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DETERMINATION OF COEFFICIENTS OF ENERGY LOSSES OCCURRING IN A CONSTANT CAPACITY PUMP WORKING IN A TYPICAL HYDROSTATIC DRIVE

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Abstract

In order to assess possibilities of energy saving during hydrostatic drive system operation, should be learned, and described losses occurring in system. Awareness of proportion of energy, volume, pressure, and mechanical losses in elements is essential for improving functionality and quality of hydrostatic drive systems characterized by unquestionable advantages. In systems with too low efficiency there is increase of load, mainly in case of pump load, which can lead to higher risk of failure, necessity of repair or replacement, as well as to shorten service life of system. Coefficients k_i, given in subject literature by Paszota, describe relative value of individual losses in element. They make it possible to assess proportions of losses and assess value of energy efficiency (volumetric, pressure, mechanical) resulting from losses occurring at nominal pressure p_n of system in which element is used. As a result, thanks to knowledge of coefficients k_i of individual losses, it is possible to determine losses and energy efficiency of components operating in hydraulic system as well as efficiency of system with defined structure of motor speed control as function of speed and load coefficient of motor. Knowledge of coefficients of energy losses occurring in system elements (pump, hydraulic motor, conduits, and motor) allows building models of losses and energy efficiency of element working in system and energy efficiency of system as whole composed of elements. Mathematical models of losses and energy efficiency in system must take into account conditions resulting from applied structure of system, from level of nominal pressure, from rotational speed of motor driving pump shaft, from viscosity change of applied hydraulic oil. Article presents method of determining coefficients of axial piston pump used in typical hydrostatic drive system with proportional control. Values that can be assumed for these loss coefficients for other hydraulic pumps are also given.

Keywords: *coefficients of losses, energy losses, hydrostatic system, axial piston pump*

1. Introduction

Hydrostatic systems play very important role in modern machines. A large number of currently constructed machines have more or less developed hydrostatic or electro-hydraulic drive systems and in many cases, these systems are the most important parts of machines. Components – hydraulic linear motors (cylinders) – are widely used in machines used on land and on ships. The undoubted advantages of the cylinders are reciprocating motion, reliability, simple construction, effective power to weight ratio. The control system with a servo control valve or a proportional control valve, controlling cylinder, is used, among others, in the ship's rudder drive, propeller blade control, inboard equipment drive, or fixed pitch propeller on small vessels (e.g. ferries).

There are unverified areas related to the behaviour of components in hydraulic systems of various constructions. Awareness of the proportion of energy, volume, pressure, and mechanical losses in elements is essential for improving the functionality and quality of hydrostatic drive systems characterized by unquestionable advantages. Energy efficiency of hydrostatic transmissions, especially those with throttling control of the hydraulic speed of a linear hydraulic motor, as well as the efficiency of the hydraulic system of the servomechanism can actually be higher than the values most often quoted in publications on this subject. Currently, for example, the efficiency of a servo-drive system is still often presented incorrectly. Namely, the maximum efficiency of such a system (with the ideal pump and hydraulic motor) is given as equal to η=0.385 at the motor supply pressure equal to p=2/3pmax. This approach leads to the use of more than the required power unit does – pump, work at lower efficiency, increase in the cost of the system itself and its operating costs. In systems with too low efficiency, there is an increase in load, mainly in the case of pump load, which can lead to a higher risk of failure, necessity of repair or replacement, as well as to shorten the service life of the system. Too low efficiency of the system resulting most often from intensive throttling of liquid flow is also a source of rapid deterioration of performance, mainly lubricating properties of hydraulic oil, due to too high operating temperature of hydrostatic transmissions. The ability to calculate the actual value of the overall efficiency of the hydraulic system, as a function of many parameters influencing it, becomes a tool for the full assessment of the quality of the system.

