POLISH MARITIME RESEARCH 4(62) 2009 Vol 16; pp. 16-21
10.2478/v10012-008-0051-3

The operating field of a hydrostatic drive system parameters of the energy efficiency investigations of pumps and hydraulic motors

Zygmunt Paszota, Prof.
Gdansk University of Technology

ABSTRACT

The operating field of hydrostatic drive system is presented. Subdivision of the hydraulic motor and pump work parameters into parameters independent of and dependent on the operation of displacement machines and the system is justified. A research project is proposed aimed at development of methods determining the energy characteristics of pumps and rotational hydraulic motors as well as modified methods of determining the energy characteristics of hydrostatic drive systems with selected structures of hydraulic motor speed control. The paper is an extended version of reference [11].

Keywords: hydrostatic drive and control; pumps; hydraulic motors; Sankey diagram; energy efficiency; laboratory and simulation investigations

INTRODUCTION

The working machines with hydrostatic drive and control and also with hydraulic servomechanisms are, due to their drive and control characteristics, very popular in the world-wide economy. However, designers of those machines, manufacturers of the pumps, hydraulic motors and hydrostatic drive systems used in the machines do not have at their disposal a simulation tool for precise determination of their energy behaviour in the entire field of the change of parameters and operating conditions. That makes it impossible to evaluate the drive energy efficiency at an arbitrary point of the operating field and to search for energy saving solutions.

The existing state of affairs is an effect of common views on the way of conducting of laboratory tests of pumps and hydraulic motors used in the hydrostatic drives and control systems and on the method of determining the energy losses in them. That method is an outcome of the traditional reading the energy balance of a hydrostatic drive system illustrated by the Sankey diagram.

The Sankey diagram informs, in relation to the energy balance of a drive system, that the width of the input power stream is equal to the combined width of the output power streams.

In the case of the pump, hydraulic motor and hydrostatic drive system energy balance, the Sankey diagram suggests a situation where the power stream of energy losses should be deducted from the input power stream. **However**, in the case of a hydrostatic drive system, in its operating field (Fig. 1), i.e. in the hydraulic motor $0 \leq \overline{\omega}_M < \overline{\omega}_{Mmax}$ range of speed coefficient and the $0 \leq \overline{M}_M < \overline{M}_{Mmax}$ range of load coefficient, **the power stream of energy losses should be added to the output power stream, because it is the output (and not input) power parameters that decide about the power of losses**.

Only the upper limits $\overline{\omega}_{Mmax}$ and $\overline{M}_{Mmax}$, describing the hydraulic motor operating field in the hydrostatic drive system (limits influencing the maximum achievable drive output power), are defined by the maximum pump working parameters. They are a result of the theoretical (maximum in the variable capacity pump) capacity q_{Pt} per one pump shaft revolution and of the pump shaft rotational speed n_P (dependent in turn on the pump driving electric motor or internal combustion engine characteristics) and also of the nominal pressure p_n of the system (maximum pressure of the continuous system operation) determined in the pump discharge conduit. Besides, they depend on the used structure of the hydraulic motor speed control and on the losses in the system elements. Therefore, the upper limits of hydraulic motor operating field are different in systems with different motor speed control structures.

The simulation methods of determining, by means of mathematical models and computer programs, of the power of losses, energy efficiency and operation field of the hydrostatic drive and control systems of machines are a way to the precise qualitative and quantitative evaluation of the energy behaviour of those systems. The key question is proper selection of the parameters deciding about the system energy behaviour.

The simulation methods allow to investigate and compare the energy saving system solutions and also precise and simple evaluation of the quality of used pumps and hydraulic motors, motor speed control structures, impact of the viscosity and type of working liquid (hydraulic oil, oil-water emulsion, water).

The computer simulation of hydrostatic drive energy behaviour is limited because there are no laboratory methods for determination of pump and hydraulic motor energy characteristics allowing to calculate the respective energy loss coefficients. It is possible only in the situation when the energy losses (and the loss coefficients describing them) are determined as a function of the parameters that they directly depend on.

In a hydrostatic drive system, the power in the power stream increases in the opposite direction to the direction of power stream because of the need to overcome the power of energy losses. Presentation of the power of energy losses in the system must be constructed in the direction from the rotational hydraulic motor shaft or linear motor piston rod to the pump shaft.

