INFLUENCE OF THE POSITION OF THE SWASH PLATE ROTATION AXIS ON THE VOLUMETRIC EFFICIENCY OF THE AXIAL PISTONS PUMPS

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Abstract: *The article describes the influence of the position of the swash plate axis of rotation on the size of the dead space and the volumetric efficiency of the axial pistons pump. Superposition of the two motions have been proposed - the rotation of the swash plate with respect to axis intersecting the axis of rotation of the pump shaft and the shifts of swash plate yielding a result of the rotation axis offset effect. In the article it is also made the comparison of the volumetric efficiency of the pump with the axis intersecting with the axis of rotation, and the construction with the shifted axis.*

Keywords: HYDRAULIC DRIVE, HYDRAULIC PUMPS, FLUID COMPRESSIBILITY

1. Introduction

Axial piston pumps are one of pomp types used in power hydraulics. Such pumps are widely used due to their advantages – easy delivery control and high delivery pressures. Depending on structure, these pumps may be divided to pump types with swinging rotor and with swinging swash plate. Figure 1 shows the pump with swinging swash plate. When the shaft 1 rotates, the cylinder drum 5, which is mounted to the shaft, moves pistons 4. Piston feet 3 touch the swinging disk 2, thus forcing the reciprocating movement of pistons, when the cylinder drum rotates. Fixed timing gear 6 touches the cylinder drum, and it connects proper cylinders to either suction or delivery manifold. Pump delivery is controlled by changing disk 2 swing angle.

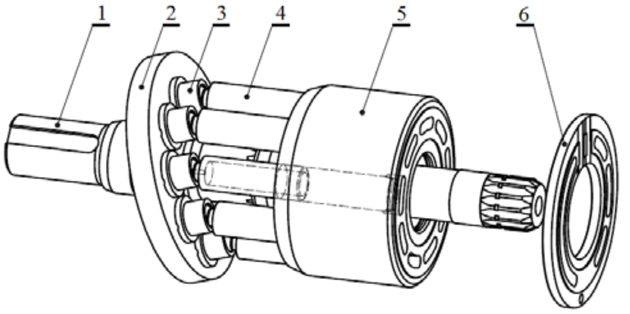

Fig. 1 *Construction of axial piston pump with swinging swash plate: 1-shaft; 2-swinging swash plate; 3-feet; 4-piston; 5-cylinder drum; 6-timing plat*[3].

For all solutions of serially produced pumps, the disk centre line intersects with the shaft centre line[1]. That is forced by structural demands – moments of forces balance with each other for such positions of centre lines only, therefore, relatively small force is needed to adjust the disk position. Such solution has one relatively serious disadvantage – when the disk angle is reduced, so called dead space increases, negatively influencing the volumetric efficiency, which, in particular, is well seen for small disk swing angles[3]. However, the effect of shifted centre line may be obtained by composition of two movements – rotation of the disk around its original centre line and additional disk movement. This is a subject matter of this article.

2. Volumetric efficiency

The volumetric efficiency is a relation of actual flow intensity to theoretical flow intensity, which depends on piston travel and diameter and number of pistons, as well as on pump shaft rpms.

$$\eta_v = \frac{Q_{rz}}{D \cdot tg\varphi \cdot \frac{\pi \cdot d^2}{4} \cdot z \cdot n} \qquad (1)$$

Where η_v is the volumetric efficiency, Q_{rz} – actual flow, D – pistons spacing, φ – angle of the swash plate, d – diameter of the piston, z – number of pistons, n – speed of rotation.

Leakages are the first and main source of volume losses (see Fig. 2). These include, first of all, leakages supplying hydrostatic supports between piston feet and keep plate and between cylinder drum and timing gear disk. The leakages mentioned above are necessary for pump operation. They depend on delivery pressure and medium viscosity.