2. Description of the compared systems and components used on the test stand

The research concerned two systems with a directional proportional control valve fed by a constant-capacity pump:

a) using an overflow valve – a constant pressure structure – p=cte,
b) using a pressure-operated overflow valve from the cylinder's inlet conduit – variable pressure structure – p=var (Fig. 1).

The most common control system for a proportional linear hydraulic motor is the system, in which the proportional directional control valve is supplied with a constant capacity pump cooperating with an overflow valve stabilizing the constant supply pressure level p=cte.

It can be stated that the pump and overflow valve assembly in the p=cte system is a unit ready to supply the system at maximum pressure and maximum efficiency. However, it is not usually used to the extent that the cylinder at a given moment is loaded with a force that requires a pressure drop below the nominal pressure.

This system obtains a high-energy efficiency, equal to the efficiency of the system without a throttling control, only at the point with the maximum values of the load coefficient and the motor speed coefficient. When the cylinder load decreases, especially when the motor speed drops, the efficiency η of the system decreases sharply [3].

The hydraulic system of the drive and proportional control of the linear hydraulic motor can be supplied by a constant capacity pump cooperating with an overflow valve stabilizing the pressure of the proportional control valve at the nominal pressure level or a pump cooperating with a pressure control overflow valve at the inlet to the receiver. The variable pressure system p=var (Fig. 1) enables to reduce losses in the pump, in the control unit and in the hydraulic linear motor [3].

In the variable pressure system p=var, the structural pressure and volume losses in the throttling control unit can be severely reduced, the mechanical losses in the cylinder and the pump as well as volumetric losses in the pump. The mathematical description of losses and efficiency is presented in the items [4].

The variable pressure structure p=var represents a system with a constant capacity pump cooperating with an overflow valve controlled by the cylinder supply pressure (Fig. 1). A solution is beneficial from the point of view of energy efficiency of the cylinder as well as the pump and the entire control system. The variable pressure structure p=var with the SPS overflow control valve with the current control valve discharge pressure to the inlet chamber of the cylinder allows to adjust the pressure level in the pump discharge outline to the cylinder load, so that it reduces the pressure loss in the control valve liquid outflow into the tank. In addition, this system maintains a constant piston speed independent of the load. This is the effect of maintaining a practically constant pressure drop Δp_{DE1} in the throttling gap of the proportional directional control valve [3].