In the references [1 – 4] it is shown that **the so far practiced presentation of the results of laboratory tests of rotational hydraulic motor losses and energy efficiency as a function of parameters dependent on those losses should be abandoned**.

References [5 – 10] present and analyse the areas of power of energy losses in the hydraulic system elements with different rotational hydraulic motor speed control structures. Those considerations allow to understand the roles deciding of the power of losses corresponding to the current motor work parameters required by the hydraulic motor driven device, i.e. the motor load M_M and speed ω_M (n_M). Those considerations allow to draw conclusions about the conditions of achieving high energy efficiency η of a selected structure system. It is also possible to compare the power of losses due to the used hydraulic motor speed control structure. The power P_{Pc} consumed by the pump from its driving electric (or internal combustion) motor is the power necessary for providing the useful power $P_{Mu} = M_M \omega_M$ required of the hydraulic motor by the driven device.

In references [7] and [9] **a diagram is proposed and explained presenting the direction of increase of power stream flowing from the pump shaft to the hydraulic motor shaft or piston rod, but increasing from the hydraulic motor shaft or piston rod to the pump shaft**. **The increase of power in the power stream is caused by the necessity of compensating the power of energy losses in the hydrostatic drive and control system elements**. The pump shaft power is a function (sum) of the hydraulic motor shaft or piston rod power and power of losses in the system elements. **The proposed diagram replaces the Sankey diagram**.

INDEPENDENT AND DEPENDENT PARAMETERS OF HYDRAULIC MOTOR AND PUMP OPERATION

The work of a rotational or linear hydraulic motor as an element of hydrostatic drive and control system, directly connected with the driven machine (device) must provide parameters required by the driven machine (speed ω_M (n_M) of the shaft or v_M of the piston rod and shaft load M_M or piston rod load F_M) and also ensure the required machine movement direction.

The mechanical parameters of a motor (speed ω_M (n_M) or v_M and also load M_M or F_M) change in the range from zero to maximum values ω_{Mmax} (n_{Mmax}) or v_{Mmax}, M_{Mmax} or F_{Mmax}.

The required current speed ω_M (n_M) or v_M and required current load M_M or F_M of the driven machine are an effect of its work cycle and the work task. The current driven machine speed and load values are independent of the type and structure of that machine driving system control (e.g. an electrical or hydrostatic system).

The current speed and current load of a hydrostatic system driven machine have a direct or indirect impact on the mechanical, volumetric and pressure losses in the hydraulic motor, pump and other system elements, a system with determined motor speed control structure. The losses are also an effect of the viscosity or kind of the used working liquid (hydraulic oil, oil-water emulsion, water).

The current speed ω_M (n_M) or v_M and current load M_M or F_M of the driven machine influence, in consequence, **the current hydraulic motor absorption capacity Q_M and pressure decrease Δp_M and also (depending on the used motor speed control structure) the current pump capacity Q_P and discharge pressure p_{P2}.**

If in effect of the increasing, required by the driven machine (device) hydraulic motor speed ω_M (n_M) or v_M, as well as in effect of the increasing, required by the driven machine motor load M_M or F_M, and also in effect of the mechanical, volumetric and pressure losses of the hydrostatic drive system elements, the maximum drive system capacity (determined by the maximum pump capacity Q_{Pmax} or maximum pump discharge conduit pressure p_{P2max} limited to the system nominal pressure p_n) is fully used, then further increase of ω_M (n_M) or v_M as well as M_M or F_M will not be possible.

Maximum pump capacity Q_{Pmax} is smaller than its theoretical capacity Q_{Pt}. The pump theoretical capacity Q_{Pt} is a product of the theoretical capacity q_{Pt} per one pump shaft revolution and the unloaded pump shaft speed n_{P0}. The pump Q_{Pmax} capacity, however, results from the loaded pump speed n_P, lower than the speed n_{P0}, and from volumetric losses in the pump.

The system nominal pressure p_n is a maximum permissible continuous operation pressure p_{P2max} determined in the pump discharge conduit.

The maximum speed values ω_{Mmax} (n_{Mmax}) or v_{Mmax} as well as the maximum load values M_{Mmax} or F_{Mmax} of the hydraulic motor used in a hydrostatic drive system are limited by the maximum pump capacity Q_{Pmax} or by the system (pump) nominal pressure p_n and also by the corresponding mechanical, volumetric and pressure losses in the remaining system elements, the losses being also an effect of viscosity or the kind of working liquid used. **Therefore, the ω_{Mmax} (n_{Mmax}) or v_{Mmax}, M_{Mmax} or F_{Mmax} values are dependent variables**.

