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INFLUENCE THE APPLIED CONTROL STRUCTURE ON ENERGY EFFICIENCY OF THE HYDROSTATIC SYSTEM

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Abstract

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 the ship steering gear drive, in the controllable pitch propeller control, in the variable capacity pump control system for hydraulic deck equipment motors or fixed pitch propellers in small ships (for example ferries). 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 and speed coefficients. The article presents a comparison of the energy behaviour of two widespread structures of hydrostatic systems: a standard individual systems with a throttling steering fed by a constant capacity pump. Both hydraulic solutions are described and equations of the total efficiency η of the system are presented. Diagrams of energy efficiency of two hydraulic systems working at the same parameters of a speed and a load of hydraulic linear motor, which were different due to structure are presented and compared, as well ability of energy saving. This publication also presents analyses and compares the areas of the power fields of energy losses occurring in the elements of two hydraulic systems with different structures of the hydraulic linear motor speed control.

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

1. Introduction

The development of hydraulic drive of ship deck machines (also machines used in other industries) is connected with the search for energy-efficient solutions. Examples of applications on ships are the drives of deck crane, of steering machine, and also the main propulsion of small ships.

The energy efficiency of the hydrostatic transmission especially with the throttling steering of the hydraulic motor speed, and energy efficiency of the hydraulic servomechanisms can be higher in real conditions than most often given values in literature of the subject [8]. Possibility of calculating the real complete energy efficiency of the hydraulic system in a function of many parameters deciding about this efficiency becomes an instrument of comprehensive evaluation of the quality of designed system. The possibility such evaluation is essential also for the sake of applying the hydrostatic systems of steering and adjusting in variety of machines and devices, and also for the sake of increasing power of the hydrostatic drive in times of increasing costs of energy production [8].

The system with constant feed pressure achieves high-energy efficiency, equal to the efficiency of the system without the throttling control, only in the points of the maximum $\overline{M}_M$ coefficient and $\overline{\omega}_M$ coefficient of the controlled hydraulic motor or cylinder. The system efficiency decreases rapidly with decreasing motor load and particularly with the simultaneously decreasing motor speed.

There are possibilities of reducing energy losses in the elements of proportional control system (in the pump, in the throttling assembly and in the hydraulic linear motor – cylinder), therefore there are possibilities of increasing the energy efficiency of a directional control valve system [1].

The use of a variable capacity pump with Load Sensing regulator in the proportional control system (Fig. 3) gives a possibility of elimination of structural volumetric losses, significant reduction of the structural pressure losses, reduction of mechanical losses in the linear hydraulic motor – cylinder and also reduction of mechanical and pressure losses in the pump.

In the system with too low energy efficiency, the load increases, mainly of the pump, which causes increased hazard of failure and necessity to repair or exchange it, and also leads to shorter period of exploitation. Too low energy efficiency, resulting most often from intensive throttling of liquid stream, is a source of quick worsening exploitation features, especially grease properties of hydraulic oil, which is the result of too high temperature of work factor – medium of the power of hydrostatic transmission [8].

The mathematical description of the energy behaviour of such a system is presented in [2-4].

The laboratory verification of simulation description of energy efficiency of elements and the system as a whole is presented in doctor dissertation [5]. The work was carried out in the Laboratory Hydraulics and Pneumatics, Faculty of Mechanics, Gdansk University of Technology, and the results are presented in publications [5-9].

2. Description of compared hydraulic systems

The most often used hydraulic rotational or linear motor (cylinder) proportional control system, in the case of proportional directional valve with $\Delta p_{DE1} = \Delta p_{DE2}$ (Fig. 1) is a system where the directional control valve is fed by a constant capacity pump cooperating with an overflow valve stabilizing the feed pressure level (p = const). The overflow valve SP (Fig. 1) determines the system nominal pressure. The pressure decrease in the cylinder compensates the load on the cylinder. The proportional directional valve generates two pressure drops at the cylinder inlet and outlet. The pump in the p = const system must generate, before the overflow valve, pressure not lower than pressure required by the cylinder. Therefore, the hydraulic cylinder or the system-working cylinder may require pressure, depending on the load, in the range from zero to the nominal value. When the load approaches the nominal value, pressure decrease in the directional valve throttling slots tends to zero. It may be said that the pump - overflow valve assembly in the p = const system is ready to feed the system with the maximum pressure and maximum capacity, but most often it is not used to that extent as the working element is loaded with a force that requires pressure drop smaller than the nominal value. A constant pressure system achieves a high-energy efficiency, equal to the efficiency of a system without throttling control, only at the point of maximum values of the controlled hydraulic linear motor load coefficient $\overline{M}_M$ and speed coefficient $\overline{\omega}_M$. The system efficiency η decreases rapidly with decreasing motor load and particularly with simultaneous decreasing motor speed [5, 7, 8].

