POLISH MARITIME RESEARCH 3(75) 2012 Vol 19; pp. 36-44
10.2478/v10012-012-0029-z

Effect of the working liquid compressibility on the picture of volumetric and mechanical losses in a high pressure displacement pump used in a hydrostatic drive

Part II

Mechanical losses in a pump

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ABSTRACT

Working liquid compressibility may considerably change the values and proportions of coefficients of the volumetric and mechanical energy losses in the displacement pump used in a hydrostatic drive system. This effect can be particularly seen in the operation under high pressure and also in the system, where aeration of the working liquid can occur. In the Part II the mathematical model is presented of the torque of mechanical losses in the pump and its laboratory verification. Conclusions are drawn regarding the effect of working liquid compressibility on the mechanical and volumetric losses in the pump.

Keywords: hydrostatic drive system; pressure displacement pump; energy losses; energy efficiency; energy saving; liquid compressibility; Sankey diagram

8. MODEL OF THE MECHANICAL LOSSES IN THE PUMP „WORKING CHAMBERS - SHAFT" ASSEMBLY

The pump shaft torque M_P (required by the pump of its driving motor) must be greater than the torque M_{Pi} indicated in the pump working chambers because of the necessity of balancing also the torque M_{Pm} of mechanical losses in the „working chambers - shaft" assembly. The assembly forms the working chambers and changes their capacity and also connects the working chambers with the shaft. Therefore, the torque M_P required on the pump shaft is a sum of the torque M_{Pi} indicated in the working chambers and the torque M_{Pm} of mechanical losses in the pump „working chambers - shaft" assembly [19]:

$$M_P = M_{Pi} + M_{Pm} \qquad (8.1)$$

Torque M_{Pm} of mechanical losses in the pump with variable capacity q_{Pgv} per one shaft revolution is, at the maximum value of q_{Pgv} i.e. $q_{Pgv} = q_{Pt}$ (with $b_P = q_{Pgv}/q_{Pt} = 1$), equal to the torque of mechanical losses in that pump working as a pump with constant capacity q_{Pt} per one shaft revolution. The theoretical and mathematical models describing the torque M_{Pm} of mechanical losses in the pump with variable capacity q_{Pgv} per one shaft revolution may be based on models of M_{Pm} describing the torque of mechanical losses in the pump with constant capacity q_{Pt} per one shaft revolution (with $b_P = 1$). Considering the models describing the torque of pump mechanical losses, we assume, that, in the hydrostatic transmission, pump is driven with practically constant rotation speed n_P, and the decrease of shaft speed (decrease of the pump driving motor speed as a result of the increase of torque M_P loading the motor shaft) to a value $n_P < n_{P0}$ (n_{P0} – rotational speed of unloaded pump driving motor) is negligible from the point of view of the impact of speed n_P on the value of torque M_{Pm} of mechanical losses.

Torque M_{Pm} of mechanical losses in the pump is mainly an effect of friction forces between elements of the „working chambers - shaft" assembly, depending, among others, on the torque M_{Pi} indicated in the working chambers – $M_{Pi} = q_{Pgv}\,\Delta p_{Pi}/2\Pi = b_P\,q_{Pt}\,\Delta p_{Pi}/2\Pi$.

Friction forces between elements of the „working chambers - shaft" assembly are, to some extent, also an effect of the load on those elements of the inertia forces from rotational and reciprocating motion and depend on the pump capacity q_{Pgv} per one shaft revolution (b_P coefficient).

Particularly, in piston (axial or radial) pumps with casing (crankcase) filled with liquid, friction forces also occur between elements of the „working chambers - shaft" assembly and the liquid and depend on the liquid viscosity ν.

The value of torque $M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of mechanical losses in the pump „working chambers - shaft" assembly, loaded with indicated increase Δp_{Pi} of pressure in the working chambers, in the pump operating at the capacity $q_{Pgv} = b_p\,q_{Pt}$ per one shaft revolution and discharging the working liquid with (constant) reference viscosity ν_n, can be described as a sum of torque $M_{Pm|\Delta p_{Pi}=0, b_P, \nu_n}$ of mechanical losses in the unloaded pump (torque of the losses when the indicated increase Δp_{Pi} of pressure in the pump working chambers is equal to zero – $\Delta p_{Pi} = 0$) and increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of torque of mechanical losses, the increase being an effect of loading the assembly structure elements with torque M_{Pi} indicated in the pump working chambers (torque M_{Pi} appearing when the indicated increase Δp_{Pi} of pressure in the pump working chambers is greater than zero – $\Delta p_{Pi} > 0$):

$$M_{Pm|\Delta p_{Pi}, b_P, \nu_n} = M_{Pm|\Delta p_{Pi}=0, b_P, \nu_n} + \Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n} \qquad (8.2)$$

Torque M_{Pi} indicated in the pump working chambers is proportional to the increase Δp_{Pi} of pressure in the chambers and to the active volume of the chambers created during one pump shaft revolution, which is equal to the theoretical capacity q_{Pt} (V_P) per one shaft revolution in a pump with constant capacity per one shaft revolution or to the geometrical capacity $q_{Pgv} = b_P\, q_{Pt}$ per one shaft revolution in a pump with variable capacity per one shaft revolution.

Therefore, the „working chambers - shaft" assembly structure elements are loaded:

– in a pump with theoretical (constant) capacity q_{Pt} per one shaft revolution – with indicated torque:

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

– in a pump with geometrical (variable) capacity q_{Pgv} per one shaft revolution – with indicated torque:

$$M_{Pi} = \frac{q_{Pgv}\Delta p_{Pi}}{2\Pi} = \frac{b_P\, q_{Pt}\Delta p_{Pi}}{2\Pi}$$

which, in effect, causes a differentiated intensity of the increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of the torque of mechanical losses, determined, with different values of coefficient $b_P = q_{Pgv}/q_{Pt}$, as a function of the increase Δp_{Pi} of pressure in the pump working chambers.

In the theoretical and mathematical models describing the torque $M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of mechanical losses a hypothesis is assumed, that the **increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of the torque of mechanical losses in the pump is proportional to the torque M_{Pi} indicated in its working chambers**.