The current mechanical operating parameters of the hydraulic motor used in a hydrostatic drive system (current motor speed ω_M (n_M) or v_M and current motor load M_M or F_M) **are independent values in the motor**, deciding of the motor losses and of the hydraulic parameters (the current motor absorbing capacity Q_M and current pressure decrease Δp_M also depending on the motor mechanical, volumetric and pressure losses). **The current motor absorbing capacity Q_M and current pressure decrease Δp_M are dependent variables in the motor**.

In the hydraulic motor (hydrostatic drive system) operating field ($0 \le \omega_M$ (n_M) $< \omega_{Mmax}$ (n_{Mmax}), $0 \le M_M < M_{Mmax}$) or ($0 \le v_M < v_{Mmax}$, $0 \le F_M < F_{Mmax}$), the pressure and flow intensities in the system and also the energy losses in the motor, in the pump and in the whole system, power of energy losses and energy efficiencies of the system elements should be considered the functions of the current speed ω_M (n_M) or v_M and the current load M_M or F_M required by the system driven machine (device). Also **the torque M_P that the pump loads the driving** (electric or internal combustion) **motor and the speed n_P that the motor drives the pump with should be considered the functions of the current speed and the current load required by the system driven machine**.

The decrease of speed n_P that the electric or internal combustion motor drives the pump with is connected with the increase of torque M_P that the pump loads the motor with. The decrease of speed depends on the operating characteristics of the motor, which is not a component of the hydrostatic drive system. Therefore, **the pump driving speed n_P should be treated as a parameter independent of the system (of the pump)**.

NON-DIMENTIONAL COEFFICIENTS OF THE HYDRAULIC MOTOR PARAMETERS, COEFFICIENTS OF ENERGY LOSSES IN THE SYSTEM ELEMENTS

The energy efficiency of the hydraulic drive system and its elements is described by mathematical models as functions of the hydraulic motor hydrostatic drive system speed coefficient $\overline{\omega}_M$ and hydraulic load coefficient $\overline{M}_M$.

The current angular speed ω_M (rotational speed n_M) required of a rotational motor or the linear speed v_M required of a linear motor, operating in a hydrostatic drive system, **are replaced** in the energy efficiency mathematical models **by the motor speed non-dimentional coefficient** $\overline{\omega}_M$:

$$\overline{\omega}_M = \frac{\omega_M}{\omega_{Mt}} = \frac{n_M}{n_{Mt}} = \frac{\omega_M\, q_{Mt}}{2\Pi\, Q_{Pt}} = \frac{n_M\, q_{Mt}}{Q_{Pt}}$$

or

$$\overline{\omega}_M = \frac{v_M}{v_{Mt}} = \frac{v_M\, S_{M1}}{Q_{Pt}}$$

The rotational hydraulic motor speed coefficient $\overline{\omega}_M$ is a ratio of the current angular speed ω_M (rotational speed n_M), required of the motor by driven machine, to:

theoretical angular speed

$$\omega_{Mt} = \frac{2\Pi Q_{Pt}}{q_{Mt}}$$

theoretical rotational speed

$$n_{Mt} = \frac{Q_{Pt}}{q_{Mt}}$$

which would correspond with the theoretical capacity Q_{Pt} of the motor driving pump and with the theoretical motor absorbing capacity q_{Mt} per one shaft revolution. The speed ω_{Mt} (n_{Mt}) would be achievable on the condition, that there are no volumetric losses in the hydrostatic drive system (including the pump and the hydraulic motor) and the pump is driven by an (electric or internal combustion) motor operating with constant rotational speed $n_P = n_{P0}$ independent of its load.

The theoretical angular speed ω_{Mt} (rotational speed n_{Mt}) of a rotational motor is treated as a constant reference value for the motor current angular speed ω_M (rotational speed n_M).

The linear hydraulic motor speed coefficient $\overline{\omega}_M$ is a ratio of the current linear speed v_M, required of the motor by driven machine, to:

theoretical linear speed

$$v_{Mt} = \frac{Q_{Pt}}{S_{M1}}$$

which would correspond with the theoretical capacity Q_{Pt} of the motor driving pump and with effective area S_{M1} of the motor piston in the inlet chamber. The speed v_{Mt} would be achievable on the condition, that there are no volumetric losses in the hydrostatic drive system (including the pump and the hydraulic motor) and the pump is driven by an (electric or internal combustion) motor operating with constant rotational speed $n_P = n_{P0}$ independent of its load.