There are possibilities of decreasing energy losses in elements of the system with proportional control (in the pump, in the throttling steering unit and in the hydraulic motor, particularly in the hydraulic linear motor), so possibilities of increasing the energy efficiency of the system with throttling valve.

The variable pressure (p = var) structure is represented by a system with constant capacity pump cooperating with an overflow valve controlled by the cylinder inlet pressure (Fig. 2). This is an advantageous solution from the viewpoint of the cylinder energy efficiency as well as of the pump and the whole control system efficiency. The variable pressure (p = 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.

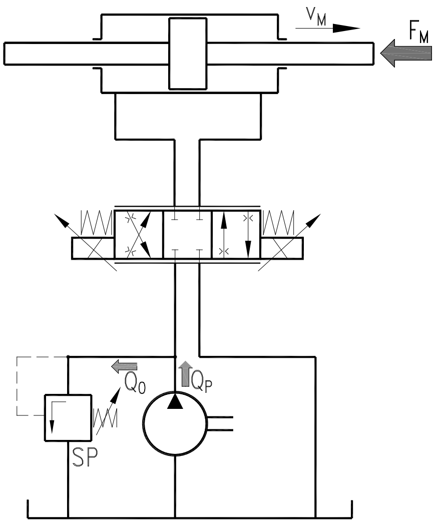

Fig. 1. System with proportional directional control valve fed by a constant capacity pump with the use of an overflow valve in a constant pressure system – p = const

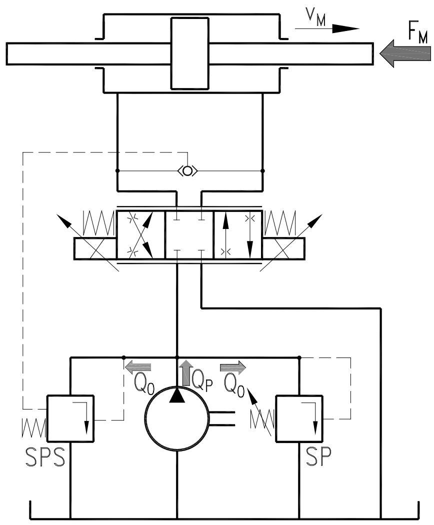

Fig. 2. System with proportional directional control valve fed by a constant capacity pump with the use of a pressure controlled overflow valve in a variable pressure system – p = var

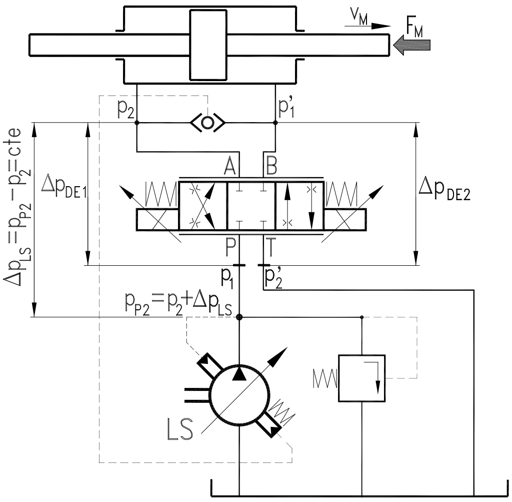

Fig. 3. System with proportional directional control valve fed by a variable capacity pump with Load Sensing regulator – Q_p = var

3. Laboratory investigations

Research of energy efficiency of the hydraulic elements and systems can be classed as basic studies in hydrostatic drive and control systems, taking into consideration detailed analysis of sources for uprising particular energy losses.