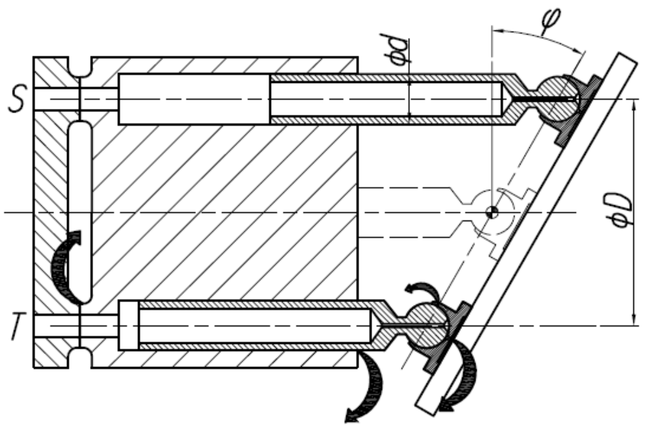

Fig. 2 *Main sources of leakages in the axial piston pump: S- suction manifold, T- delivery manifold.*

Medium compressibility is the other factor influencing the volumetric efficiency. It is seen for high pressures. The coefficient of compressibility β (formula 2)[2] determines the change in oil volume for increasing pressure. The bulk modulus K is a reverse of coefficient of compressibility, and it reaches the value of 1700 MPa for hydraulic oils[1].

$$\beta = -\frac{dV}{V_0} \cdot \frac{1}{dp} \qquad (2)$$

Where β is the coefficient of compressilibity, dV – variation of volume, dp – variation of pressure, V_0 – initial volume

3. Influence of compressibility of fluid in the dead space on the volumetric efficiency

The dead space is the working chamber volume at the end of delivery phase. When the pump runs, each piston makes a reciprocating movement. During suction phase, the piston moves backward and the cylinder chamber is connected to suction manifold, and the medium is sucked. Then, for the extreme backward piston position, the suction manifold is disconnected and the delivery manifold is opened. The piston starts to move the timing gear plate and removes the medium from the cylinder. For the other extreme piston position, oil outflow is disconnected. The residual oil under high pressure is located in the dead space. Thus, when the suction manifold opens, the medium cannot be sucked at the beginning of cycle, because the oil contained in the dead space expands, causing back-flow to the suction duct (figure 3).

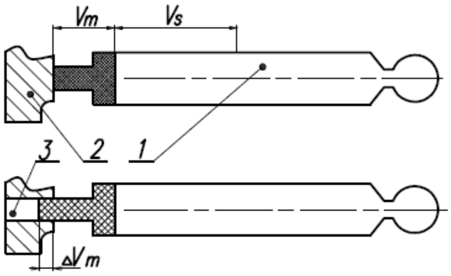

Fig. 3 Back-flow of expanding fluid to the suction manifold: 1-piston; 2-timing plate; 3-suction manifold; V_s-displacement volume; V_m-death space volume; ΔV_m-volume of the expanded fluid.

Therefore, the volume of actually sucked oil is reduced, thus reducing the volumetric efficiency. The higher delivery pressure and the larger dead space, the higher influence of the phenomenon mentioned above. When neglecting leakages and focussing on phenomena resulting from liquid compressibility in the dead space, the pump volumetric efficiency is as follows[2]:

$$\eta_{vp}^{s} = 1 - \frac{V_m}{V_s} \cdot \frac{\beta \cdot \Delta p}{1 - \beta \cdot \Delta p} \quad (3)$$

Where η_{vp}^{s} is volumetric efficiency without the impact of leaks, V_m – volume of the death space, V_s – displacement volume, β – compressilibity factor, Δp – increase of pressure.

4. Change the position of the axis of rotation of the swash plate

For traditional axial-flow piston pumps, minimum dead space capacity corresponds to maximum disk swing angle. When reducing disk swing angle φ, the capacity of dead space increases according to the formula[4]:

$$V_m(\varphi) = V_{mmin} + \frac{\pi \cdot D \cdot d^2}{8} \cdot (tg\varphi_{max} - tg\varphi) \quad (4)$$

Where V_m is volume of the death space, $V_{m\,min}$ – minimal volume of the death space for φ_{max} [mm^3], D – pistons spacing, d – diameter of the piston

The relative dead space capacity ε may be defined by comparing the dead space capacity to the displacement volume[2]:

$$\varepsilon = \frac{V_m}{V_s} \quad (5)$$

For small disk swing angle values, the dead space capacity is even few dozen times larger than the displacement volume, resulting in considerable influence of medium compressibility in the dead space on volumetric efficiency.