The impact of inertia forces of the „working chambers - shaft" assembly elements, performing the rotational and reciprocating motion in the pump, on the torque M_{Pm} of mechanical losses can be presented, under the assumption that rotational speed n_P of the pump driving motor changes only in a small range, as a function of capacity q_{Pgv} (b_P coefficient) per one shaft revolution of a variable capacity pump. Inertia forces do not depend on the value of increase Δp_{Pi} of pressure in the working chambers, therefore their impact on the torque M_{Pm} of mechanical losses in the pump may be included in the evaluation of the torque $M_{Pm|\Delta p_{Pi}=0, b_P, \nu_n}$ of mechanical losses determined at the increase $\Delta p_{Pi} = 0$.

The impact of the friction forces between the „working chambers - shaft" assembly elements and the liquid on the torque M_{Pm} of mechanical losses in the pump can be presented, under the assumption that speed n_P changes in a small range, as a relation of M_{Pm} to the viscosity ν and to the capacity q_{Pgv} (b_P coefficient) per one shaft revolution (Fig. 8.1).

It is assumed, that the **impact of liquid viscosity ν on the friction forces between the „working chambers - shaft" elements and the liquid in the piston pump casing (crankcase), and in effect on the torque M_{Pm} of mechanical losses in the pump, can be evaluated** at one level of the increase Δp_{Pi} of pressure indicated in the working chambers, e.g. **at the increase $\Delta p_{Pi} = 0$**. This assumption is connected with a simplification assuming that there is no significant impact of the increase Δp_{Pi} of pressure on the liquid viscosity ν and with assuming in the model describing the torque M_{Pm} of mechanical losses the liquid viscosity ν determined in the pump inlet conduit (at pressure p_{P1} equal to zero (at liquid absolute pressure equal to atmospheric pressure)).

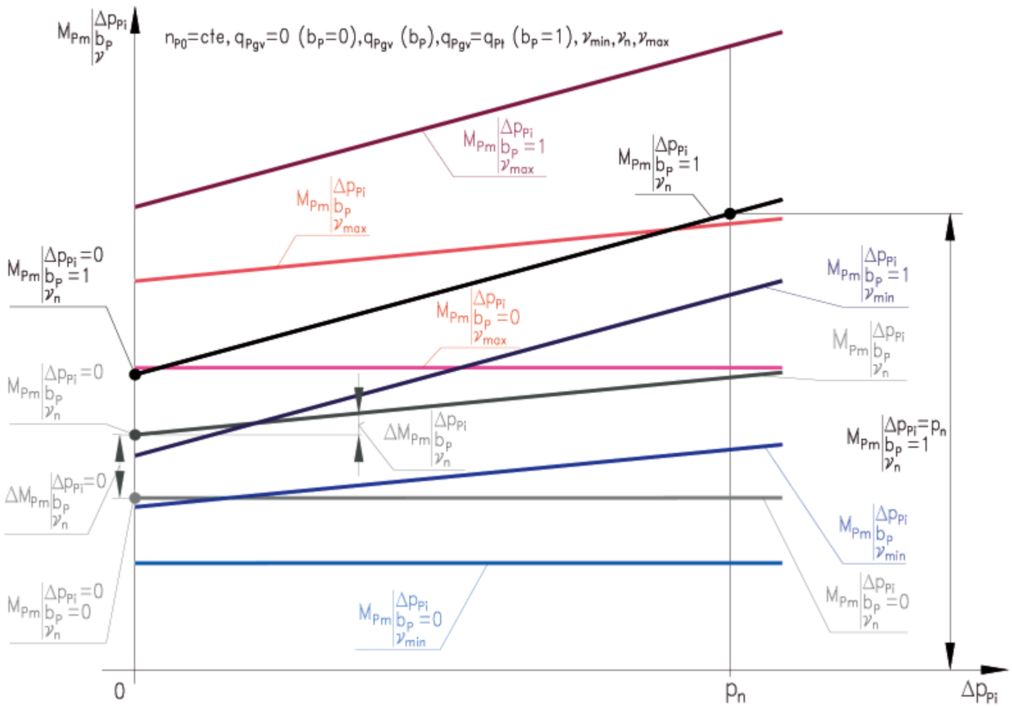

Fig. 8.1. *Torque $M_{Pm|\Delta p_{Pi}, b_P, \nu}$ of mechanical losses in a pump (especially in a piston (axial or radial) pump with crankcase filled with liquid) and with variable capacity $q_{Pgv} = b_P\, q_{Pt}$ per one shaft revolution, as a function of the indicated increase Δp_{Pi} of pressure in the pump working chambers – graphical interpretation of theoretical model (9); 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}*

The impact of inertia forces of structure elements performing the rotational or reciprocating motion in the pump and also the impact of liquid viscosity ν on the torque M_{Pm} of mechanical losses in the pump is then described in the model of the torque $M_{Pm|\Delta p_{Pi}=0, b_P, \nu}$ of those losses in an unloaded pump (at $\Delta p_{Pi} = 0$) supplied with working liquid of changing viscosity ν.

The proposed **theoretical models, describing the torque $M_{Pm|\Delta p_{Pi}=0, b_P, \nu}$ of mechanical losses in an unloaded pump (at the indicated increase $\Delta p_{Pi} = 0$ of pressure in the working chambers) and at changing working liquid viscosity ν,** have the form:

- in the pump with theoretical (constant) capacity q_{Pt} ($b_P = 1$) per one shaft revolution:

$$M_{Pm|\Delta p_{Pi}=0, b_P=1, \nu} = M_{Pm|\Delta p_{Pi}=0, b_P=1, \nu_n}\left(\frac{\nu}{\nu_n}\right)^{a_{vm}} \quad (8.3)$$

- in the pump with geometrical (variable) capacity q_{Pgv} ($q_{Pgv} = b_P\, q_{Pt}$) per one shaft revolution:

$$M_{Pm|\Delta p_{Pi}=0, b_P, \nu} =$$

$$= \left(M_{Pm|\Delta p_{Pi}=0, b_P=0, \nu_n} + \Delta M_{Pm|\Delta p_{Pi}=0, b_P, \nu_n}\right)\left(\frac{\nu}{\nu_n}\right)^{a_{vm}} \quad (8.4)$$

where:

$$\Delta M_{Pm|\Delta p_{Pi}=0, b_P, \nu_n} = M_{Pm|\Delta p_{Pi}=0, b_P, \nu_n} +$$

$$- M_{Pm|\Delta p_{Pi}=0, b_P=0, \nu_n} = \quad (8.5)$$

$$= \left(M_{Pm|\Delta p_{Pi}=0, b_P=1, \nu_n} - M_{Pm|\Delta p_{Pi}=0, b_P=0, \nu_n}\right) b_P$$

Exponent a_{vm} in expressions (8.3) and (8.4) **describes the impact of the ratio ν/ν_n of working liquid ν to reference viscosity $\nu_n = 35 mm^2s^{-1}$ on the value of torque of mechanical losses especially in a piston displacement machine with liquid filling the casing (crankcase)** (in the pump and in the hydraulic motor).

The increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu}$ of the torque of mechanical losses in the pump, due to the load of the assembly elements with the indicated torque M_{Pi} resulting from the indicated increase Δp_{Pi} of pressure in the pump working chambers, is independent of the inertia forces of elements performing the rotational or reciprocating motion in the pump. It is also practically independent of the working liquid viscosity ν; therefore, it may be determined at one viscosity value, e.g. at the liquid reference viscosity ν_n (Fig. 8.1).

In the mathematical models describing the torque M_{Pm} of mechanical losses in the pump, coefficients k_i of losses are used relating (comparing) the components describing the torque M_{Pm} of losses in theoretical models to the pump theoretical torque M_{Pt}. The pump theoretical torque M_{Pt} is also a reference value used in the description of the torque M_{Pi} indicated in the pump working chambers:

– theoretical torque:

$$M_{Pt} = \frac{q_{Pt} p_n}{2\Pi}$$

of the pump, with theoretical (constant) capacity q_{Pt} per one shaft revolution ($b_P = 1$), is determined with the increase Δp_P of pressure in the pump equal to the system nominal pressure $p_n - \Delta p_P = p_n$, and with the assumption that there are no pressure and mechanical losses in the pump,

– indicated torque:

$$M_{Pi} = \frac{q_{Pt}\Delta p_{Pi}}{2\Pi} = \frac{q_{Pt} p_n}{2\Pi}\frac{\Delta p_{Pi}}{p_n} = M_{Pt}\frac{\Delta p_{Pi}}{p_n}$$

in working chambers of the pump with theoretical (constant) capacity q_{Pt} per one shaft revolution ($b_P = 1$) is determined with the indicated increase Δp_{Pi} of pressure in the working chambers,

– indicated torque:

$$M_{Pi} = \frac{q_{Pgv}\Delta p_{Pi}}{2\Pi} = \frac{b_P q_{Pt}\Delta p_{Pi}}{2\Pi} =$$

$$= \frac{q_{Pt} p_n}{2\Pi} b_P \frac{\Delta p_{Pi}}{p_n} = M_{Pt} b_P \frac{\Delta p_{Pi}}{p_n}$$

in working chambers of the pump with geometrical (variable) capacity $q_{Pgv} = b_P\, q_{Pt}$ per one shaft revolution is determined with the indicated increase Δp_{Pi} of pressure in the working chambers.

The theoretical and mathematical models describe the torque M_{Pm} of mechanical losses in the pump with theoretical (constant) capacity q_{Pt} per one shaft revolution or with geometrical (variable) capacity $q_{Pgv} = b_P\, q_{Pt}$ per one shaft revolution:

– $q_{Pt} = q_{P|\Delta p_{Pi}=0, p_{P1i}=0, b_P=1, \nu_n}$ is theoretical capacity per one shaft revolution of the pump with constant capacity per one revolution ($b_P = 1$) determined at $\Delta p_{Pi} = 0$, $p_{P1i} = 0$ and ν_n, which is equal to the working volume of the working chambers created during one shaft revolution,

– $q_{Pgv} = b_P\, q_{Pt}$ is geometrical capacity per one shaft revolution of the pump with variable capacity per one revolution at $\Delta p_{Pi} = 0$, $p_{P1i} = 0$ and ν_n, which is equal to the working volume of the working chambers created during one shaft revolution. Capacity q_{Pgv} per one shaft revolution changes in the $0 \le q_{Pgv} \le q_{Pt}$ range and coefficient $b_P = q_{Pgv}/q_{Pt}$ of the pump capacity changes in the $0 \le b_P \le 1$ range.

The proposed **mathematical models describing the torque M_{Pm} of mechanical losses in the pump, related to theoretical models of the torque of mechanical losses**, take the form:

- in a pump with theoretical (constant) capacity q_{Pt} per one shaft revolution ($b_P = 1$):

$$M_{Pm|\Delta p_{Pi}, \nu} = k_{4.1} M_{Pt}\left(\frac{\nu}{\nu_n}\right)^{a_{vm}} + k_{4.2} M_{Pt}\frac{\Delta p_{Pi}}{p_n} =$$

$$= \left[k_{4.1}\left(\frac{\nu}{\nu_n}\right)^{a_{vm}} + k_{4.2}\frac{\Delta p_{Pi}}{p_n}\right] M_{Pt} = \quad (8.6)$$

$$= \left[k_{4.1}\left(\frac{\nu}{\nu_n}\right)^{a_{vm}} + k_{4.2}\frac{\Delta p_{Pi}}{p_n}\right]\frac{q_{Pt} p_n}{2\Pi}$$

where:

$$k_{4.1} = \frac{M_{Pm|\Delta p_{Pi}=0, b_P=1, \nu_n}}{M_{Pt}} = \frac{M_{Pm|\Delta p_{Pi}=0, b_P=1, \nu_n}}{\frac{q_{Pt} p_n}{2\Pi}} \quad (8.7)$$

$$k_{4.2} = \frac{\Delta M_{Pm|\Delta p_{Pi}, b_P = 1, \nu_n}}{M_{Pi}} = \frac{\Delta M_{Pm|\Delta p_{Pi}, b_P = 1, \nu_n}}{\frac{q_{Pt} \Delta p_{Pi}}{2\Pi}} =$$

$$= \frac{M_{Pm|\Delta p_{Pi}, b_P = 1, \nu_n} - M_{Pm|\Delta p_{Pi} = 0, b_P = 1, \nu_n}}{\frac{q_{Pt} \Delta p_{Pi}}{2\Pi}} = \tag{8.8}$$

$$= \frac{M_{Pm|\Delta p_{Pi} = p_n, b_P = 1, \nu_n} - M_{Pm|\Delta p_{Pi} = 0, b_P = 1, \nu_n}}{\frac{q_{Pt} p_n}{2\Pi}} =$$

$$= \frac{M_{Pm|\Delta p_{Pi} = p_n, b_P = 1, \nu_n} - M_{Pm|\Delta p_{Pi} = 0, b_P = 1, \nu_n}}{M_{Pt}}$$