The energy efficiency, which is the one of the most important features describing a system, is defined as a proportion of current, demanded by powered device, useful power P_{Mu} of hydraulic motor, to responding its value P_{Mu}, power P_{Pc}, taken by the pump on its shaft from powered electric or combustion engine. In case of improper choice of a hydraulic system type, it can cause increase of hydraulic fluid temperature, so viscosity of the fluid, what in turn causes decrease of energy efficiency of particular elements, what influences motion graphs of the system. That is why energy efficiency can be a decisive factor about possibility of application of a hydraulic system in a particular case. However, detailed analysis of the energy efficiency quite often leads to constructing refinements of different elements of the hydraulic system. However, increasing quality of hydraulic systems cannot be realized solely by improvement of the elements [5, 7, 8].

Figure 4 presents that curves of energy efficiency of researched systems, described in a laboratory and by the computer simulation are very close together. By broken lines are presented curves of energy efficiency η of the system for condition of maximum using by an efficiency system of the pump, that is to say in situation, in which intensity Q_M of stream flown to hydraulic cylinder by the proportional valve is equal the capacity Q_P of the pump. In this case exists possibility of obtaining maximum energy efficiency η of the two systems, which is equal η = 0.746 (at $\overline{M}_M = 0.855$ and $\overline{\omega}_M = 0.939$).

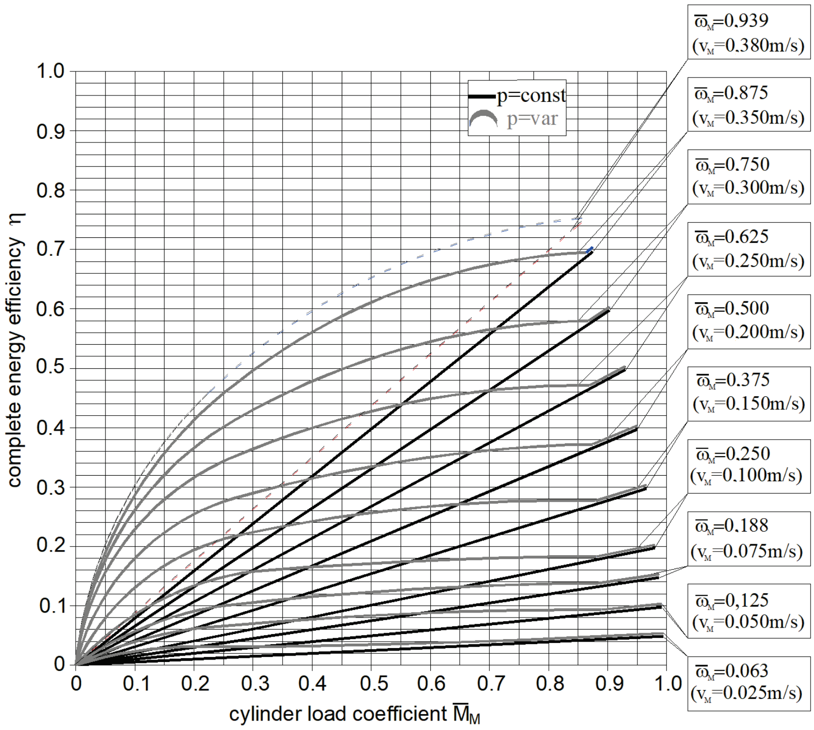

Fig. 4. Dependence of the complete energy efficiency η of the constant pressure system (p=const) and the variable pressure system (p = var) from the cylinder load coefficient $\overline{M}_M$ at the different speed coefficients $\overline{\omega}_M$ the energy efficiency η of the system described by means of a computer simulation on the basis of laboratory assigned coefficients k_i of the losses in hydraulic elements

Using a complete capacity Q_P of the pump is possible then, when an overflow valve SP, used in the p = const and p = var system, would be an ideal valve, that is to say such a valve, which enables work to intensity $Q_0 = Q_P – Q_M$ approaching to zero ($Q_0 \rightarrow 0$).

Due to application of the p = var system, we gain very much at smaller cylinder load F_M and at smaller cylinder speed v_M. In Fig. 4 there can be noticed splendid increasing energy efficiency of the variable pressure system at different cylinder speeds v_M and at different cylinder loads F_M.

As we can see in Fig. 4, from the two of curves, which are lying on the bottom of the graph and regarding to a complete energy efficiency η (that is to say from the graphs at $\overline{\omega}_M = 0.063$ ($v_M = 0.025$m/s)) results, that the energy efficiency η of the both researched systems is small, because of the smallest cylinder speed v_M, at which were studied the p = const and p = var systems, assumes barely 6.3% (0.025 m/s) of the maximum gained cylinder speed.

