($\bar{\omega}_M$ = 0.875) was the highest speed of the cylinder realized during researches) [2]

In zone of maximum cylinder speed, so in the zone of using capacity of the pump, the energy efficiency of the p = const and p = var systems with throttling control approaches to the energy efficiency of the Q_P = var system.

5. The fields of power of energy losses in elements of system with the Load Sensing

Figure 6 illustrates the fields of power of energy losses in elements of an individual system with the hydraulic linear motor – cylinder speed series throttling control, fed by a variable capacity pump cooperating with the Load Sensing regulator in a variable pressure p = var system.

The use, as a supply source of the hydraulic cylinder speed series throttling control assembly, of a set consisting of a variable capacity pump cooperating with a Load Sensing (LS) regulator, totally eliminates the structural volumetric losses in a system. Power ΔP_{stv} of structural volumetric losses is equal to zero, because the current pump capacity Q_P is adjusted, by the LS regulator, to the current flow intensity Q_M set by the throttling assembly.

In the hydraulic cylinder speed series throttling control assembly Load Sensing feeding system, the power $\Delta P_{stp} = \Delta p_{DE} Q_M$ of structural pressure losses occurring in the throttling control assembly during loading the hydraulic cylinder with smaller load (force F_M) will be considerably reduced. With an elimination of the power ΔP_{stv} of the structural volumetric losses in the throttling control assembly, the LS system allows to decrease to a negligible value the sum of power ΔP_{st} of structural energy losses resulting from the use of series throttling as a form of precise hydraulic linear motor speed control [3].

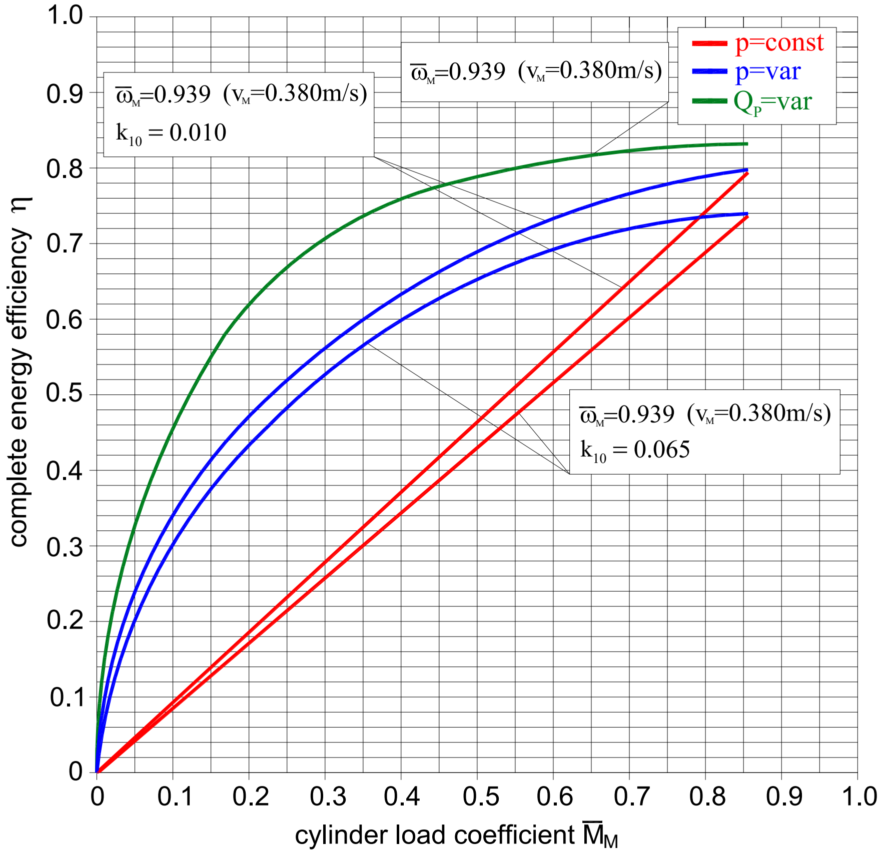

Fig. 5. Dependence of the complete energy efficiency η of the constant pressure system (p = const), the variable pressure system (p = var) at coefficient $k_{10} = 0.065$ of the proportional valve applied in researches and in case of prospective application of the bigger valve – z $k_{10} = 0.010$ and the volumetric control system with the variable capacity pump (Q_P = var) from the cylinder load coefficient $\bar{M}_M$ at the speed coefficient $\bar{\omega}_M$ which equals $\bar{\omega}_M = 0.939$ ($v_M = 0.380$ m/s) resulted from maximum capacity Q_{Pmax} of the pump. The maximum values η_{max} of the three considered systems approach

The use, of a variable capacity pump with Load Sensing regulator reduces the sum of power of energy losses in the system to a value only slightly higher than the sum of power losses in elements of a system with volumetric control of the hydraulic linear motor speed (directly by a variable pump capacity). Power P_{Pc} absorbed by the pump from electric or internal combustion motor is slightly higher than the power P_{Pc} of a variable capacity pump directly driving the hydraulic linear motor.

6. Summary

The hydraulic system is designed first of all taking into consideration the nominal parameters of the cylinder load and speed. For such parameters, the energy efficiency of the elements and complete system is described. Meanwhile the exploitation conditions can vary in full range changes of the cylinder load $\bar{M}_M$ and speed $\bar{\omega}_M$ coefficients.