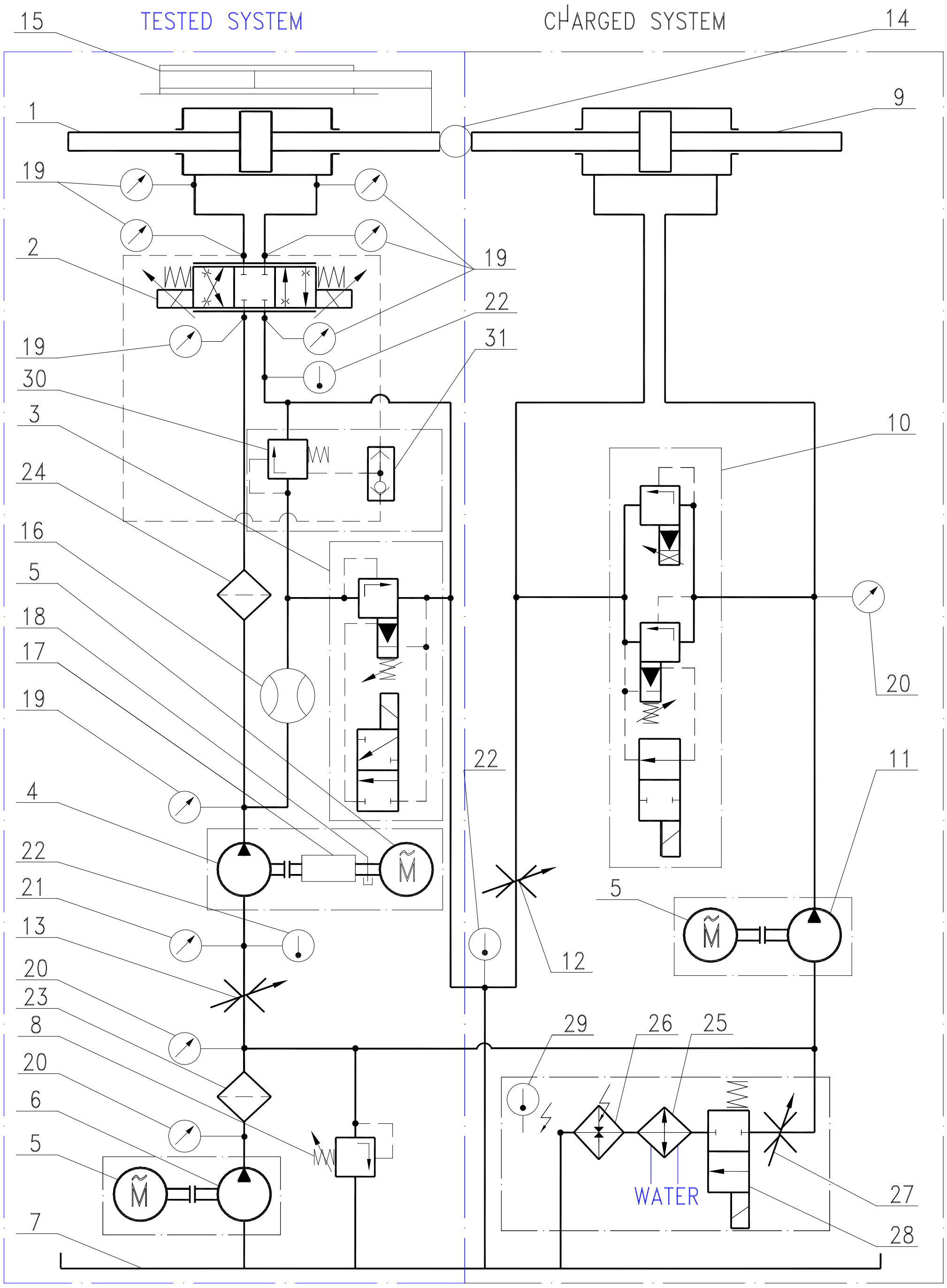

Fig. 1. Diagram of the tested system with a proportional directional control valve supplied with a constant capacity pump cooperating with an overflow valve controlled in a variable pressure system – p=var [3] ; 1 – tested cylinder, 2 – proportional control valve, 3, 8, 10 – overflow valve, 4, 11 – axial multi-piston pump, 5 – electric motor, 6 – pre-feed screw pump, 7 – tank, 9 – charged cylinder, 12, 13, 27 – throttling valve, 14 – force transducer, 15 – displacement transducer, 16 – piston flowmeter, 17 – torque transducer, 18 – rotational speed transducer, 19 – pressure transducer, 20 – manometer, 21 – vacuum gauge, 22 – temperature transducer , 23, 24 – filter, 25 – cooler, 26 – electric heater, 28 – manifold, 29 – thermostat, 30 – controlled overflow valve, 31 – circulation switch

The following components were used in the tested systems [3]:

– axial piston pump type A7.VSO.58DR from Hydromatic, with swivelling rotor, working at constant theoretical capacity $Q_{Pt} = 0.000805$ m^{3}s^{-1} (48.30 dm^{3}min^{-1}),
– proportional directional control valve type 4WRA10E60-21/G24N9K4 from Rexroth, with identical throttling slots $f_{DE1} = f_{DE2}$,

– double piston hydraulic cylinder type CD-63/36x500 from Hydroster, piston diameter D = 63 mm and piston rod diameter d = 36 mm,
– indirect overflow valve type DBW10A3-52/315XU GE 62 4N9K4 from Rexroth,
– overflow control valve type ZDC10PT-23/XM from Rexroth (used only in the variable pressure system – p=var).