Figure 4 shows three positions of swinging disk – position A with centre line (axis of rotation) in point 1 and swing angle φ_{max}, which corresponds to maximum pump delivery, position B with axis of rotation in point 1 and swing angle φ, and position C with shifted axis of rotation located in point 3 and swing angle φ. In case of traditional solution – i.e. when the disk axis of rotation crosses with shaft axis of rotation (point 1) – minimum dead space capacity $V_{m\,min}$ corresponds to maximum disk swing angle φ_{max}. When the disk axis of rotation is located in point 2, the dead space capacity is always constant, and it is $V_{m\,min}$. From meeting the condition that the dead space capacity is not less than $V_{m\,min}$ independently of disk swing angle value, it results that the disk axis of rotation may be located in any point of the straight line section, which connects points 1 and 2. Then, independently of position of disk axis of rotation, the dead space capacity will be $V_{m\,min}$ for disk swing angle value of φ_{max}.

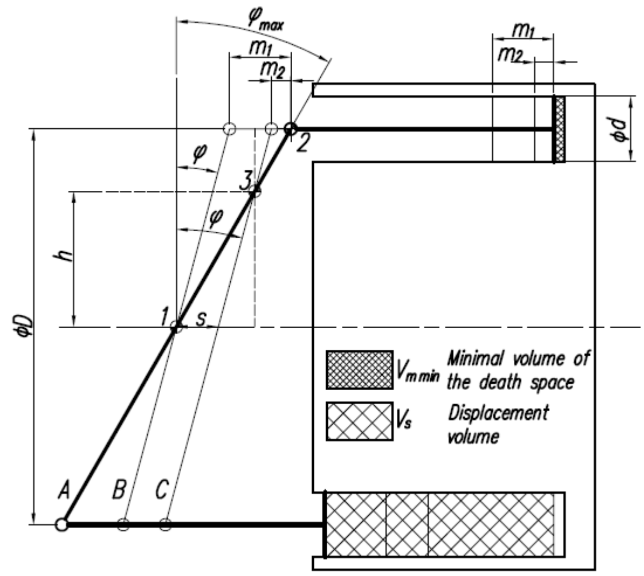

Fig. 4 Influence of shifted axis and rotation of the swash plate on death space volume : A-swash plate with axis of rotation in point 1 and maximum swing angle; B-swash plate with axis of rotation in point 1 and swing angle φ; C-swash plate with axis of rotation in point 3 and swing angle φ.

When the disk axis of rotation is located in point 1, the dead space capacity versus disk swing angle φ is described with formula (3). When the parameter h is introduced, which describes the location of disk axis of rotation:

$$h \in \langle 0, \frac{D}{2} \rangle \quad (6)$$

the formula for dead space capacity versus disk swing angle φ and position of disk axis of rotation h is obtained:

$$V_m = V_{mmin} + m_2 \cdot \frac{\pi \cdot d^2}{4} =$$

$$= V_{mmin} + \left(\frac{D}{2} - h\right) \cdot [tg\varphi_{max} - tg\varphi] \cdot \frac{\pi \cdot d^2}{4} \quad (7)$$

In order to obtain position C of the disk, where disk axis of rotation is located in point 3, the disk shall be rotated around axis of rotation 1 by an angle φ and move the disk towards the cylinder drum by s value:

$$s = h[tg\varphi_{max} - tg\varphi] \quad (8)$$

5. Object of research

The subject matter of testing is the pump, which is modified so that the disk swing angle may be changed and the disk may be shifted in the direction parallel to shaft centre line independently on each other. Combination of those two movements – disk swing angle, when the disk rotates around the axis crossing shat axis of rotation, and disk movement parallel to shaft axis of rotation – gives the effect of shift of disk axis of rotation. The basic parameters of the pump:

- spacing pistons D = 58mm

- piston diameter d = 14,15mm

- maximum swing angle of swash plate 15,56°

- minimal volume of the death space 4,274cm^3

The pump is shown in Figure 5. Swinging disk necks (3) are located in bearing half-sleeves mounted in a cradle (2). The disk axis of rotation crosses the shaft axis of rotation. The cradle may be moved in parallel to the drive shaft axis of rotation along the guides located in the front body part, cradle position is adjusted with the use of two adjusting screws (4). The disk swinging mechanism (1) is independent of disk shifting mechanism. The swinging disk is pressed against bearing half-sleeves with the use of pressing spring (5). Such pump structure makes possible testing the influence of position of disk axis of rotation on volumetric efficiency. Testing to compare traditional and modified positions of disk axis of rotation are made for the same pump model.

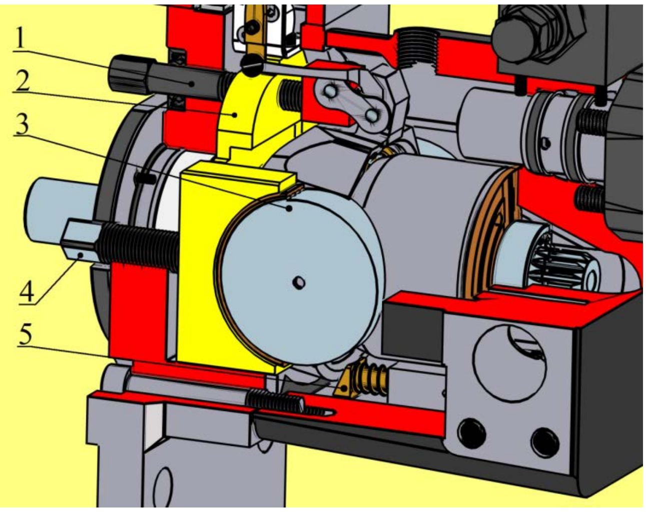

Fig. 5 *The pump, in which the swash plate may be moved in the direction paralel to the shaft centre line: 1-screw angle adjustment; 2-cradle; 3-swash plate; 4-screw adjustment of the cradle; 5-pressing spring[4].*

The diagram of testing station is shown in Figure 6. The tested pump is pre-supplied, so, overpressure of 0.2 MPa is in the suction duct. The pressure is measured upstream and downstream the pump. The intensity of flow is measured by means of the piston flow meter, and the pump is loaded with the use of relief valve. The disk swing angle is determined by measurement of adjusting screw projection. The results are compared with calculated disk swing angle for the lowest pumps delivery pressure (0.2 MPa), where the volumetric efficiency equal to 1 is assumed for that case.

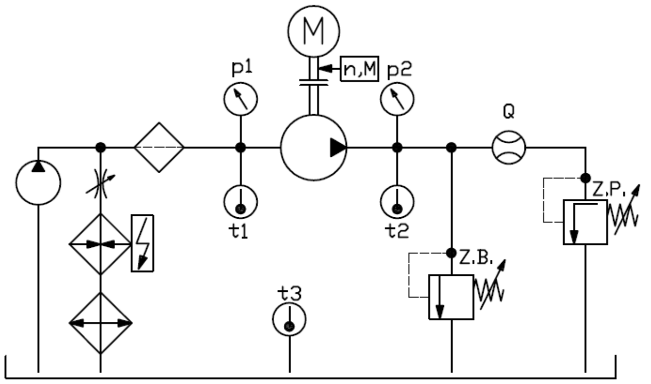

Fig. 6 *The diagram of testing station: p1, p2, p3- manometers; Q- piston flow meter; n- rpm measurement; M- torque measurement; t- termocouple; Z.B.- safety valve; Z.P.- relief valve.*