- in a pump with geometrical (variable) capacity q_{Pgv} ($q_{Pgv} = b_P\, q_{Pt}$) per one shaft revolution):

$$M_{Pm|\Delta p_{Pi}, b_P, \nu} = (k_{4.1.1} + k_{4.1.2}\, b_P) M_{Pt} \left(\frac{\nu}{\nu_n}\right)^{a_{\nu m}} +$$

$$+ k_{4.2} M_{Pt} b_P \frac{\Delta p_{Pi}}{p_n} = \tag{8.9}$$

$$= \left[(k_{4.1.1} + k_{4.1.2}\, b_P) \left(\frac{\nu}{\nu_n}\right)^{a_{\nu m}} + k_{4.2}\, b_P \frac{\Delta p_{Pi}}{p_n} \right] M_{Pt} =$$

where:

$$= \left[(k_{4.1.1} + k_{4.1.2}\, b_P) \left(\frac{\nu}{\nu_n}\right)^{a_{\nu m}} + k_{4.2}\, b_P \frac{\Delta p_{Pi}}{p_n} \right] \frac{q_{Pt} p_n}{2\Pi}$$

$$k_{4.1.1} = \frac{M_{Pm|\Delta p_{Pi} = 0, b_P = 0, \nu_n}}{M_{Pt}} = \frac{M_{Pm|\Delta p_{Pi} = 0, b_P = 0, \nu_n}}{\frac{q_{Pt} p_n}{2\Pi}} \tag{8.10}$$

$$k_{4.1.2} = \frac{M_{Pm|\Delta p_{Pi} = 0, b_P = 1, \nu_n} - M_{Pm|\Delta p_{Pi} = 0, b_P = 0, \nu_n}}{M_{Pt}} =$$

$$= \frac{M_{Pm|\Delta p_{Pi} = 0, b_P = 1, \nu_n} - M_{Pm|\Delta p_{Pi} = 0, b_P = 0, \nu_n}}{\frac{q_{Pt} p_n}{2\Pi}} \tag{8.11}$$

$$k_{4.2} = \frac{\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}}{M_{Pi}} = \frac{\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}}{\frac{b_P\, q_{Pt} \Delta p_{Pi}}{2\Pi}} =$$

$$= \frac{\Delta M_{Pm|\Delta p_{Pi}, b_P = 1, \nu_n}}{\frac{q_{Pt} \Delta p_{Pi}}{2\Pi}} = \tag{8.12}$$

$$= \frac{M_{Pm|\Delta p_{Pi} = p_n, b_P = 1, \nu_n} - M_{Pm|\Delta p_{Pi} = 0, b_P = 1, \nu_n}}{\frac{q_{Pt} p_n}{2\Pi}} =$$

$$= \frac{M_{Pm|\Delta p_{Pi} = p_n, b_P = 1, \nu_n} - M_{Pm|\Delta p_{Pi} = 0, b_P = 1, \nu_n}}{M_{Pt}}$$

Commentary:

- The sum ($k_{4.1.1} + k_{4.1.2}$) of coefficients used in mathematical model (8.9) describing the torque M_{Pm} of mechanical losses in the pump with geometrical (variable) capacity q_{Pgv} ($q_{Pgv} = b_P\, q_{Pt}$) per one shaft revolution is equal to coefficient $k_{4.1}$ used in the mathematical model (8.6) describing the torque M_{Pm} of mechanical losses in that pump working as a pump with theoretical (constant) capacity per one shaft revolution: $k_{4.1.1} + k_{4.1.2} = k_{4.1}$.
- Coefficient $k_{4.2}$ used in mathematical model (8.9) describing the torque M_{Pm} of mechanical losses in the pump with geometrical (variable) capacity q_{Pgv} ($q_{Pgv} = b_P\, q_{Pt}$) per one shaft revolution is equal to coefficient $k_{4.2}$ used in the mathematical model (8.6) describing the torque M_{Pm} of mechanical losses in that pump working as a pump with theoretical (constant) capacity q_{Pt} per one shaft revolution.

9. RESULTS OF INVESTIGATION OF THE TORQUE M_{PM} OF MECHANICAL LOSSES IN THE MEDIUM PRESSURE AXIAL PISTON VARIABLE CAPACITY PT0Z2-25 TYPE PUMP USED IN A HYDROSTATIC TRANSMISSION SYSTEM

Michał Czyński [21] performed the investigation of the PT0Z2-25 medium pressure pump as laboratory verification of the mathematical model of the hydrostatic transmission energy efficiency. The pump was loaded by a hydraulic motor and not by an overflow valve (as it was in the case of the investigation of the high pressure axial piston HYDROMATIK A7V.58. DR.1.R.P.F.00 type pump). The threat of the hydraulic oil aeration was smaller. The system operated under the nominal pressure $p_n = 16$MPa, half of the value of pressure used in the investigation of the A7V.58.DR.1.R.P.F.00 pump.

The investigation lasted relatively short time, the hydraulic oil tank was large in relation to the pump power. Therefore, the effect of oil aeration and of its compressibility was minimized.

The torque M_{Pm} of mechanical losses in the „working chambers - shaft" assembly of the pump is determined from the measurement of the shaft torque M_P and the torque M_{Pi} indicated in the working chambers. Direct measurement of the torque M_{Pi} is not possible. Therefore, it is necessary to determine the torque M_{Pm} of mechanical losses by an indirect method from the formula:

$$M_{Pm} = M_P - M_{Pi} = M_P - \frac{q_{Pgv}\, \Delta p_{Pi}}{2\Pi} = \tag{9.1}$$

$$= M_P - \frac{b_P\, q_{Pt}\, \Delta p_{Pi}}{2\Pi}$$

Fig. 9.1 presents results of the investigation of the torque M_P on the pump shaft as a function of the indicated increase Δp_{Pi} of pressure in the working chambers for different values of the coefficient b_P of the pump capacity $q_{Pgv} = b_P\, q_{Pt}$ per one pump revolution. The investigation results show a very high (close to 1) value of the determination coefficient R^2.