The theoretical linear speed v_{Mt} of a linear motor is treated as a constant reference value for the current motor linear speed v_M.

The current torque M_M required of a rotational motor or current force F_M required of a linear motor, operating in a hydrostatic drive system, **are replaced by the motor load non-dimentional coefficient** $\overline{M}_M$:

$$\overline{M}_M = \frac{M_M}{M_{Mt}} = \frac{2\Pi\, M_M}{q_{Mt} p_n}$$

or

$$\overline{M}_M = \frac{F_M}{F_{Mt}} = \frac{F_M}{S_{M1} p_n}$$

The rotational hydraulic motor load coefficient $\overline{M}_M$ is a ratio of the current torque M_M, required of the motor by driven machine, to:

theoretical torque

$$M_{Mt} = \frac{q_{Mt}\, p_n}{2\Pi}$$

which would correspond with the theoretical absorbing capacity q_{Mt} per one motor shaft revolution and with the hydrostatic system nominal pressure p_n. The torque M_{Mt} would be achievable on the condition that there are no mechanical or pressure losses in the hydraulic motor and in the remaining system elements (except the pump) and the pressure p_{P2max} in the pump discharge conduit is equal to the system nominal pressure p_n.

The rotational motor theoretical torque M_{Mt} is treated as a constant reference value for the current motor torque M_M.

The linear hydraulic motor load coefficient $\overline{M}_M$ is a ratio of the current force F_M, required of the motor by driven machine, to:

theoretical force

$$F_{Mt} = S_{M1}\, p_n$$

which would correspond with the effective area S_{M1} of the motor piston in its inlet chamber and with the system nominal pressure p_n. The force F_{Mt} would be achievable on the condition that there are no mechanical or pressure losses in the hydraulic motor and in the remaining system elements (except the pump) and the pressure p_{P2max} in the pump discharge conduit is equal to the system nominal pressure p_n.

The linear motor theoretical force F_{Mt} is treated as a constant reference value for the current motor force F_M.

The mechanical, volumetric and pressure losses in a hydraulic motor, pump and in the remaining hydrostatic drive system elements are described in the mathematical models of the losses, power of losses and energy efficiency by the coefficients k_i – relations to the values connected with the values of the hydrostatic drive system characteristic parameters:

– theoretical capacity q_{Pt} per one pump shaft revolution
– theoretical absorbing capacity q_{Mt} per one rotational hydraulic motor revolution or effective piston area S_{M1} in the linear motor inlet chamber
– theoretical pump capacity Q_{Pt}
– system nominal pressure p_n.

The basis of energy evaluation of the particular design solutions and size of the volumetric machines is a catalogue of the coefficients k_i of energy losses in various types of pumps and hydraulic motors used in the hydrostatic drive systems, operating with different levels of pump theoretical capacity Q_{Pt} and system nominal pressure p_n, with the working liquid reference viscosity ν_n.

THE MOTOR OPERATING FIELD IN A HYDROSTATIC DRIVE SYSTEM

Figure 1 presents the **operating field of a rotational or linear hydraulic motor in a hydrostatic drive system**. The operating field is determined in the plane of motor mechanical parameters, i.e. speed coefficient $\overline{\varpi}_M$ and load coefficient $\overline{M}_M$, independent of the motor and of the system.

The limit values $\overline{\varpi}_{Mmax} = f(\overline{M}_M)$ or $\overline{M}_{Mmax} = f(\overline{\varpi}_M)$ of the hydraulic motor operating field are determined by the maximum motor feed capability in the hydrostatic drive system. The values $\overline{\varpi}_{Mmax}$ and $\overline{M}_{Mmax}$ are dependent on the motor and on the system losses.

The maximum motor absorbing capacity Q_{Mmax}, achieved in the system by the applied motor speed control structure, should be equal or close to the instantaneous maximum pump capacity Q_{Pmax} (resulting from the theoretical capacity Q_{Pt}, decrease of the pump shaft rotational speed n_p and the intensity of pump volumetric losses Q_{Pv}).

