



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). Energy savings in a constant capacity pump operation can be achieved by means of overflow valve controlled by the oil outlet pressure between the directional throttling control valve and the cylinder. Although structural volumetric losses cannot be eliminated in such a system, but it is possible to reduce considerably structural pressure losses, mechanical losses and volumetric losses in the pump, and mechanical losses in the cylinder too. The paper discusses these energy savings using an earlier developed by Paszota mathematical model of losses in elements, the energy efficiency of the system and the operating range of the cylinder. The paper also presents a comparison of the energy behavior of two widespread structures of hydrostatic systems: a standard individual systems with a throttling steering fed by a constant capacity pump. Both system 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 and ability of energy saving, were presented and compared.

Keywords: energy efficiency, power of losses, hydrostatic system, throttling steering, hydraulic linear motor

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 (Paszota 2000).

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. 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 (Paszota, 2003).

The system with constant supply pressure achieves high energy efficiency, equal to the efficiency of the system without the throttling control, only in the points of the maximum speed coefficient and load 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 (Paszota 2013).

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 (Paszota, 2004).

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 (Paszota, 2007).

BASIC HYDRAULIC SYSTEM (p=const) AND ENERGY-SAVING SYSTEM WITH A CONSTANT CAPACITY PUMP FED IN VARIABLE PRESSURE (p = var)

The most often used hydraulic rotational or linear motor (cylinder) throttling control system is a system where the proportional directional control valve is fed by a constant capacity pump cooperating with an overflow valve stabilizing the feed pressure level (p=const). 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 and speed coefficient. The efficiency η decreases rapidly with decreasing motor load and particularly with simultaneous decreasing motor speed (Skorek, 2012).

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. 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 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 in the proportional directional valve throttling slot.

STRUCTURAL ENERGY EFFICIENCY OF THE CONSTANT AND VARIABLE PRESSURE SYSTEMS

In Fig. 1 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):

$$\eta_{st} = \eta_{stp} \cdot \eta_{stv} \quad (\text{Paszota 2015}) \quad (1)$$

The energy efficiency η_{st} of the two researched systems can reach high values at the borderline values of the speed coefficient $\bar{\omega}_M$ and the load coefficient \bar{M}_M of the hydraulic cylinder (Fig. 1). In the peak point, the losses connected with an overflow of the hydraulic oil to the reservoir approach to zero (so the structural volumetric energy efficiency η_{stv} reaches its maximum value, which amounts to one), and the losses connected with a pressure drop in the

proportional valve approach to zero (so the structural pressure energy efficiency η_{stp} reaches its maximum value, which amounts to one). In a case of decreasing the load of the cylinder, the energy efficiency of the constant pressure system $p = \text{const}$ decreases linearly, and so the energy efficiency of the variable pressure system $p = \text{var}$ also decreases, but much slower.