Fig. 9.2 and 9.3 present the calculated values of the torque M_{Pi} indicated in the pump working chambers and the torque M_{Pm} of mechanical losses in the pump „working chambers - shaft" assembly.

Fig. 9.4 presents the torque $M_{Pm|\Delta p_{Pi}=0, b_P, \nu_n}$ of mechanical losses in the unloaded pump as a function of the pump capacity coefficient b_P.

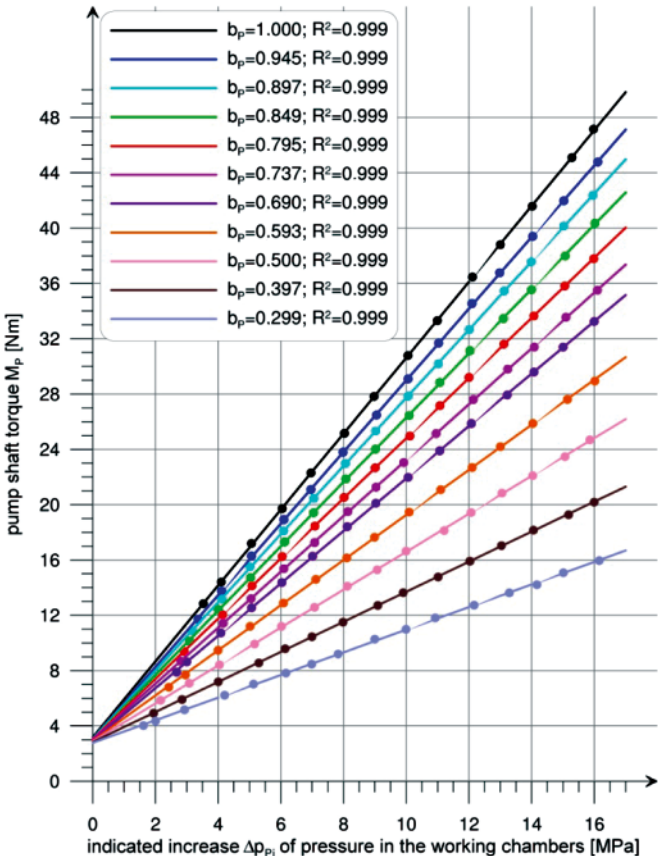

Fig. 9.1. *Torque M_p on the PTOZ2-25 pump shaft as a function of the indicated increase Δp_{Pi} of pressure in the working chambers, for different values of the pump capacity coefficient b_P, at v_n working liquid viscosity [21]*

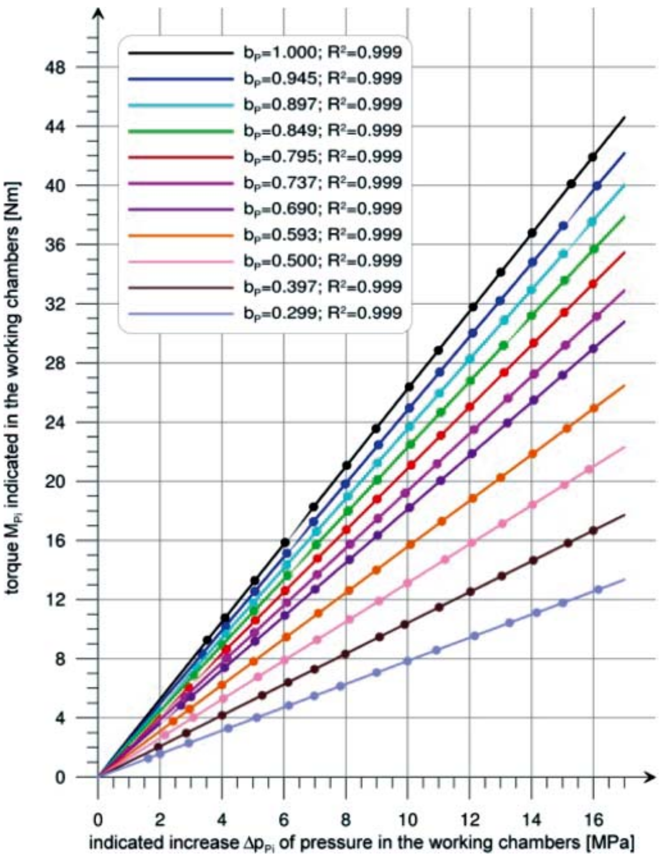

Fig. 9.2. *Torque M_{Pi} indicated in the PTOZ2-25 pump working chambers as a function of the indicated increase Δp_{Pi} of pressure in the working chambers, for different values of the pump capacity coefficient b_P, at v_n working liquid viscosity [21]*

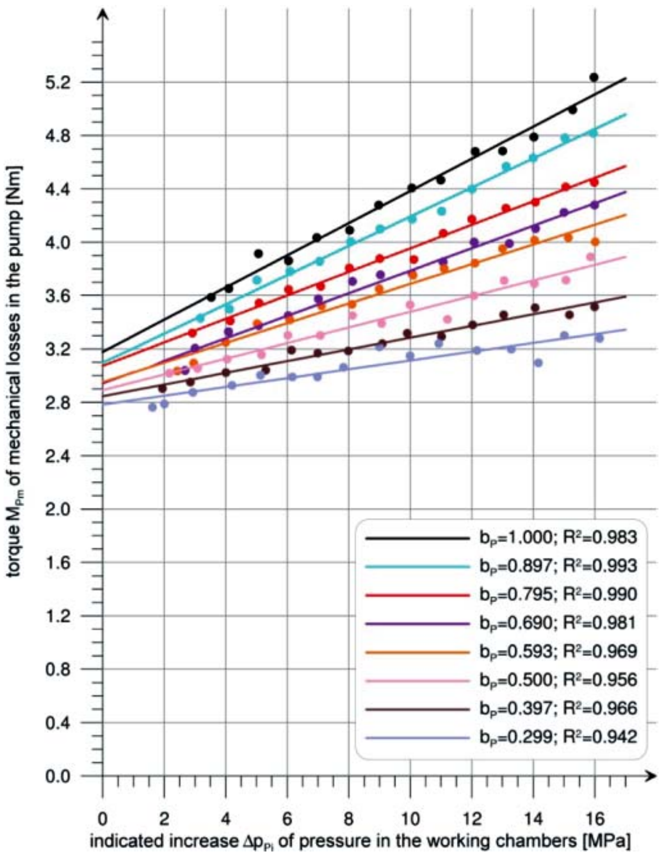

Fig. 9.3. *Torque M_{Pm} of mechanical losses in the PTOZ2-25 pump „working chambers - shaft" assembly as a function of the indicated increase Δp_{Pi} of pressure in the working chambers, for different values of the pump capacity coefficient b_P, at v_n working liquid viscosity [21]*