The maximum possible motor pressure decrease Δp_{Mmax} should be equal or close to the system nominal pressure p_n determined in the pump discharge conduit, reduced by the pressure losses Δp_C in the system conduit. (In the motor series throttling speed control structure, the maximum slot area of the throttling valve, proportional directional valve or servo-valve should allow to minimize the pressure decrease $\Delta p_{DE|Q_{Mmax}}$ with the set $Q_{Mmax} \approx Q_{Pmax}$).

Therefore, the limit values $\overline{\varpi}_{Mmax}$ of the hydraulic motor speed coefficient are a function of the current motor load coefficient $\overline{M}_M$, coefficients k_i of the volumetric losses in the hydrostatic system elements (including coefficient k_2 of the pump shaft rotational speed decrease Δn_p) and a function of the ratio of working liquid viscosity ν to the reference viscosity ν_n – ν/ν_n.

On the other hand, the limit values $\overline{M}_{Mmax}$ of the hydraulic motor load coefficient are a function of the current motor speed coefficient $\overline{\varpi}_M$, coefficients k_i of the mechanical and pressure losses in hydrostatic system elements and a function of the ratio of working liquid viscosity ν to the reference viscosity ν_n.

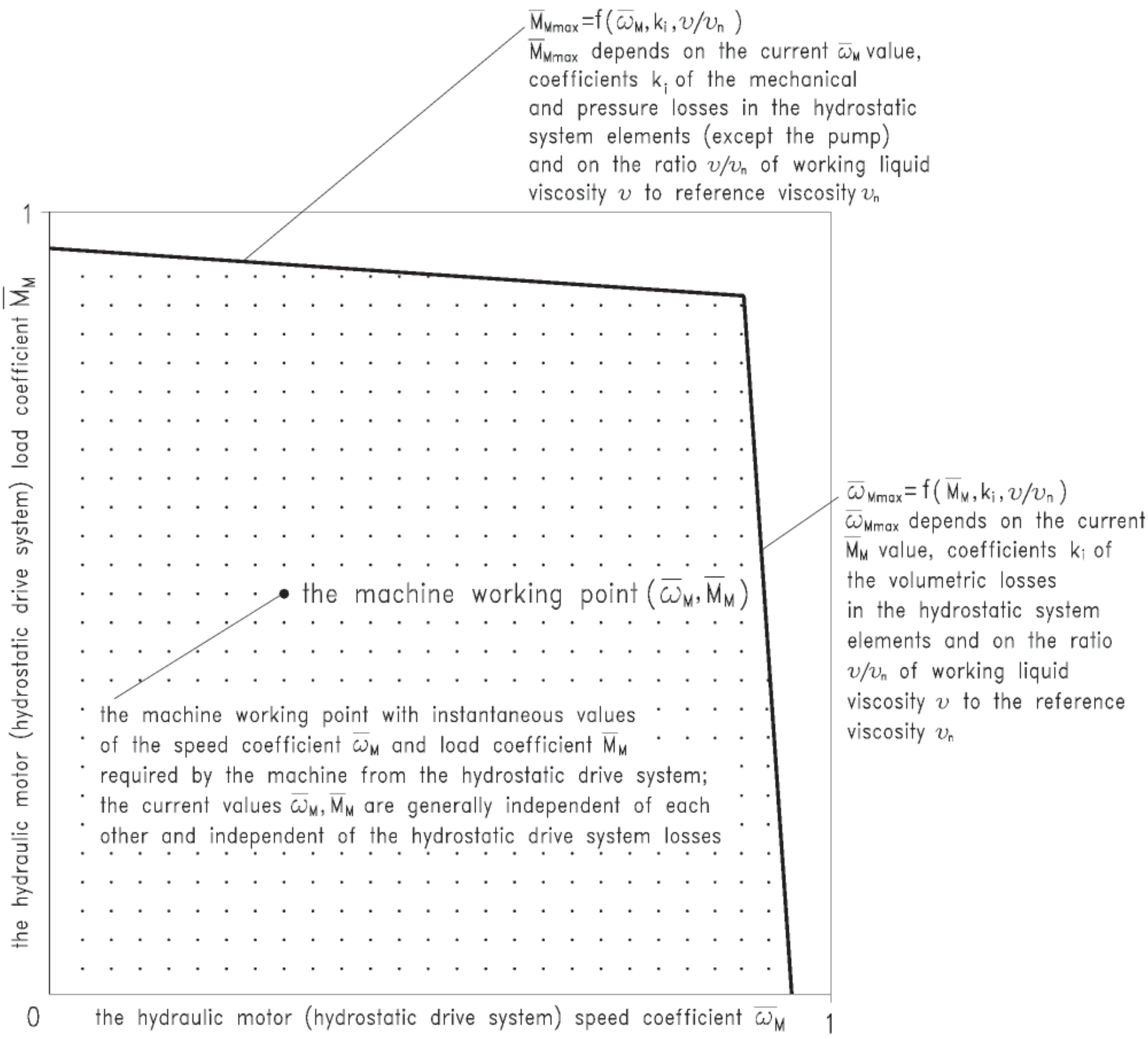

***Fig. 1.** The range of hydraulic motor speed coefficient $\overline{\varpi}_M$ and load coefficient $\overline{M}_M$ ($0 \le \overline{\varpi}_M < \overline{\varpi}_{Mmax}$, $0 \le \overline{M}_M < \overline{M}_{Mmax}$) in a machine hydrostatic drive and control system*

THE INVESTIGATIONS OF THE HYDRAULIC MOTOR AND PUMP IN A DRIVE SYSTEM

In a hydraulic motor and hydrostatic drive system operating field ($0 \le \overline{\omega}_M < \overline{\omega}_{Mmax}$, $0 \le \overline{M}_M < \overline{M}_{Mmax}$), the losses, power of losses and energy efficiency of the various motor versions, operating:

- in a drive system with different speed control structures
- with different pump theoretical capacity Q_{Pt}
- with different system nominal pressures p_n
- with different working liquid viscosity ν,