A ratio of application of the efficiency of the pump assumes in a given case slightly above 6%, however the remaining part of the liquid's stream is flown at the overflow valve SP (SPS) to the reservoir. The cylinder uses in this case a small portion of liquid's stream Q_P, which is generated by the constant pump.

For example, the energy efficiency η of the p = const system, at the cylinder load coefficient $\overline{M}_M$, which equals $\overline{M}_M = 0.50$ and at the speed coefficient which equals $\overline{\omega}_M = 0.063$

(v_M = 0.025 m/s), assumes η = 0.025. However, the energy efficiency η of the p = var system, at the same cylinder load and speed coefficients, is a bit higher and assumes η = 0.034.

However, when we enlarge the cylinder speed v_M, we take intensity of the stream and at the same time smaller intensity of the stream Q_0 flows by the overflow valve SP (SPS) to the reservoir.

In this connection, the energy efficiency η increases. It results the fact that the structural volumetric energy efficiency η_{stv} increases. For example, the complete energy efficiency η of the p = const system, at the same cylinder load coefficient $\overline{M}_M$, as in the previous example and at speed coefficient $\overline{\omega}_M$, which equals $\overline{\omega}_M$ = 0.875 (v_M = 0.350 m/s), assumes η = 0.40. However, the energy efficiency η of the p = var system at the same coefficients of the cylinder load and speed, assumes η = 0.61.

At the cylinder load coefficient $\overline{M}_M$, which equals $\overline{M}_M$ = 0.875, the complete energy efficiency η of both studied systems, at cylinder speed coefficient $\overline{\omega}_M$, which equals $\overline{\omega}_M$ = 0.063 (v_M = 0.025 m/s), assumes only η = 0.045. In turn, the complete energy efficiency η of both systems, at the same cylinder load coefficient $\overline{M}_M$ and at the speed coefficient, which equals $\overline{\omega}_M$ = 0.875 (v_M = 0.350 m/s), achieves the highest value, which equals η = 0.70 [5].

From the point of view of the complete energy efficiency η of the system, the best profit occurs at the cylinder load coefficient $\overline{M}_M$ = 0.2. The complete energy efficiency η of the p = const system assumes then η = 0.165, and the energy efficiency of the p = var system assumes η = 0.43, so it is about 2.2 times higher from the energy efficiency of the constant pressure system. In this zone, the hydraulic systems often work, because then the zone of middle loads begins.

4. Comparing the energy efficiency of three hydraulic systems

Figures 5 and 6 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 $\overline{M}_M$ at different cylinder speed coefficients $\overline{\omega}_M$.

In case of a system with volumetric control of the variable capacity pump (Q_P = var), enlargement of the cylinder load coefficient $\overline{M}_M$ causes violent increase of the complete energy efficiency η of the system (Fig. 5). However, the energy efficiency of the studied structures with throttling control supplied by the constant capacity pump is at small speed coefficient $\overline{\omega}_M$ clearly lower than energy efficiency with volumetric steering with the same $\overline{\omega}_M$, 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. 5). In Fig. 6 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 Q_P = var system causes about 2-time growth of its energy efficiency (from η = 0.39 at $\overline{\omega}_M$ = 0.063 and $\overline{M}_M$ = 0.875 to η = 0.78 at $\overline{\omega}_M$ = 0.875 and $\overline{M}_M$ = 0.875).

Figure 6 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 $\overline{\omega}_M$ = 0.939 (v_M = 0.380 m/s) resulted from maximum capacity Q_{Pmax} of the pump [4].

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.