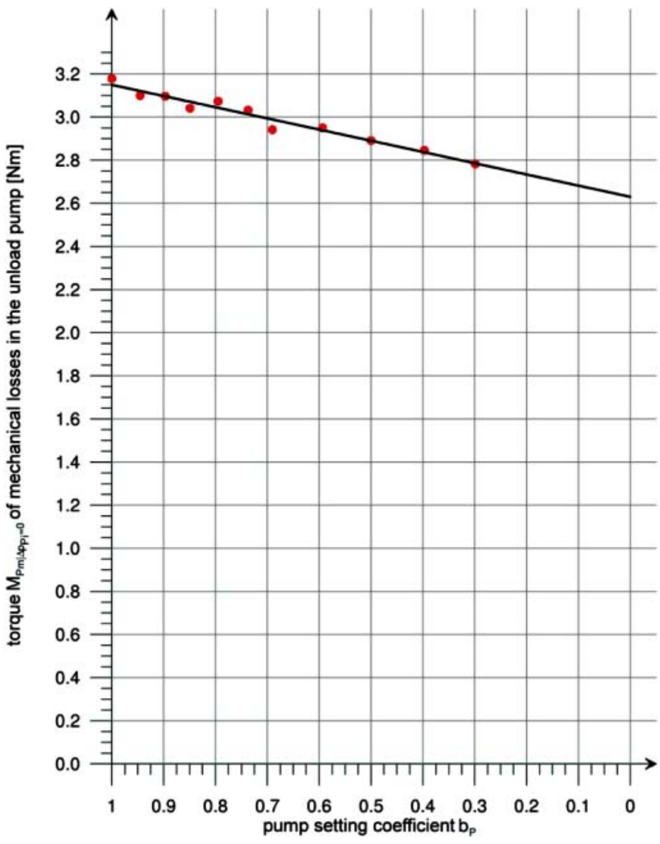

Fig. 9.4. *Torque $M_{Pm|\Delta p_{Pi}=0,\, b_P, v_n}$ of mechanical losses in the unloaded PTOZ2-25 pump „working chambers - shaft" assembly as a function of the pump capacity coefficient b_P, at v_n working liquid viscosity [21]*

Fig. 9.5 presents the values of increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of torque of mechanical losses calculated from the investigation results. The increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ is proportional to the increase Δp_{Pi} of pressure in the working chambers and to the instantaneous value $q_{Pgv} = b_P\, q_{Pt}$ of the pump geometrical working volume.

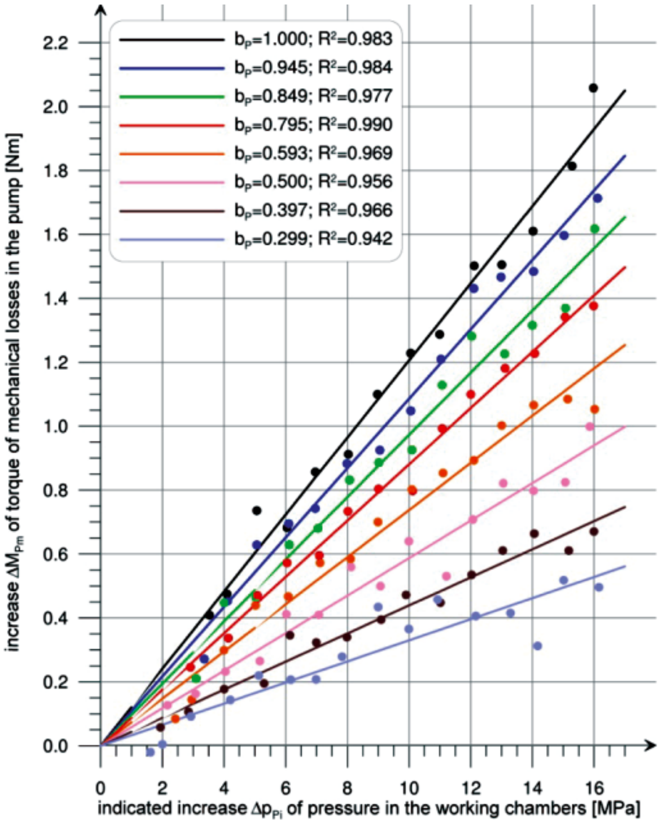

Fig. 9.5. Increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of the torque of mechanical losses in the PTOZ2-25 pump „ working chambers - shaft" assembly as a function of the indicated increase Δp_{Pi} of pressure in the working chambers, for different values of the pump capacity coefficient b_P at ν_n working liquid viscosity [21]

Results of the investigation of the torque M_{Pm} of mechanical losses in the PTOZ2 - 25 pump „working chambers - shaft" assembly confirm the mathematical models ((8.9), (8.12)) of mechanical losses in the displacement pump at reference viscosity $\nu_n = 35 mm^2 s^{-1}$.

The above presented results can be achieved with the assumption of little hydraulic oil compressibility, i.e. compressibility with insignificant effect on the change of pump theoretical working volume q_{Pt} (V_P) of the working chambers (or the geometrical working volume $q_{Pgv} = b_P\, q_{Pt}$) under the influence of the change of indicated increase Δp_{Pi} of pressure in the working chambers.

The picture of the increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of the torque of mechanical losses in the pump „working chambers - shaft" assembly, complying with the mathematical model (equations (8.9) and (8.12), Fig. 9.5), can be a tool for evaluation of the coefficient $k_{lc|p_n}$ of compressibility of the working liquid used in the hydrostatic transmission system. The picture of $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ different from that model is then a result of the liquid compressibility, which reduces the values $q_{Pt|\Delta p_{Pi}=p_n}$ (or $q_{Pgv|\Delta p_{Pi}=p_n} = b_P\, q_{Pt|\Delta p_{Pi}=p_n}$) under the influence of the increase Δp_{Pi} and, in consequence, reduces the value of indicated torque M_{Pi} in the working chambers and torque M_P on the pump shaft.