The studied systems with serial throttling control of cylinder speed, supplied by the constant capacity pump, can achieve, in period of maximum cylinder load $F_{M\max}$ and simultaneous maximum speed $v_{M\max}$ of this cylinder, the same maximum complete energy efficiency $\eta_{\max}$ of the system. The value of this energy efficiency is closed-up to the maximum value of energy efficiency $\eta_{\max}$ of the system with volumetric control of cylinder speed (variable capacity pump). The variable pressure system (p = var) becomes then the constant pressure system (p = const), so work conditions of the two systems become the same and simultaneously there can be practically cut out the structural losses in the throttling control unit.

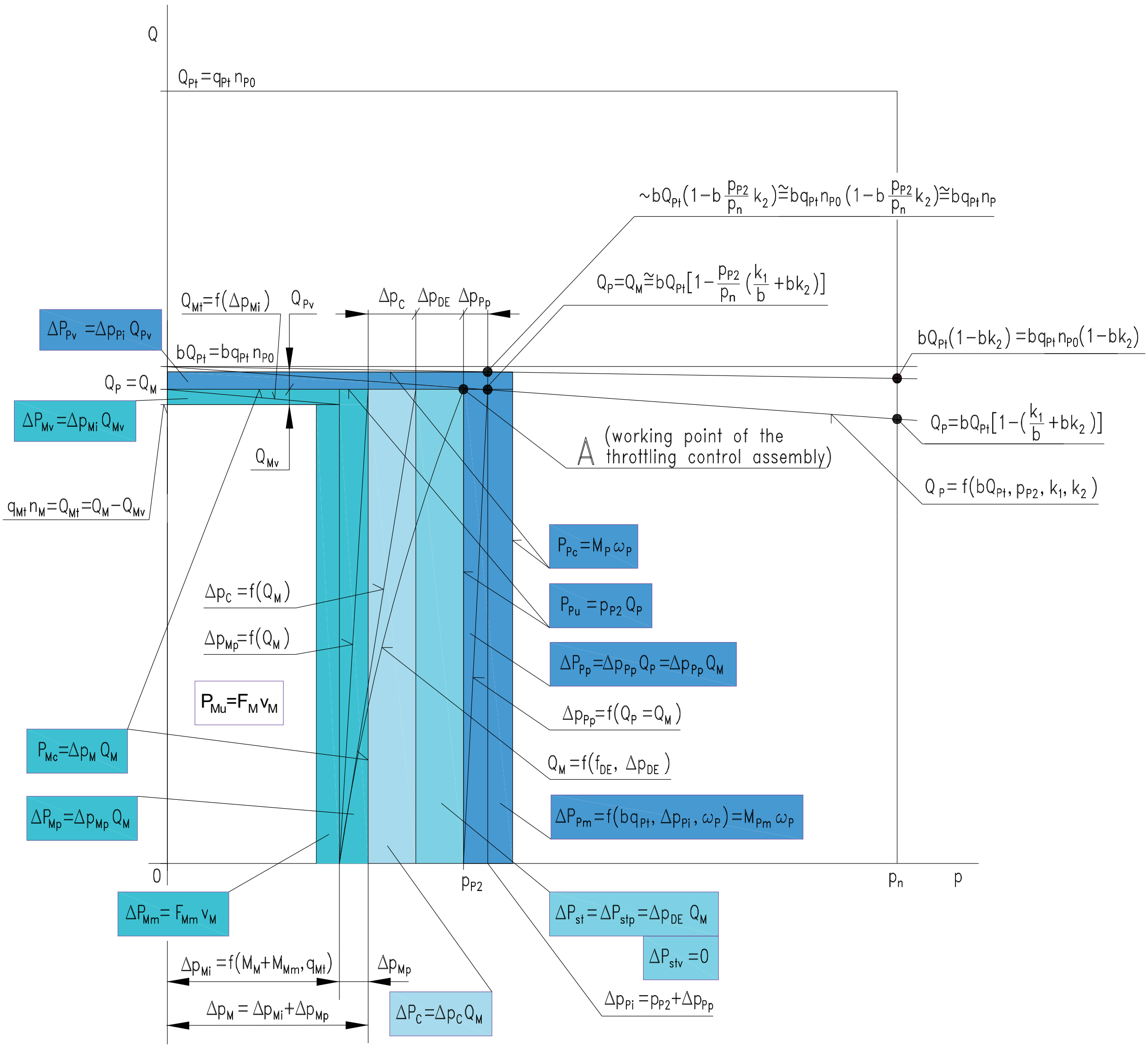

Fig. 6. Graphical interpretation of the power of losses in a hydrostatic drive and control system elements. Individual system with the hydraulic cylinder speed series throttling control fed by a variable capacity pump cooperating with Load Sensing regulator in the variable pressure system: p = var [1]

Primary conclusion resulting from the given examples is the following: maximum possible to achieve values of the energy efficiency are in two different systems, equal. The energy efficiency of the systems with throttling control, supplied by the constant capacity pump is at small cylinder speed coefficient $\overline{\omega}_M$ clearly lower in comparison with the energy efficiency of the system with volumetric control, because the structural losses in these systems are big. The growth of the cylinder speed in the p = const and p = var systems causes proportional increase of the energy efficiency of these systems, whereas much smaller, relative growth of the energy efficiency of the system supplied by the variable capacity pump. For example, 14-time growth of the cylinder speed in studied systems causes about 14-time growth of their energy efficiency (at coefficients $\overline{\omega}_M$ = 0.875 and $\overline{M}_M$ = 0.875). For comparison, 14-time increase of cylinder speed in the Q_P = var system causes about double growth its energy efficiency (at coefficients $\overline{\omega}_M$ = 0.875 and $\overline{M}_M$ = 0.875).

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