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TESTS CONSTANT CAPACITY PUMP CONSIDERING THE EFFECT OF VISCOSITY AND COMPRESSIBILITY AERATED OIL

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Abstract

The paper presents mathematical models of the losses occurring in the pump with a theoretical capacity per revolution of the shaft and geometric (variable) performance on the rotation of the shaft. These models were used to laboratory testing and simulation energy losses in the pump to judge the energy efficiency of the pump and the efficiency of the hydrostatic drive. 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 of the system. In an era of increasingly expensive energy, even a small reduction of losses can bring tangible economic benefits during machine operation. It is important to know the energy efficiency of the transmission, under not only nominal conditions, but also changes in the whole range of working conditions: the speed and load of the hydraulic motor, the viscosity of hydraulic oil particularly in the parameters or the longest occurring during operation. It is necessary, therefore, to design stage gear on a detailed analysis of energy enabling optimal selection of components and operating parameters of the system.

Keywords: *displacement pump, control volume, the losses of volume, oil viscosity, compressibility aerated oil*

1. Introduction

Since the nineties the nineteenth century, when it was used mineral oil as the working medium in the first hydrostatic drive machine, this drive is continually evaluates. The result is the extraordinary prevalence in various technical fields – including in ocean engineering and shipbuilding, especially in the field of marine propulsion equipment on board. Examples of marine applications the solutions are drive and control the crane deck, steering gear and the main drive smaller vessels.

The advantage of the hydrostatic drive is the possibility of obtaining a high flux density transmitted power, so that the elements of the system are characterized by compactness, i.e. a small mass per unit generated, transmitted and received power. Another advantage is the possibility of continuously adjustable speed of movement of the driven machine.

Hydraulic drive system comprising a variable displacement pump with a pressure-generating capacity and engine displacement processing this energy into mechanical energy rotary motion is called hydrostatic transmission controlled volume [1]. In this case, the process of controlling the motor speed, as opposed to the transmission throttling control, not associated with the generation of additional power losses and is the solution of the hydrostatic drive with the highest energy efficiency. This drive is used in cases of high-power, long-term work in situations transmission and everywhere, where energy saving is profitable even with expensive investments and greater operational requirements.

Today, in an era of increasingly expensive energy, even a small reduction of losses can bring tangible economic benefits during machine operation. Thus, it is important to know the energy efficiency of the transmission, under not only nominal conditions, but also changes in the whole range of working conditions: the speed and load of the hydraulic motor, the viscosity of hydraulic oil particularly in the parameters or the longest occurring during operation [2]. It is necessary, therefore, to design stage gear on a detailed analysis of energy enabling optimal selection of components and operating parameters of the system [1].

Laboratory tests and simulated effect of viscosity on the energy losses in pump systems with fixed and variable performance has not been carried out. Testing a variable displacement pump with fixed recommended oil viscosity ν_n=35 mm^{2}s^{-1}, was conducted by M. Czyński in the doctoral thesis "Laboratory model of energy efficiency hydrostatic transmission". In contrast, studies the impact of the viscosity of hydraulic oil at a losses in variable capacity pump were conducted by J. Koralewski at the Technical University of Gdansk.

There are energy losses in the elements of hydraulic system, which are, inter alia, a function of the viscosity of the working fluid and the energy losses, which practically not depends on the viscosity.

So far, the author in his tests of elements and entire hydraulic systems accepted as reference, kinematic viscosity ν_n=35 mm^{2}s^{-1} of mineral oil, which is recommended by the manufacturers for the optimal functioning of their production of hydraulic components. The next stage is to determine the effect of viscosity ν of oil on the characteristics of the hydraulic system components, and its impact on the losses of power and efficiency in the studied systems. Then, using equations defining coefficients "k_i" energy losses in the hydrostatic drive system elements describing the losses in the pump, the pipes, the engine and the distributor will be determined and substituted into mathematical models, which will enable comparison of the results of simulation studies with the results of laboratory tests. The proposed using coefficients k_i, mathematical models describing the losses enables to assess the behaviour of the energy components and the system as a function of the ratio ν/ν_n viscosity of hydraulic oil to reference viscosity ν_n.