The nominal pressure of the tested systems was p_n = 16 MPa. Hydraulic oil Azola 46 with kinematic viscosity υ = 35 mm^2s^{-1} (at temperature υ = 43 ºC) and volumetric mass ρ = 873.3 kgm^{-3} was used.

The tested systems worked with the same parameters, its F_M load, and the v_M speed of the hydraulic cylinder.

Fig. 2. View of the laboratory stand

The considerations make it possible to compare the power quantities ΔP of individual losses resulting from the applied linear speed motor control structure and the power consumed P_{Pc} by the pump from the electric motor, the useful power $P_{Mu} = F_M \cdot v_M$.

Fig. 2 shows the laboratory view from the side of the cylinders: two-pole test (left) and loading (right side).

3. Research methodology

In order to compare losses and energy efficiency of two tested structures: constant pressure p=cte and variable pressure p=var, consisting of the cylinder, conduits, proportional directional control valve, valves: overflow SP (p=cte and p=var) and pressure operated overflow SPS (p=var) and pumps, measurements were made using a laboratory computer using the National Instruments LabView 6.0 program for this purpose. The results of the measurements were processed in Excel. The computer with measuring transducers was connected using a PCI 1713 Advantech measuring card. Thus, 4 signals from pressure transducers, the position signal of the piston rod by means of a linear displacement transducer were registered, on the basis of which its velocity v_M was determined, and the signal of the force F_M charging the piston rod.

Each cycle of measurements was started from oil heating to the temperature of υ = 43°C, at which its kinematic viscosity is ν = 35 mm^2s^{-1}.

At the beginning, the test cylinder performed reciprocating movements without load. Then, after starting the pump supplying the load cylinder, the load was systematically increased. When the measurement results were repeatable, i.e. stable, tests were started.

Thanks to the automated work of the systems, the cylinders moved in both directions without interruption, while during the working movement, the load cylinder loaded the tested cylinder. In the return movement, both cylinders moved independently. This avoids the aeration of the cylinder chambers, which would adversely affect the measurement results.

In order to compare two systems, the same series of speed and piston rod loads were assumed when testing them. During the measurements, the test cylinder was loaded with the force of F_M equal to 0 kN, 5 kN, 10 kN, 15 kN, 20 kN, 25 kN, 30 kN. On the other hand, the cylinder's piston moved at a speed v_M equal to 0.025 m/s, 0.05 m/s, 0.075 m/s, 0.1 m/s, 0.15 m/s, 0.2 m/s, 0.25 m/s, 0.3 m/s, 0.35 m/s, 0.4 m/s [3].

The obtained characteristics of individual elements used in the system were used to calculate the coefficients "k_i" of losses occurring in the system.

In order to compare the tested systems, tests were carried out with the same speed and load parameters of the cylinder, while maintaining the viscosity at value 35 mm^2s^{-1} recommended by the oil producer. Next, the power and efficiency losses were calculated in individual elements and entire systems to compare them.

4. Determination of the coefficients "k_i" connected with tested pump and examples for other hydraulic pumps

Knowledge of the coefficients k_i of energy losses occurring in the hydrostatic system element (in the pump, in the hydraulic motor, but also in the conduits and in the motor) allows to build mathematical models of losses and energy efficiency of the element working in the system and energy efficiency of the system as a whole composed of elements. Mathematical models of losses and energy efficiency in the system must take into account the conditions resulting from the applied structure of the system, from the nominal pressure, from the rotational speed of the motor driving the pump shaft, from the viscosity change of the applied working fluid (hydraulic oil) [1,2].

The coefficients k_i describe the relative value of individual losses in an element. They enable assessment of the proportions of losses and assessment of energy efficiency values (volumetric, pressure, mechanical) resulting from losses occurring at the nominal pressure p_n of the system in which the element is used [1].