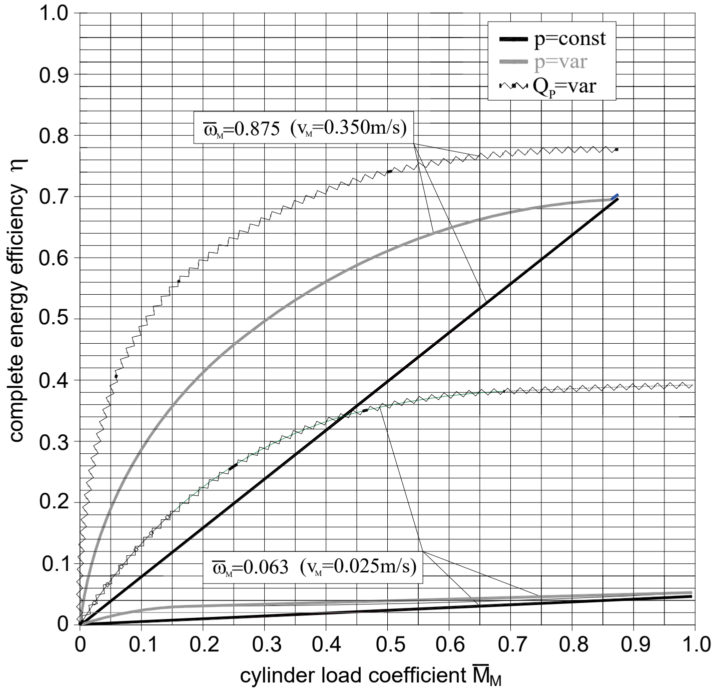

Fig. 5. Dependence of the complete energy efficiency η of the constant pressure system (p = const), the variable pressure system (p = var) and the volumetric control system with the variable capacity pump (Q_P = var) from the cylinder load coefficient $\overline{M}_M$ at the different speed coefficients $\overline{\omega}_M$ (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 ($\overline{\omega}_M$ = 0.875) was the highest speed of the cylinder realized during researches)

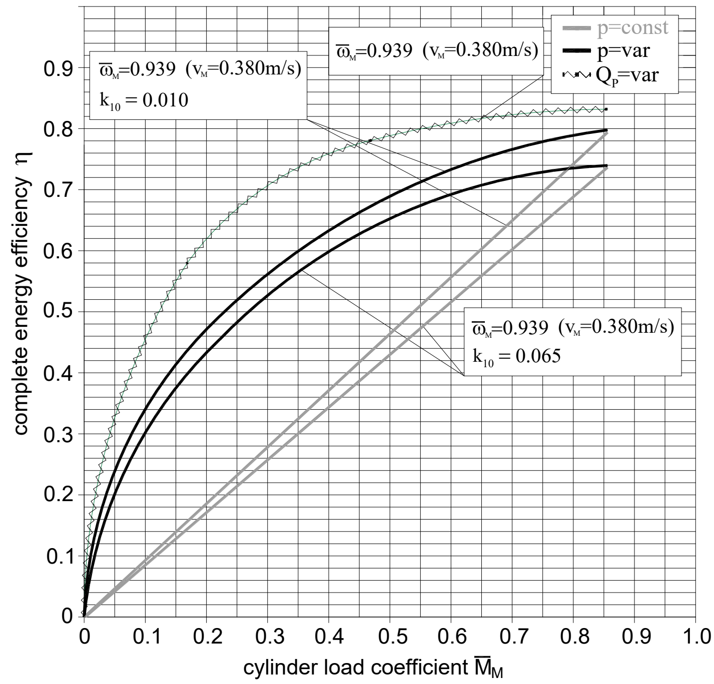

Fig. 6. Dependence of the complete energy efficiency η of the constant pressure system (p = const), the variable pressure system (p = var) at coefficient k_{10} = 0.065 of the proportional valve applied in researches and in case of respective application of the bigger valve – z k_{10} = 0.010 and the volumetric control system with the variable capacity pump (Q_P = var) from the cylinder load coefficient $\overline{M}_M$ at the speed coefficient $\overline{\omega}_M$ which equals $\overline{\omega}_M$ = 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 [5]

5. 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 $\overline{M}_M$ and speed $\overline{\omega}_M$ coefficients [1].

Two studied systems (p = const and p = var) with serial throttling control of cylinder speed, supplied by the constant capacity pump, can achieve, in period of maximum cylinder load F_{Mmax} and simultaneous maximum speed v_{Mmax} of this cylinder, the same maximum complete energy efficiency η_{max} of the system. The value of this energy efficiency is closed-up to the maximum value of energy efficiency η_{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 [5].

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.

Comparison, in Figs. 5 and 6, of the complete energy efficiency η of the studied systems with proportionally controlled cylinder, supplied by the constant capacity pump in the constant pressure system (p = const) and in the variable pressure system (p = var) with the energy efficiency of the system with volumetric control supplied by the variable capacity pump (Q_P = var), shows influence of the cylinder load coefficient $\overline{M}_M$ on increase of the complete energy efficiency η of three hydraulic systems. 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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