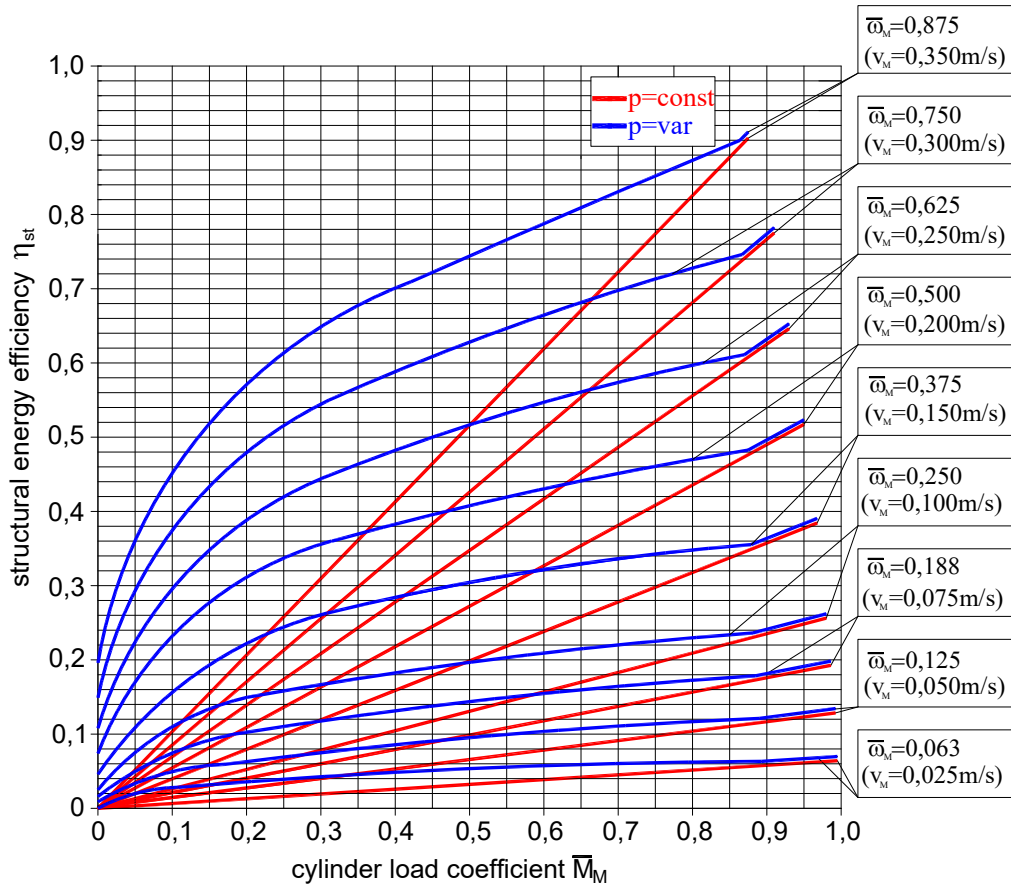


Fig. 1. Dependence of the structural energy efficiency η_{st} of the constant pressure system ($p = \text{const}$) and the variable pressure system ($p = \text{var}$) from the cylinder load coefficient \bar{M}_M at the different cylinder speed coefficients $\bar{\omega}_M$

The structural energy efficiency η_{st} of the constant pressure system $p = \text{const}$ assumes, at the cylinder load coefficient of the hydraulic linear motor which equals $\bar{M}_M = 0.10$ and the speed coefficient which equals $\bar{\omega}_M = 0.875$ ($v_M = 0.350\text{m/s}$), the value $\eta_{st} = 0.10$. However, the structural energy efficiency η_{st} of the $p = \text{var}$ system, at the same coefficients of the cylinder load and speed of the cylinder assumes $\eta_{st} = 0.44$. In turn the structural energy efficiency η_{st} of the $p = \text{const}$ 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 $\bar{\omega}_M = 0.875$ ($v_M = 0.350\text{m/s}$), the value $\eta_{st} = 0.82$. However, the structural energy efficiency η_{st} of a $p = \text{var}$ system assumes $\eta_{st} = 0.87$, at the same coefficients of the cylinder load and the speed of the cylinder (Skorek, 2010).

To sum up, considerable increase of the structural energy efficiency η_{st} of the $p = \text{var}$ 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 $p = \text{var}$, 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 $p = \text{var}$ system is little, mainly because of the volumetric losses, connected with withdrawing the excess of hydraulic oil to the reservoir.

COMPLETE ENERGY EFFICIENCY OF THE SYSTEMS DESCRIBED BY MEANS OF A COMPUTER SIMULATION ON THE BASIS OF LABORATORY ASSIGNED COEFFICIENTS k_i OF THE LOSSES IN HYDRAULIC ELEMENTS

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 (Skorek, 2013).

Fig. 2 presents, that curves of energy efficiency of researched systems, described in a laboratory and by the computer simulation, are very close together.