As a result, thanks to the knowledge of the coefficients k_i of individual losses, it is possible to determine the losses and energy efficiency of components operating in the drive system as well as the efficiency of the overall system with a defined structure of the motor speed control as a function of the speed coefficient and load coefficient of the hydraulic motor.

The following is a set of definitional formulas allowing to determine the coefficients k_i of energy losses in the pump, working in typical hydrostatic drive with proportional throttling control of the cylinder, with the reference viscosity of υ_n = 35 mm^2s^{-1}.

The flow of the liquid stream in the tested axial multi piston pump (Fig. 3, 4) is carried out by changing the volume of the working chambers, which alternatively suck or extrude the liquid. The determined theoretical capacity Q_{Pt} was obtained by changing the angle of the rotor.

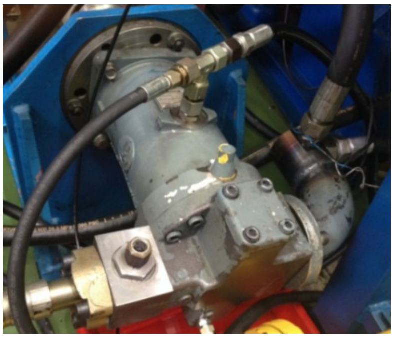

Fig. 3. View of tested pump

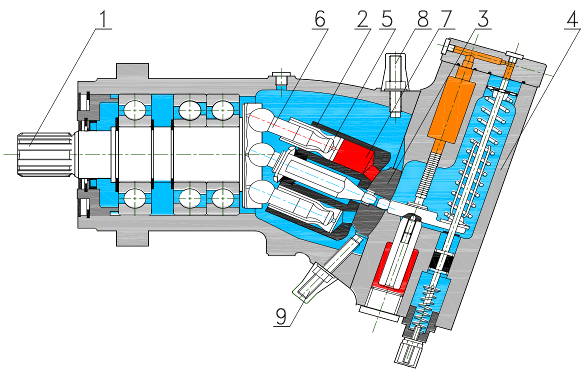

Fig. 4. Axial piston pump with HYDROMATIC: A7.VSO.58DR rotary rotor: 1 – shaft, 2 – cylinder block, 3 – collector, 4 – system for zeroing the efficiency at max. set pressure, 5 – piston, 6 – connector with ball joints, 7 – cylinder chamber, 8, 9 – limiters

– **Coefficients k_i**

k_1 – coefficient of intensity q_{Pv} of volumetric losses in the pump working chambers determined during one shaft revolution:

$$k_1 = \frac{Q_{Pv|\Delta p_{Pi}=p_n}}{n_{P|\Delta p_{Pi}=p_n}} \frac{1}{q_{Pt}} \text{ [2].} \tag{1}$$

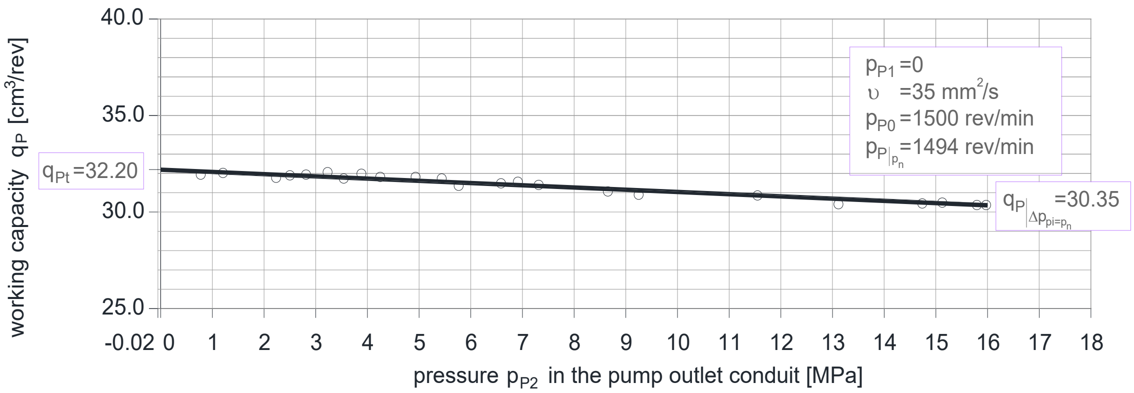

Fig. 5. Characteristics of the working capacity q_P of the pump type A7.VS0.58.DR as a function of p_{P2} pressure in the outlet conduit [3]