2. Mathematical models of energy losses in the capacity pump

Described the losses in the systems using the coefficients "k_i" and the creation of mathematical models defining the energy efficiency of hydraulic systems takes many years to Z. Paszota [3-6]. With the knowledge of these coefficients of losses, it is possible to determine the losses and the energy efficiency of elements working in the drive system (total, volume, pressure and mechanical) as well as the overall efficiency of a structured control the motor speed as a function of the speed coefficient and load coefficient hydraulic motor.

The losses of volume Q_{Pv} in the pump piston are primarily the result of:

- leakage of working medium through the gap between the elements of displacement and the
- walls of the chambers, the divider slots (if any), external leakage,
- compressibility of the liquid,
- displacement of the pump changes as a result of changes in pressure and temperature.

Z. Paszota in his works [4-6] presents a model volumetric losses and accepts the description of the assumptions and simplifying's the impact of certain coefficients on these losses, and the impact of these coefficients on the losses of volume is reflected in a specific ratio and exponents of power series describing the dependence of losses the growth of indicated Δp_{Pi} pressure in the working chambers and kinematic viscosity ν of hydraulic oil.

Coefficient k_1 of volumetric losses Q_{Pv}, identified per one shaft revolution of fixed or variable capacity pump by increase of pressure Δp_{Pi}, equalled to the nominal pressure p_n in the hydraulic system – $\Delta p_{Pi}=p_n$ and the oil viscosity ν_n, the losses which are sustained to the theoretical working volume q_{Pt} of pump, describes the formula [5] :

$$k_1 = \frac{q_{Pv|q_{Pt};\Delta p_{Pi}=p_n;\nu_n}}{q_{Pt}} = \frac{Q_{Pv|q_{Pt};\Delta p_{Pi}=p_n;\nu_n}}{n_{P|q_{Pt};\Delta p_{Pi}=p_n;\nu_n}} \frac{1}{q_{Pt}} \text{ [5],} \tag{1}$$

On the size of the mechanical losses, directly affects indicated torque M_{Pi} in the chambers of the pump. The indicated torque M_{Pi} in pump arises from the increase Δp_{Pi} indicated pressure in the working chambers. The increase of pressure Δp_{Pi} is result in turn from the value of stabilized pressure in constant pressure valve *SP* or variable pressure valve *SPS*.

The indicated torque M_{Pi} in the working chamber of the pump can be expressed by the formula:

$$M_{Pi} = \frac{q_{Pt}\,\Delta p_{Pi}}{2\Pi}. \tag{2}$$

The mechanical efficiency η_{Pm} of the pump can be defined as:

$$\eta_{Pm} = \frac{M_{Pi}}{M_P} = \frac{M_{Pi}}{M_{Pi} + M_{Pm}}, \tag{3}$$

where:

M_P – torque on the pump shaft,

M_{Pm} – torque of mechanical losses in the pump.

The torque of mechanical losses M_{Pm} in the pump is the sum of two components:

$$M_{Pm} = M_{Pm|\Delta p_{Pi}=0} + \Delta M_{Pm|\Delta p_{Pi}}, \tag{4}$$

where:

$M_{Pm|\Delta p_{Pi}=0}$ – torque of mechanical losses in the unloaded pump (increase Δp_{Pi}=0), i.e. at the time by torque M_{Pi}=0,

$\Delta M_{Pm|\Delta p_{Pi}}$ – increase torque of mechanical losses in the pump associated with the increase of pressure Δp_{Pi}.

The torque M_{Pm} was determined during the tests as the difference the torque M_P on the pump shaft and the torque of cylinder M_{Pi} in its working chambers:

$$M_{Pm} = M_P - M_{Pi} = M_P - \frac{q_{Pt}\,\Delta p_{Pi}}{2\Pi}. \tag{5}$$

On the other hand, in works [3, 6] Z. Paszota proposes mathematical models describing the torque M_{Pm} of mechanical losses in the pump, referring to theoretical models of the mechanical losses, which have the form:

$$M_{Pm|\Delta p_{Pi},\nu} = \left[k_{4.1}\left(\frac{\nu}{\nu_n}\right)^{a_{vm}} + k_{4.2}\frac{\Delta p_{Pi}}{p_n}\right]\frac{q_{Pt}\,p_n}{2\Pi}, \text{ [3, 6]}, \tag{6}$$

where:

$$k_{4.1} = \frac{M_{Pm|\Delta p_{Pi}=0,\, b_P=1,\nu_n}}{M_{Pt}} = \frac{M_{Pm|\Delta p_{Pi}=0,\, b_P=1,\nu_n}}{\frac{q_{Pt}\,p_n}{2\Pi}}, \text{ [3, 6]}, \tag{7}$$

$$k_{4.2} = \frac{M_{Pm|\Delta p_{Pi}=p_n, b_P=1,\,\nu_n} - M_{Pm|\Delta p_{Pi}=0, b_P=1,\,\nu_n}}{M_{Pt}}, \text{ [3, 6]}. \tag{8}$$