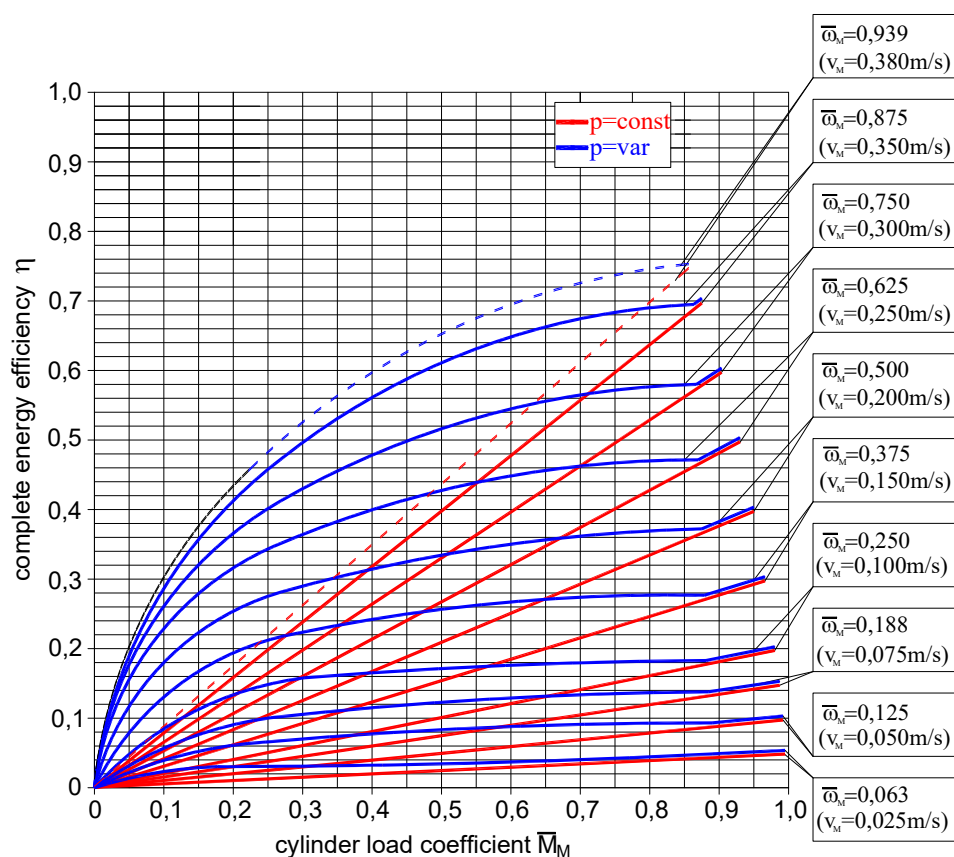


Fig. 2. Dependence of the complete energy efficiency η of the constant pressure system ($p = \text{const}$) and the variable pressure system ($p = \text{var}$) from the cylinder load coefficient \bar{M}_M at the different speed coefficients $\bar{\omega}_M$; the energy efficiency η of the system described by means of a computer simulation on the basis of laboratory assigned coefficients k_i of the losses in hydraulic elements

Source: (Skorek, 2013)

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 $\eta = 0.746$ (at $\bar{M}_M = 0.855$ and $\bar{\omega}_M = 0.939$).

Using a complete capacity Q_P of the pump is possible then, when an overflow valve SP, used in the $p = \text{const}$ and $p = \text{var}$ 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 $p = \text{var}$ system, we gain very much at smaller cylinder load F_M and at smaller cylinder speed v_M . In Fig. 2 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. 2, 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 $\bar{\omega}_M = 0.063$ ($v_M = 0.025\text{m/s}$)) 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 $p = \text{const}$ and $p = \text{var}$ systems, assumes barely 6.3% (0.025m/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 $p = \text{const}$ system, at the cylinder load coefficient \bar{M}_M , which equals $\bar{M}_M = 0.50$ and at the speed coefficient which equals $\bar{\omega}_M = 0.063$ ($v_M = 0.025\text{m/s}$), assumes $\eta = 0.025$. However, the energy efficiency η of the $p = \text{var}$ system, at the same cylinder load and speed coefficients, is a bit higher and assumes $\eta = 0.034$.

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 $p = \text{const}$ system, at the same cylinder load coefficient \bar{M}_M , as in the previous example and at speed coefficient $\bar{\omega}_M$, which equals $\bar{\omega}_M = 0.875$ ($v_M = 0.350\text{m/s}$), assumes $\eta = 0.40$. However, the energy efficiency η of the $p = \text{var}$ system at the same coefficients of the cylinder load and speed, assumes $\eta = 0.61$.

At the cylinder load coefficient \bar{M}_M , which equals $\bar{M}_M = 0.875$, the complete energy efficiency η of both studied systems, at cylinder speed coefficient $\bar{\omega}_M$, which equals $\bar{\omega}_M = 0.063$ ($v_M = 0.025\text{m/s}$), assumes only $\eta = 0.045$. In turn, the complete energy efficiency η of both systems, at the same cylinder load coefficient \bar{M}_M and at the speed coefficient, which equals $\bar{\omega}_M = 0.875$ ($v_M = 0.350\text{m/s}$), achieves the highest value, which equals $\eta = 0.70$ (Skorek, 2013).