Therefore, based on the results obtained during laboratory tests (Fig. 5) the coefficient k_1 is:

$$k_1 = \frac{q_{Pt} - q_{P|\Delta p_{Pi}=p_n}}{q_{Pt}} = \frac{32.20 - 30.35}{32.20} = 0.0574 \text{ [3].} \tag{2}$$

k_2 – coefficient of decrease Δn_P of the constant capacity pump shaft rotational speed, with indicated increase Δp_{Pi} of pressure in working chambers equal to hydrostatic system nominal pressure p_n, in relation to the no-load shaft rotational speed n_{P0}:

$$k_2 = \frac{n_{P0} - n_{P|\Delta p_{Pi}=p_n}}{n_{P0}} = \frac{1500 - 1494}{1500} = 0.004 \text{ [3].} \tag{3}$$

k_3 – coefficient of pressure losses Δp_{Pp} in the pump channels and distributor, determined with flow intensity Q_P equal to the pump theoretical capacity Q_{Pt}:

$$k_3 = \frac{\Delta p_{Pp|Q_P=Q_{Pt}}}{p_n} \text{ [2].} \tag{4}$$

According to the above formula, the coefficient k_3 of the tested pump is:

$$k_3 = \frac{0.38}{160} = 0.0024 \ [3], \tag{5}$$

$k_{4.1}$ – coefficient of torque M_{Pm} of mechanical losses in the "working chambers – shaft" assembly of no-load pump with constant capacity q_{Pt} per one shaft revolution, with indicated increase Δp_{Pi} of pressure in the working chambers equal to zero:

$$k_{4.1} = \frac{2\Pi M_{Pm|\Delta p_{Pi}=0}}{q_{Pt}\ p_n} [2]. \tag{6}$$

Thus, based on the obtained results of laboratory tests (Fig. 6), the value of $k_{4.1}$ coefficient of pump type A7.VS0.58.DR is:

$$k_{4.1} = \frac{3.21[\mathrm{Nm}]}{81.99[\mathrm{Nm}]} = 0.0391 \ [3]. \tag{7}$$

$k_{4.2}$ – coefficient of the increase ΔM_{Pm} of torque of mechanical losses in the "working chambers – shaft" assembly of pump with constant capacity q_{Pt} per one revolution:

$$k_{4.2} = \frac{\Delta M_{Pm|\Delta p_{Pi}=p_n}}{M_{Pn}} = \frac{2\Pi \Delta M_{Pm|\Delta p_{Pi}=p_n}}{q_{Pt}\ p_n} \ [2]. \tag{8}$$

Thus, the value of the pump $k_{4.2}$ coefficient A7.VS0.58.DR is:

$$k_{4.2} = \frac{2\Pi\cdot 1.23\ [\mathrm{Nm}]}{32.20\cdot 10^{-6}\,[\mathrm{m^3/obr}]\cdot 160\cdot 10^5\,[\mathrm{N/m^2}]} = 0.015 \ [3]. \tag{9}$$

In the analysed systems, axial multi piston pump of type A7.VSO.58DR with swivelling rotor from Hydromatic was used, operating at a fixed theoretical output Q_{Pt}=0.000805m^3/s (48.3dm^3/min).