should be determined (and compared) as a function of mechanical parameters required by the driven machine, i.e. as a function of the motor speed coefficient $\overline{\omega}_M$ and load coefficient $\overline{M}_M$ and also as a function of the ratio of working liquid viscosity ν to the reference viscosity ν_n. The motor operating field ($0 \le \overline{\omega}_M < \overline{\omega}_{Mmax}$, $0 \le \overline{M}_M < \overline{M}_{Mmax}$) or the function $\overline{\omega}_{Mmax} = f(\overline{M}_M)$, $\overline{M}_{Mmax} = f(\overline{\omega}_M)$ should also be determined (and compared).

In a hydraulic motor and hydrostatic drive system operating field ($0 \le \overline{\omega}_M < \overline{\omega}_{Mmax}$, $0 \le \overline{M}_M < \overline{M}_{Mmax}$), the losses, power of losses and energy efficiency of the various pump versions, with constant or variable capacity, operating:

- in a drive system with different speed control structures,
- with different pump theoretical capacity Q_{Pt}
- with different system nominal pressures p_n
- with different working liquid viscosity ν,

should be determined (and compared) as a function of the pump capacity coefficient $\overline{Q}_P = Q_P/Q_{Pt}$ and a function of the pump discharge pressure coefficient $\overline{p}_{P2} = p_{P2}/p_n$ resulting from the current values of mechanical parameters required by the hydrostatic system driven machine, i.e. the hydraulic motor speed coefficient $\overline{\omega}_M$ and load coefficient $\overline{M}_M$ and also as a function of the ratio of working liquid viscosity ν to the reference viscosity ν_n. The current pump operation coefficients $\overline{Q}_P$ and $\overline{p}_{P2}$ are determined by the current hydraulic motor coefficients $\overline{\omega}_M$ and $\overline{M}_M$ and also by the energy losses in the motor, in the conduits and the losses resulting from the used motor speed control structure.

THE INVESTIGATIONS OF A PUMP IN THE CONDITIONS INDEPENDENT OF THE DRIVE SYSTEM (INVESTIGATIONS OF AN ISOLATED PUMP)

Investigations of the losses, power of losses and energy efficiency of a pump, together with evaluation of the coefficients k_i of the pump losses, are carried out in the conditions independent of the hydrostatic drive system.