From the point of view of the complete energy efficiency η of the system, the best profit occurs at the cylinder load coefficient $\bar{M}_M = 0.2$. The complete energy efficiency η of the $p = \text{const}$ system assumes then $\eta = 0.165$, and the energy efficiency of the $p = \text{var}$ system assumes $\eta = 0.43$, 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.

It is worth saying, that the structural energy efficiency η_{st} (Fig. 1) changes its value in similar way. So it is mainly the structural energy efficiency η_{st} that decides about changing the complete energy efficiency η .

COMPARING THE ENERGY EFFICIENCY OF THE RESEARCHED SYSTEMS PROPORTIONALLY CONTROLLED WITH ENERGY EFFICIENCY OF THE SYSTEM WITH VOLUMETRIC CONTROL OF THE VARIABLE CAPACITY PUMP

Fig. 3 presents the complete energy efficiency η of the constant pressure system ($p = \text{const}$), the variable pressure system ($p = \text{var}$) and the system with the variable capacity pump ($Q_P = \text{var}$) 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 ($Q_P = \text{var}$), enlargement of the cylinder load coefficient \bar{M}_M causes violent increase of the complete energy efficiency η of the system (Fig. 3). 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.

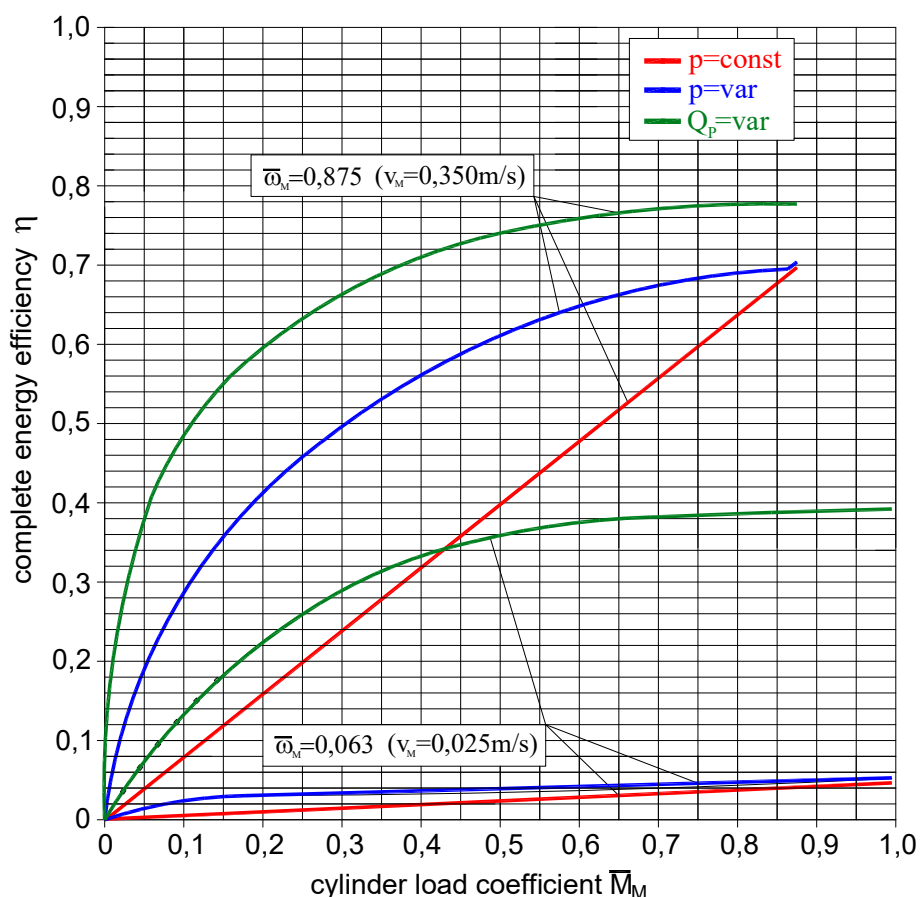


Fig. 3. Dependence of the complete energy efficiency η of the constant pressure system ($p = \text{const}$), the variable pressure system ($p = \text{var}$) and the volumetric control system with the variable capacity pump ($Q_P = \text{var}$) from the cylinder load coefficient \bar{M}_M at the different 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 $v_M = 0.350\text{m/s}$ ($\bar{\omega}_M = 0.875$) was the highest speed of the cylinder realized during researches).