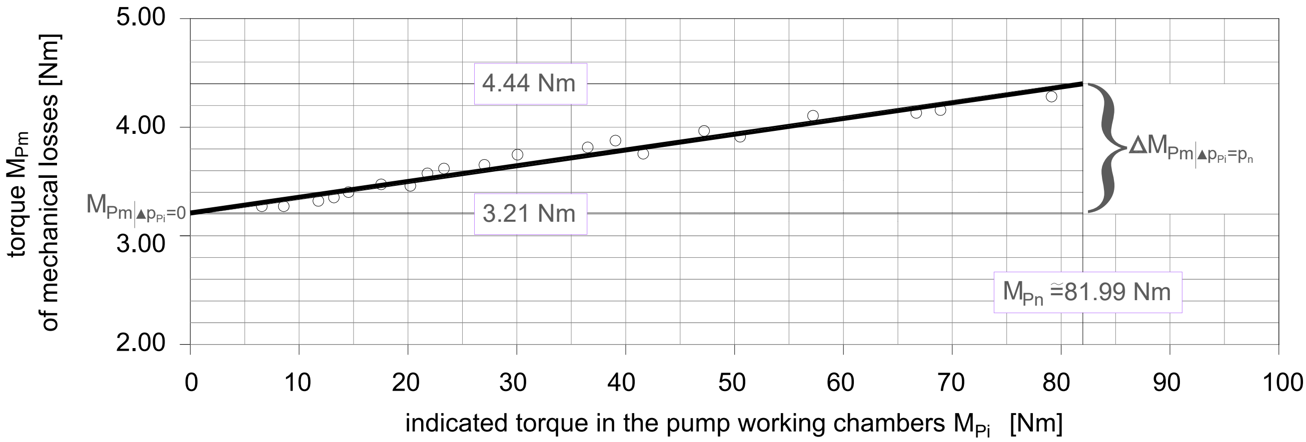

Fig. 6. Torque characteristic M_{Pm} of mechanical losses in the Hydromatik A7.VS0.58.DR pump with its maximum Q_P efficiency as a function of indicated torque M_{Pi} [3]

– **Characteristics of basic pump types – approximate values of k_i coefficients**

It is proposed to apply, in the industrial energy calculations, the approximate values of coefficients k_i of the basic types of pumps.

By way of example, the approximate values of the coefficients k_i of axial piston pumps with swivelling rotor [1] are given in the Tab. 1.

Tab. 1. Examples of coefficients k_i of axial piston pumps [2]

p_n=10 MPa	p_n=25 MPa
k_1 = 0.01	k_1 = 0.02
k_3 = 0.01	k_3 = 0.01
$k_{4.1}$ = 0.01	$k_{4.1}$ = 0.05
$k_{4.2}$ = 0.01	$k_{4.2}$ = 0.02

5. Summary

The results of research and analyses should be able to create and exploit the possibilities simulation programs support the process of designing hydraulic systems. Pursuing such programs will allow fast determination of the efficiency of the hydrostatic transmission with a positive displacement pump, composed of any item at any point of the fieldwork. These programs enable the selection of optimal in terms of energy, the operating parameters.

Coefficients k_i, given in the subject literature by Z. Paszota, describe the relative value of individual losses in the element. They make it possible to assess the proportions of losses and assess the value of energy efficiency (volumetric, pressure, mechanical) resulting from losses occurring at the nominal pressure p_n of the system in which the element is used. As a result, thanks to the knowledge of the coefficients k_i of individual losses, it is possible to determine the losses and energy efficiency of components operating in the hydraulic system as well as the efficiency of the overall system with a defined structure of the motor speed control as a function of the speed coefficient and load coefficient of the hydraulic motor. Knowledge of the coefficients k_i of energy losses occurring in the hydrostatic system elements (in the pump, in the hydraulic motor, but also in the conduits and in the hydraulic motor speed control group) allows to build mathematical models of losses and energy efficiency of the element working in the system and energy efficiency of the system as a whole composed of elements.

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