Figure 1 shows dependence of the mechanical losses $M_{Pm|\Delta p_{Pi},b_P,\nu}$ in oil-filled pump, with theoretical working cubic capacity q_{PT} per one rotation shaft of increase indicated pressure Δp_{Pi} in the chambers of the pump.

3. Laboratory results

An axial piston pump with axial swivel cylinder block type A7.VSO.58DR Company HYDROMATIC (Fig. 2) was tested on test stand. The fluid flows in the pump displacement achieved by a change in volume of the working chambers for alternatively suck the oil or extruding. Established theoretical capacity Q_{PT} obtained by changing the angle of the rotor.

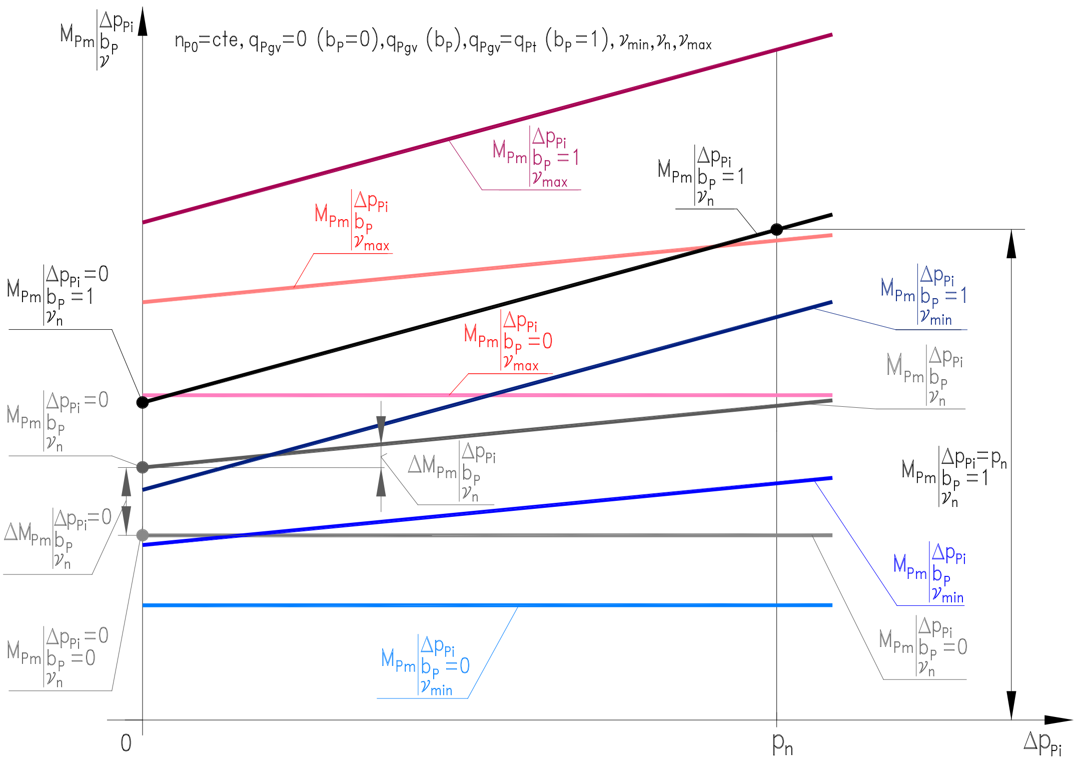

Fig. 1. Torque $M_{Pm|\Delta p_{Pi}, b_P, \nu}$ of mechanical losses in a piston (axial or radial) pump with crankcase filled with liquid and with variable capacity $q_{Pgv}=b_P q_{Pt}$ per one shaft revolution, as a function of the indicated increase Δp_{Pi} of pressure in the pump working chambers – graphical interpretation of theoretical model; capacity q_{Pgv} per one shaft revolution (coefficient b_P of pump capacity): $q_{Pgv}=0$ ($b_P=0$), q_{Pgv} (b_P), $q_{Pgv}=q_{Pt}$ ($b_P=1$); liquid viscosity ν_{min}, ν_n and ν_{max}. Torque $M_{Pm|\Delta p_{Pi}, b_P, \nu}$ of mechanical losses in the pump without the crankcase filled with liquid is practically independent of the liquid viscosity ν and is determined at the liquid reference viscosity ν_n. [3, 6]