In consequence, the value of the increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of the torque of mechanical losses, calculated from expression (9.1), decreases as a difference between torque M_P measured on the pump shaft and torque M_{Pi} calculated with the assumption of non-decreased value q_{Pt} (V_P) (or $q_{Pgv} = b_P\, q_{Pt}$). Also decreases the value of coefficient $k_{4.2}$ of the increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$.

10. RESULTS OF INVESTIGATION OF THE TORQUE M_{PM} OF MECHANICAL LOSSES IN THE HIGH PRESSURE PISTON PUMP HYDROMATIK A7V.DR.1.R.P.F.00 TYPE LOADED BY AN OVERFLOW VALVE

Jan Koralewski [22], apart from the verification investigation of the model of volumetric losses of the axial piston variable

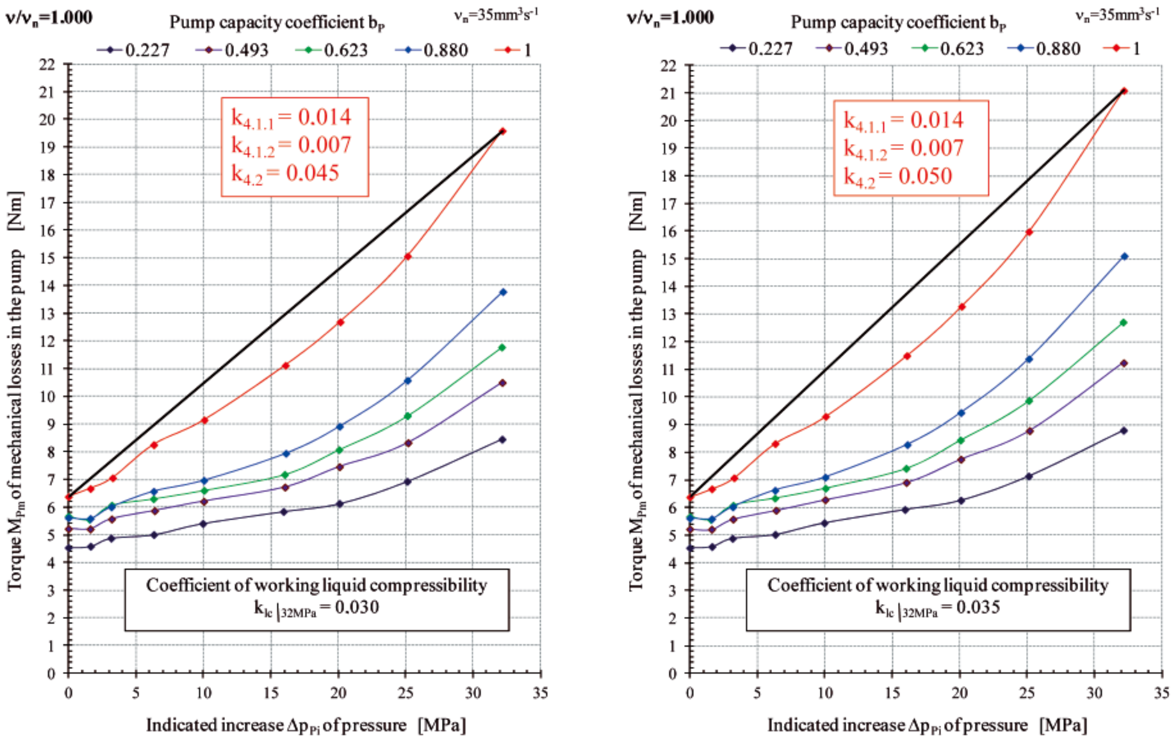

Fig. 10.2. Picture of the torque M_{Pm} of the pump mechanical losses in the pump "working chambers - shaft" assembly as a function of the indicated increase Δp_{Pi} of pressure in the working chambers, at the constant value $\nu/\nu_n = 1$ of the ratio of working liquid viscosity ν to the reference viscosity ν_n, at the pump capacity coefficient b_P and at different assumed values of working liquid compressibility $k_{lc|32MPa}$: 0.030; 0.035 (pump of the HYDROMATIK A7V.DR.1.R.P.F.00 type) [22]