6. Research methodology and results

The testing is a comparative one, and its aim is to prove that the position and location of swing disk axis of rotation influences the volumetric effectiveness. The testing is carried out as follows – first, measurements of flow and rpms for traditional location and position of disk axis of rotation is carried out, while the pump load is increased gradually. Then, the disk is, without changing the disk swing angle, shifted towards the cylinder drum by such a value of s so that the effect of shifted axis of rotation is obtained (h = D/2). By shifting the disk towards the cylinder drum the dead space volume is reduced to obtain the value of $V_{m\,min}$ = 4.274 cm^3. The change of dead space volume depending on disk swing angle and location of its axis of rotation is shown in Figure 7. When the disk axis of rotation, which crosses the shaft centre line, the volume of dead space increases, when the disk swing angle is reduced. For the disk axis of rotation as shown at the right-hand side in Figure 7 (h = D/2), the volume of the dead space is constant and it does not depend on disk swing angle.

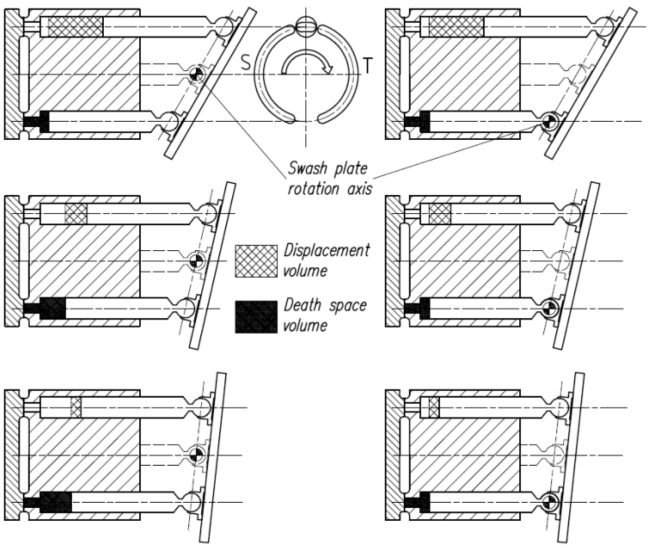

Fig. 7 *Influence of swing angle of swash plate and position of rotation axis of swash plate on death space.*

On the basis of formula (2), theoretical volumetric efficiency curves (see Figure 8) have been determined for parameters of the pump tested (with hollow pistons). Those values have been compared with pump version with solid pistons, for which minimum dead space capacity is reduced from 4.274 to 1.06 cm^3.

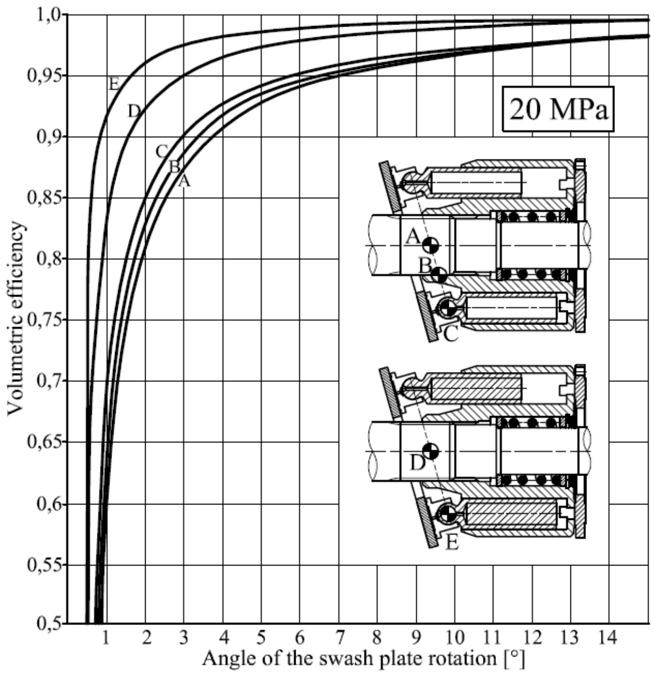

Fig. 8 Influence of disk axis of rotation shift on the volumetric efficiency component resulting from medium compressibility in the dead space for the pressure of p = 20 MPa and oil bulk modulus of K = 1800; A-E – the positions of the axis of rotation of swash plate.