In order to determine the characteristics of the pump $q_P = f(p_{P2})$ for $p_{min} \leq p_{P2} \leq 160$ *bar* assembled position measurement and a positive displacement pump with a fixed performance was analysed, taking into account changes in temperature (viscosity) of hydraulic oil. The study was performed at 5 temperatures υ hydraulic oil (kinematic viscosity ν of oil): υ=20ºC (ν=120 mm^{2}s^{-1}), υ=30ºC (ν=65 mm^{2}s^{-1}), υ=43ºC (ν=35 mm^{2}s^{-1}), υ=50ºC (ν=26 mm^{2}s^{-1}), υ=65ºC (ν=16 mm^{2}s^{-1}).

Various characteristics of the obtained test pump depending on the temperature (viscosity) of oil, as shown in Fig. 3. It can be seen that the higher the pressure, the greater the leakage in the pump and hence lower the actual expense. The graph also shows that along with decreasing oil temperature (increase in viscosity) pump capacity increases, which is due to lesser intensity of leaks in the pump body, occurring between mating components. This is shown by curves that change their inclination assuming an increasing angle to the axis of the abscissa – p_{P2} pressure in the pressure pipe. The legend oil temperature refers to the following curves starting from the top of the chart.

Investigation of the effect of viscosity of hydraulic oil at losses in volume piston pump with variable done J. Koralewski. Fig. 4 shows the distribution of volumetric losses $Q_{Pv}=f(\Delta p_{Pi})$ on rotation of the shaft at a losses $q_{Pvc}=f(\Delta p_{Pi})$ arising from the compressibility of the oil and the losses $q_{Pvl}=f(\Delta p_{Pi})$ resulting from oil leaks at different ε ratio of trapped oil in the test pump, the geometric working volume q_{Pgv} and theoretical working volume Q_{Pt} rotation of the pump shaft. Fig. 4 presents the subdivision of volumetric losses $q_{Pv}=f(\Delta p_{Pi})$ in the tested pump into loss $q_{Pvc}=f(\Delta p_{Pi})$ due to liquid compressibility and loss $q_{Pvl}=f(\Delta p_{Pi})$ due to oil leakage at different values of liquid aeration coefficient ε, with the theoretical working capacity q_{Pt} per one pump shaft revolution. The lines of $q_{Pvl}=f(\Delta p_{Pi})$ loss due to oil leakage do not change at different oil aeration coefficient ε value, but lines $q_{Pvc}=f(\Delta p_{Pi})$ due to oil compressibility differ clearly, as well as lines $q_{Pv}=q_{Pvl}+q_{Pvc}=f(\Delta p_{Pi})$ of volumetric losses $q_{Pv}=f(\Delta p_{Pi})$ as a sum of $q_{Pvl}=f(\Delta p_{Pi})$ of loss due to leakage and $q_{Pvc}=f(\Delta p_{Pi})$ of loss due to liquid compressibility [2, 6, 7].