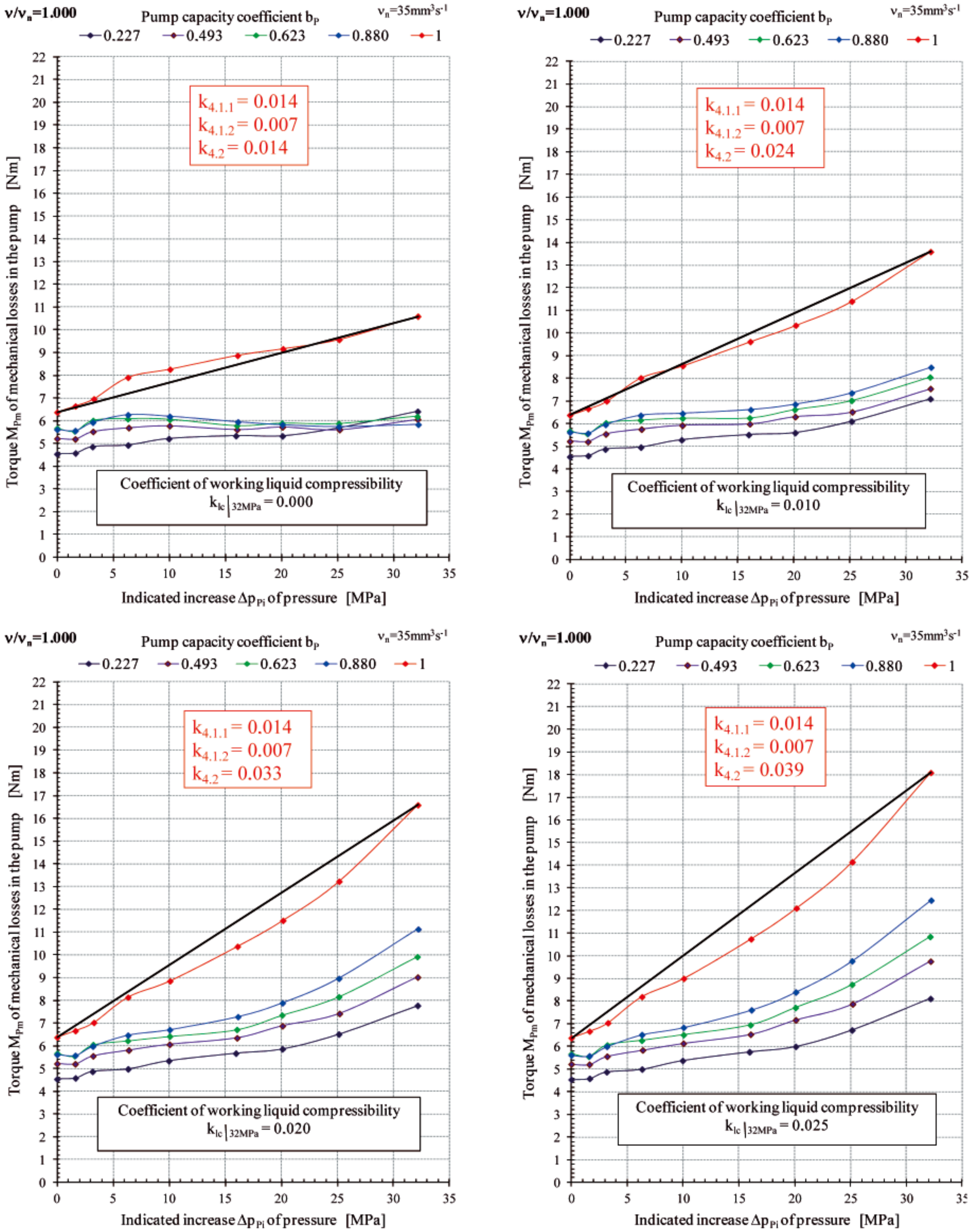

Fig. 10.1. *Picture of the torque M_{Pm} of the pump mechanical losses in the pump „ working chambers - shaft" assembly as a function of the indicated increase Δp_{Pi} of pressure in the working chambers, at the constant value $\nu/\nu_n = 1$ of the ratio of working liquid viscosity ν to the reference viscosity ν_n, at different values of the pump capacity coefficient b_P and at different assumed values of working liquid compressibility $k_{lc|32MPa}$: 0.000; 0.010; 0.020; 0.025 (pump of the HYDROMATIK A7V.DR.1.R.P.F.00 type) [22]*

displacement pump of bent axis design (HYDROMATIK A7V. DR.1.R.P.F.00) (presented in chapter 7), performed, on the test stand presented in Fig. 7.2, the investigation verifying the models (8.9), (8.10), (8.11) and (8.12) of torque of mechanical losses in that pump.

Investigations of volumetric and mechanical losses were carried out simultaneously. The pump was loaded with an overflow valve.

As the investigation of torque $M_{Pm|\Delta p_{Pi}=0}$ of mechanical losses in the „working chambers - shaft" assembly of the unloaded pump (at $\Delta p_{Pi} = 0$) confirmed fully the (8.9), (8.10), and (8.11) models, the investigation of the increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ connected with the indicated increase Δp_{Pi} of pressure in the pump working chambers did not confirm the model (8.12).

Picture of the increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of the torque of mechanical losses determined by indirect method from expression (9.1) at the assumption, that the coefficient of working liquid compressibility is equal to zero ($k_{lc|32MPa}$ = = 0.000), presented in Fig. 10.1, differed significantly from the expected picture from models (8.9) and (8.12) and from Fig. 9.5.

Calculations performed with the assumed coefficient of working liquid compressibility $k_{lc|32MPa} > 0$ showed the picture (Fig. 10.1, Fig. 10.2) of the increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ closer to models (8.9) and (8.12) and to Fig. 9.5.

Picture most similar to those of models (8.9) and (8.12) is diagram $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ obtained with the assumed coefficient of working liquid compressibility $k_{lc|32MPa} = 0.030$ (Fig. 10.2).

With the assumption of the coefficient $k_{lc|32MPa} = 0.030$, **the obtained increase** of the coefficient $k_{4.2}$ of $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of the torque of mechanical losses **was from the value $k_{4.2} = 0.014$ to the value of the order of $k_{4.2} = 0.045$**.

At the same time, taking into account the compressibility of working liquid defined by the coefficient $k_{lc|32MPa} = 0.030$, **the decrease is obtained** of the coefficient k_1 of volumetric losses in the pump (expression (7)) **from the value $k_1 = 0.065$ to the value of the order of $k_1 = 0.035$**.

CONCLUSIONS

1. Energy investigation of the high pressure displacement **variable capacity pump** operating in an aerated hydrostatic system allows to determine **approximate value of the coefficient $k_{lc|p_n}$ of working liquid compressibility** by comparing the picture of the increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of the torque of mechanical losses in the „working chambers - shaft" assembly with the model of increase of that torque.
2. The approximate determination of the value of $k_{lc|p_n}$ coefficient of working liquid compressibility allows to modify the coefficient $k_{4.2}$ of the increase $\Delta M_{Pm|\Delta p_{Pi}, b_P, \nu_n}$ of the torque of mechanical losses in the pump „working chambers - shaft" assembly (towards its increase) and the value of coefficient k_1 of volumetric losses in the pump working chambers (towards its decrease).
3. Therefore, it is possible to determine approximately the effect of the working liquid compressibility on the picture of values and proportion of the mechanical losses and volumetric losses in the pump.
4. Working liquid compressibility reduces the pump theoretical working volume $q_{Pt|\Delta p_{Pi} = p_n}$ of the working chambers (or the geometrical working volume $q_{Pgv|\Delta p_{Pi} = p_n} = b_P\, q_{Pt|\Delta p_{Pi} = p_n}$) under the influence of the change of indicated increase Δp_{Pi} of pressure in the working chambers.
5. The simplifying assumption is adopted, that the sum of the values of working liquid compressibility coefficient $k_{lc|p_n}$ and coefficient k_1 of volumetric losses in the pump equals to the value of coefficient k_1 when it includes also the effect of working liquid compressibility.
6. In the mathematical models of pump energy efficiency and hydrostatic drive system efficiency and also of the system operation field ($0 \le \overline{\omega}_M < \overline{\omega}_{Mmax}$, $0 \le \overline{M}_M < \overline{M}_{Mmax}$), either the coefficient k_1 (which includes also the effect of working liquid compressibility) or the above defined sum ($k_{lc|p_n}$ + + k_1) must be used.

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