- **During the investigation of a constant capacity (q_{Pt} = cte) pump** (an isolated pump investigated independently of a drive system)**, the losses, power of losses and energy efficiency should be evaluated** (**and compared**) **as a function of the discharge pressure coefficient $\overline{p}_{P2} = p_{P2}/p_n$, changing in the $0 \le \overline{p}_{P2} \le 1$ range**. The pump capacity coefficient $\overline{Q}_P$ should be also evaluated (and compared) as a function of the $\overline{p}_{P2}$ and the ratio ν/ν_n of the working liquid viscosity ν to the reference viscosity ν_n.
  In the constant capacity pump (q_{Pt} = cte), **the current discharge conduit pressure p_{P2} is an independent value**, deciding of the pump losses and the current pump capacity Q_P and of the pump shaft torque M_P. Also **the pump driving motor speed n_P**, deciding of the current pump capacity Q_P, **is a value independent of the pump**.
  In a constant capacity pump (q_{Pt} = cte), **the current pump capacity Q_P and the pump shaft torque M_P are dependent values in the pump**.
- **During the investigation of a variable capacity pump** ($0 \le q_{Pgv} \le q_{Pt}$) (an isolated pump investigated independently of a drive system)**, the losses, power of losses and energy efficiency should be evaluated as a function of the pump capacity coefficient $\overline{Q}_P = Q_P/Q_{Pt}$ and as a function of the pump discharge pressure coefficient $\overline{p}_{P2} = p_{P2}/p_n$, changing in the $0 \le \overline{p}_{P2} \le 1$ range**. The pump maximum capacity $\overline{Q}_{Pmax}$ coefficient should be also evaluated (and compared) as a function of the $\overline{p}_{P2}$ coefficient and of the ratio ν/ν_n of the working liquid viscosity ν to the reference viscosity ν_n.
  In a variable capacity pump ($0 \le q_{Pgv} \le q_{Pt}$)**, the current pump capacity Q_P and the discharge conduit pressure p_{P2} are independent values** deciding of the pump losses, the current capacity q_{Pgv} per one pump shaft revolution (the current $b_P = q_{Pgv}/q_{Pt}$ coefficient) and of the pump shaft torque M_P. **Also the pump driving motor speed n_P is a value independent of the pump**.
  In a variable capacity pump, the current pump capacity q_{Pgv} per one pump shaft revolution (the current $b_P = q_{Pgv}/q_{Pt}$ coefficient) **and the pump shaft torque M_P are dependent values in the pump**.

CONCLUSIONS

- The diagram proposed in [7] and [9], presenting the direction of increase of the power stream in a hydrostatic drive system (replacing the Sankey diagram), the power flowing from the pump shaft to the hydraulic motor shaft or piston rod, but increasing from the hydraulic motor shaft or piston rod to the pump shaft, makes one understand the subdivision of the hydraulic motor and pump work parameters into the parameters independent of and dependent on the energy losses in those machines.
- The graphical presentation of subdivision of the hydraulic motor and the system (in consequence, also the pump) work parameters into the parameters independent of and dependent on the hydrostatic drive system losses is the system operating field presented in the hydraulic motor speed coefficient $\overline{\omega}_M$ and load coefficient $\overline{M}_M$ coordinates (Fig. 1).
- Mathematical models, describing the system losses and energy efficiency as a function of parameters independent of the losses, should be based on the defined k_i coefficients of the hydraulic motor, pump and conduits, and also of the motor speed throttling control assembly, mechanical, volumetric and pressure losses.
- The values of k_i coefficients of the hydraulic motor and pump mechanical, volumetric and pressure losses should be a basis for evaluation of the motor and pump design solutions. The values of k_i coefficients of the losses should be determined nominal pressure p_n levels of the system where the hydraulic motor and pump are used and also in the full range of the working liquid kinematic viscosity.
- The simulation determination of the pump, rotational hydraulic motor and hydrostatic drive system energy efficiency will be enabled by development of the laboratory methods of determination of the pump and hydraulic motor energy characteristics as well as of the modified methods of determination of energy characteristics of the hydrostatic drive systems with selected motor speed control structures. That simulation technique of determining the energy efficiency will be a work tool of the drive system designer.

It will allow to search for energy saving solutions of pumps and hydraulic motors and also for the analysis of energy saving hydrostatic drive and control systems.

- It is suggested to open a research project aimed at considerable reduction of the scope of pump, rotational hydraulic motor and hydrostatic drive system energy efficiency laboratory investigations. The aim is to be achieved by replacing the so far executed full laboratory investigations by simpler tests determining only the k_i coefficients of losses in the system elements. The coefficients k_i are then used in the mathematical models and in simulation computer programs for the pump, hydraulic motor and hydrostatic drive system energy efficiency calculations. The mathematical models should take into account and describe the physical phenomena and power of losses, determine the pump, hydraulic motor and hydrostatic drive system operating field and the energy efficiency at any point of that field. They also should allow to evaluate the relative value of power of losses in the pump, hydraulic motor or drive system during the unloaded pump, unloaded hydraulic motor or unloaded drive system operation.