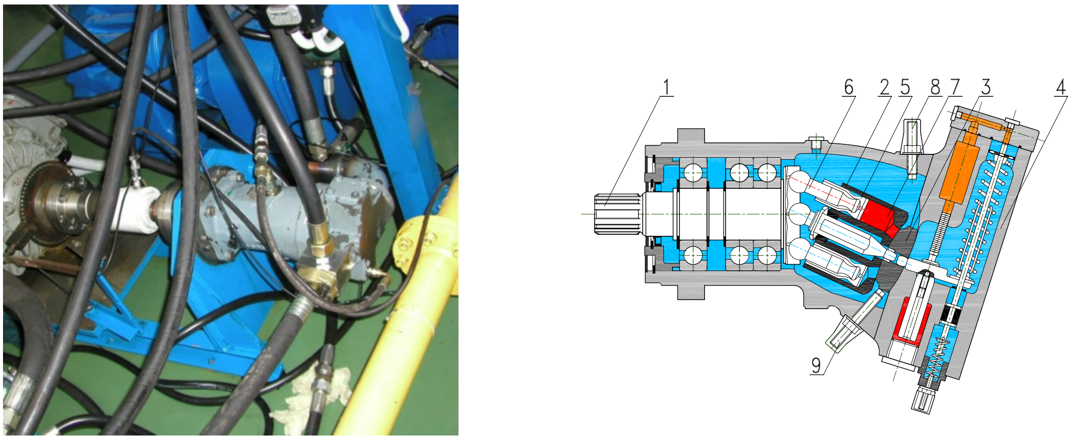

Fig. 2. Tested constant capacity pump type A7.VSO.58DR

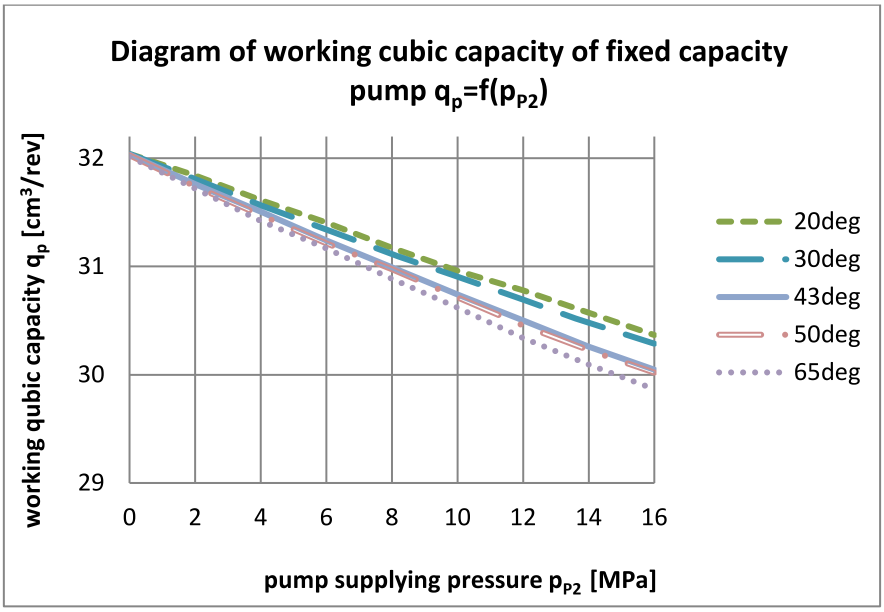

Fig. 3. Diagram of the working cubic capacity q_P of the pump A7.VS 0.58.GR as a function of pump supplying pressure p_{P2} at different hydraulic oil temperatures

The values of coefficients k_1, based on laboratory tests, of relative volumetric losses per one shaft revolution of fixed capacity pump at different temperatures (viscosity) of hydraulic oil were calculated: $k_{1|20°C}=0.052$, $k_{1|30°C}=0.055$, $k_{1|43°C}=0.062$, $k_{1|50°C}=0.063$, $k_{1|65°C}=0.067$.

Figure 5 shows diagram of the torque mechanical losses in the tested pump. The value of torque $M_{Pm|\Delta p_{Pi}=0}$ in the unloaded pump is approximated value at the point $M_{Pi}=0$ and amounted to: $M_{Pm|\Delta p_{Pi}=0}=3.21\,\mathrm{Nm}$.

The coefficient $k_{4.1}$ is the ratio of the torque of mechanical losses $M_{Pm|\Delta p_{Pi}=0}$ in the pump with constant capacity (i.e., the $M_{Pi}=0$) until the nominal torque M_{Pn} for the pump A7.VS0.58.DR is:

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

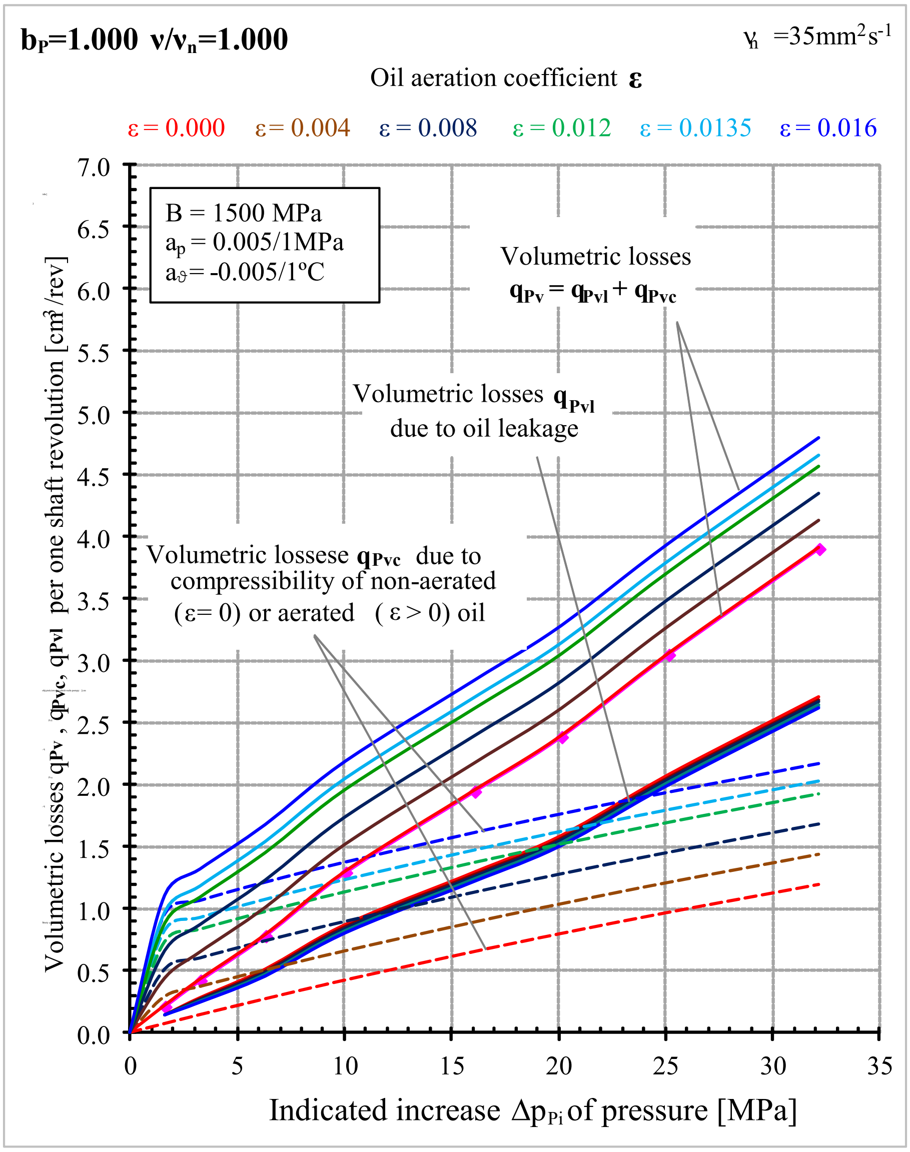

Fig. 4. Subdivision of volumetric losses $q_{Pv}=f(\Delta p_{Pi})$ in the pump into losses $q_{Pvc}=f(\Delta p_{Pi})$ due to oil compressibility and losses $q_{Pvl}=f(\Delta p_{Pi})$ due to oil leakage at different values of oil aeration coefficient ε and value ν/ν_n =1 viscosity of oil viscosity coefficient in the tested pump with theoretical working capacity q_{Pt} (b_P=1) (pump HYDROMATIK A7V.DR.1.RPF00 type) [2]

The coefficient $k_{4.2}$, which is the ratio of relative increase $\Delta M_{Pm|\Delta p_{Pi}=p_n}$ of mechanical pump losses, at increase of pressure in pump working chambers to the nominal torque M_{Pn} and for the test pump is:

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

4. Summary and conclusions

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 [1].