Source: (Skorek, 2013)

Increasing the cylinder speed causes proportional growth of the energy efficiency of the $p = \text{const}$ and $p = \text{var}$ 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. 3). In Fig. 3 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 $Q_P = \text{var}$ 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 $\bar{M}_M = 0.875$).

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 (Paszota, 2009).

Two studied systems ($p = \text{const}$ and $p = \text{var}$) with serial throttling control of cylinder speed, supplied by the constant capacity pump, can achieve, in period of maximum cylinder load $F_{M\text{max}}$ and simultaneous maximum speed $v_{M\text{max}}$ 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 ($p = \text{var}$) becomes then the constant pressure system ($p = \text{const}$), 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 (Skorek, 2013).

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 structural energy efficiency η_{st} of the $p = \text{const}$ and $p = \text{var}$ systems assumes $\eta_{\text{st}} = 0.907$ at the cylinder load coefficient \bar{M}_M , which equals $\bar{M}_M = 0.875$ and at the cylinder speed coefficient $\bar{\omega}_M = 0.875$. Considerable increase of the structural energy efficiency η_{st} of the $p = \text{var}$ system is noticeable at 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 efficiencies η_{st} of the two of compared systems are equal. Due to application of the variable pressure system $p = \text{var}$, there is achieved, at smaller cylinder loads, considerable increase of the structural energy efficiency η_{st} . At small values of the cylinder speed coefficient $\bar{\omega}_M$, the profit connected with application of the $p = \text{var}$ system is small, mainly because of volumetric losses, connected with throwing out an overflow of hydraulic oil to the reservoir.

Comparison 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 ($p = \text{const}$) and in the variable pressure system ($p = \text{var}$) with the energy efficiency of the system with volumetric control supplied by the variable capacity pump ($Q_P = \text{var}$), shows influence of the cylinder load coefficient \bar{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 $\bar{\omega}_M$ 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 $p = \text{const}$ and $p = \text{var}$ 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 $\bar{\omega}_M = 0.875$ and $\bar{M}_M = 0.875$). For comparison, 14-time increase of cylinder speed in the $Q_P = \text{var}$ system causes about double growth its energy efficiency (at coefficients $\bar{\omega}_M = 0.875$ and $\bar{M}_M = 0.875$).

REFERENCES

- Paszota Z. (2000). Energy efficiency of hydraulic cylinders in servo-mechanism. Conference proceedings. Naval Arch. Marine Eng. 2000, pp. 139-147.
- Paszota Z. (2003). Energy aspects of hydrostatic drives. Polish Maritime Research, Vol. 10, 2/2003, pp. 18-20.
- Paszota Z. (2004). An Energy Behaviour Comparison of two Kinds of Hydrostatic Drive of Ship Deck Machines. Brodogradnja – Journal of Naval Architecture and Shipbuilding Industry, Brodarski Institut Zagreb, Vol. 52, 3/2004, pp. 213-222.
- Paszota Z. (2007). Energy saving in a hydraulic servomechanism system – theory and examples of laboratory verification. Brodogradnja, 58/2007, pp. 146-157.
- Paszota Z. (2009). The operating field of a hydrostatic drive system parameters of the energy efficiency investigations of pumps and hydraulic motors. Polish Maritime Research, 4 (62) 2009, Vol. 16; pp. 16-21.
- Paszota Z. (2013). Losses and energy efficiency of drive motors and systems. Replacement of the Sankey diagram of power decrease in the direction of power flow by a diagram of power increase opposite to the direction of power flow opens a new perspective of research of drive motors and systems. Polish Maritime Research, Vol.20, No 1 (77), 2013, pp. 3-10.
- Paszota Z. (2015). Energy losses in hydrostatic drive. Monography, LAP Lambert, Academic Publishing.
- Skorek G. (2010). Energy characteristics of the hydraulic system with proportional control of cylinder, fed by a constant capacity pump in a constant pressure and variable pressure system (in Polish). Doctor dissertation. Gdansk University of Technology, Faculty of Ocean Engineering and Ship Technology.
- Skorek G. (2012). Graphical interpretation of the power of energy losses and power developed in the hydrostatic drive and control system elements. Polish Maritime Research, 4/2012, pp. 44-53.
- Skorek G. (2013). Energy efficiency of a hydrostatic drive with proportional control compared with volumetric control. Polish Maritime Research, 3/2013, pp. 14-19.

Date of submission of the article to the Editor: 06/2018

Date of acceptance of the article by the Editor: 08/2018