At the beginning, measurements of flow intensity versus load pressure for constant rpms were carried out. On the basis of those measurements volumetric efficiency curves as a function of load were determined for various disk swing angles, as shown in Figure 9. For small disk swing angles, the volumetric efficiency is reduced considerably. For the swing angle of 1.2^o and delivery pressure of 25 MPa, the volumetric efficiency is just 15 %. The data mentioned above are for disk axis of rotation, which crosses the shaft centre line.

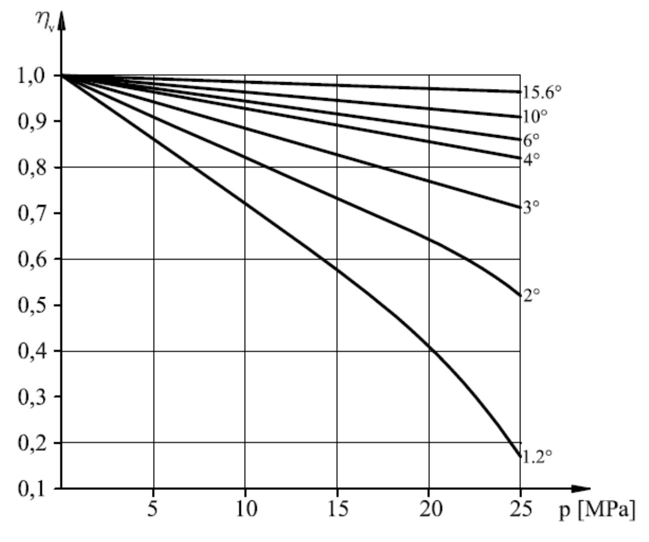

Fig. 9 Volumetric efficiency versus delivery pressure for various disk swing angles for the pump tested. Intersecting the axis of rotation of the swash plate with the axis of shaft . 1000 rpm, viscosity 60cSt.

Then, the results obtained were compared with the results of measurements obtained for disk axis of rotation shifted by the value of h = D/2. The comparison of the results mentioned above is shown in Figure 10.

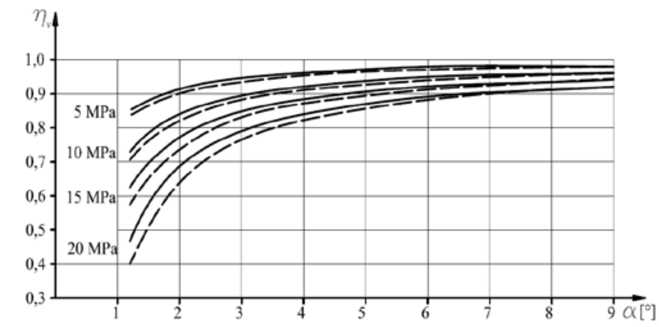

Fig. 10 Comparison between volumetric efficiencies of the pump with disk axis of rotation crossing the shaft centre line (broken line) and the pump with the disk axis of rotation shifted by the value of h = D/2 (full line) in relation to disk swing angle and delivery pressure. Measurements for n = 1000 rpm and oil viscosity of 80 cSt.

7. Conclusions

The testing made as above shows that the volumetric efficiency increases, when the disk axis of rotation is shifted. In particular, the volumetric efficiency increase is seen for small disk swing angles and high delivery pressures. For a given pressure and oil viscosity, leakages do not depend on location and position of the disk axis of rotation. Therefore, the increase in efficiency is connected with reduced dead space capacity. Further testing was made for delivery pressures up to 35 MPa for various oil viscosity and rpms. The measurements already made confirm the influence of oil compressibility in the dead space on the volumetric efficiency.

8. References

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