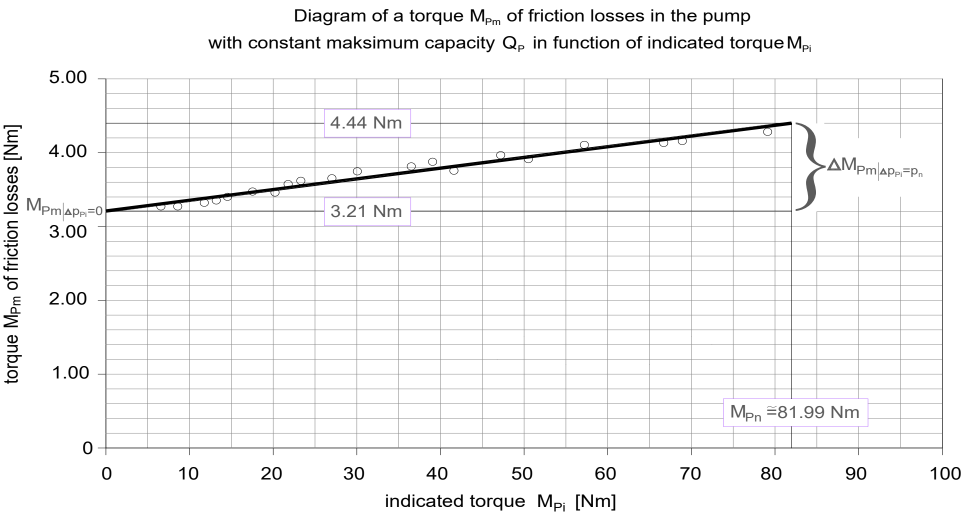

Fig. 5. Torque diagram M_{Pm} of friction losses in the pump type A7.VS 0.58.DR with maximum constant capacity Q_P as a function of the indicated torque M_{Pi} [8]

The pump cannot be sealed completely. There are always leakages between working parts, e.g.: between the piston and the cylinder, distributor and body. The part of the flow pumped by the pump returns back to the high-pressure side to the low pressure, that is, in a direction opposite to the flow direction in leakage.

The flow rate of leakage depends on the geometrical dimensions of the slots, the pressure difference and liquid viscosity. Assuming, stability of the geometric dimensions and viscosity, it can be assumed that the flow rate of leakage depends primarily on the pressure in the pump discharge channel.

The accuracy of evaluation of q_{Pt} and q_{Pgv} significantly worse when the working liquid is aerated. This is an effect of the fact, that the aerated liquid in the working chambers filled during their connection with the low pressure inlet channel decreases its volume because of great compressibility of non-dissolved air in the liquid, after connection of the working chambers with the discharge channel where pressure may be only a little higher than in the inlet channel.

Without knowledge of the coefficient ε of aeration of the oil flowing into the pump working chambers, it is impossible to determine the quantities q_{Pt} and q_{Pgv} precisely.

At the same time, precise knowledge of q_{Pt} and q_{Pgv} is important in evaluation of the volumetric and mechanical losses in the pump.

The intensity $q_{Pv}=Q_{Pv}/n_P$ of volumetric losses Q_{Pv} in the pump working chambers per one pump shaft revolution is evaluated as a difference between q_{Pt} (or q_{Pgv}) and q_P determined during the investigation at changing values of the indicated increase Δp_{Pi} of pressure in the chambers [6].

Increase $\Delta M_{Pm|\Delta p_{Pi}}$ of the torque of mechanical losses in the pump „working chambers – shaft" assembly, compared with torque $\Delta M_{Pm|\Delta p_{Pi}=0}$ of mechanical losses in the no-load pump, is an effect of increased friction forces in the assembly resulting from the influence of the torque M_{Pi} indicated in the working chambers upon the assembly and is proportional to M_{Pi} [6].

Increase $\Delta M_{Pm|\Delta p_{Pi}}$ of the torque of mechanical losses in the pump „working chambers – shaft" assembly is determined during the investigations as a difference $\Delta M_{Pm|\Delta p_{Pi}} = M_{Pm} - M_{Pm|\Delta p_{Pi}=0}$ between torques M_{Pm} of losses in the assembly of no-load pump [6].

Ability to determine the trapped air working fluid and the resulting liquid compressibility allows specifying losses volume Q_{Pv} and breakdown of these losses q_{Pvl} , losses resulting from leaks in the chambers q_{Pvc} and losses arising from the compressibility of the fluid in the chambers, which do not involve the construction of a positive displacement pump. Knowledge of the compressibility of the liquid without air allows determining the volumetric losses resulting from the leakage of the liquid in the chambers of the pump. It should be clearly separate the losses of volume due to fluid leakage and losses of volume resulting from the compressibility of liquids, and to assess the pump only take losses resulting from leaks.

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