BIBLIOGRAPHY

1. Paszota Z.: *Power of energy losses in the hydrostatic drive system elements – definitions, relations, range of changes, energy efficiencies*. *Part I – Hydraulic motor*. Chapter in the monograph: *„Research, design, production and operation of hydraulic systems”* (in Polish), Andrzej Meder and Adam Klich editors. „Cylinder” Library. Komag Mining Mechanisation Centre, Gliwice 2007
2. Paszota Z.: *Power of energy losses in the hydrostatic drive system elements – definitions, relations, range of changes, energy efficiencies*. Part II – Conduits, throttling control assembly, pump. Chapter in the monograph: *„Research, design, production and operation of hydraulic systems”* (in Polish), Andrzej Meder and Adam Klich editors. „Cylinder” Library. Komag Mining Mechanisation Centre, Gliwice 2007
3. Paszota Z.: *Power of energy losses in the hydrostatic drive system elements – definitions, relations, range of changes, energy efficiencies. Part I – Hydraulic motor*. (in Polish) Napędy i sterowanie, scientific monthly, No 11 (103), November 2007
4. Paszota Z.: *Power of energy losses in the hydrostatic drive system elements – definitions, relations, range of changes, energy efficiencies*. *Part II – Conduits, throttling control assembly, pump* (in Polish), Napędy i sterowanie, scientific monthly, No 12 (104), December 2007
5. Paszota Z.: *Graphical presentation of the power of energy losses and power developed in the elements of hydrostatic drive and control system*. *Part I – Rotational hydraulic motor speed series throttling control systems*. Chapter in the monograph: *„Research, design, production and operation of hydraulic systems”* (in Polish), Adam Klich, Edward Palczak and Andrzej Meder editors. „Cylinder” Library. Komag Mining Mechanisation Centre, Gliwice 2008
6. Paszota Z.: *Graphical presentation of the power of energy losses and power developed in the elements of hydrostatic drive and control system*. *Part II – Rotational hydraulic motor speed parallel throttling control and volumetric control systems*. Chapter in the monograph: *„Research, design, production and operation of hydraulic systems”* (in Polish), Adam Klich, Edward Palczak and Andrzej Meder editors. „Cylinder” Library. Komag Mining Mechanisation Centre, Gliwice 2008
7. Paszota Z.: *Direction of increase of power stream in the hydrostatic drive and control system*. *Graphical presentation of the power of energy losses and power developed in the elements of hydrostatic drive and control system*. *Part I – Rotational hydraulic motor speed series throttling control systems*. (in Polish), Napędy i sterowanie, scientific monthly, No 10 (114), October 2008
8. Paszota Z.: *Direction of increase of power stream in the hydrostatic drive and control system*. *Graphical presentation of the power of energy losses and power developed in the elements of hydrostatic drive and control system*. *Part II – Rotational hydraulic motor speed parallel throttling control and volumetric control systems*. (in Polish), Napędy i sterowanie, scientific monthly, No 11 (115), November 2008
9. Paszota Z.: *Graphical presentation of the power of energy losses and power developed in the elements of hydrostatic drive and control system*. *Part I – Rotational hydraulic motor speed series throttling control systems*. Polish Maritime Research 03/2008
10. Paszota Z.: *Graphical presentation of the power of energy losses and power developed in the elements of hydrostatic drive and control system*. *Part II – Rotational hydraulic motor speed parallel throttling control and volumetric control systems*. Polish Maritime Research 04/2008
11. Paszota Z.: *The operating field of a hydrostatic drive system*. Chapter in the monograph: *„Research, design, production and operation of hydraulic systems”* (in Polish), Adam Klich, Antoni Kozieł and Edward Palczak editors. „Cylinder” Library. Komag Mining Mechanisation Centre, Gliwice 2009

CONTACT WITH THE AUTHOR

Prof. Zygmunt Paszota
Faculty of Ocean Engineering
and Ship Technology
Gdansk University of Technology
Narutowicza 11/12
80-233 Gdansk, POLAND
e-mail: zpaszota@pg.gda.pl