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ADVANCED AXIAL PISTON SWASH PLATE PUMP PARAMETERS RECOMMENDATIONS

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Abstract

Mechatronics is one of rapidly developing spheres of world machinery and science. In many cases its quality depends on drive unit quality. Sensitivity and efficiency of a hydraulic drive unit are mostly determined by hydraulic machines parameters. For the reasons set out an axial-piston swash plate pump with reduced friction forces was designed.
Present work describes the method of computation of the basic parameters of piston – cylinder bore mechanism that provide operability and improved characteristics of the developed axial-piston swash plate pump, and an experimental proof of this computation. The research object is an axial-piston swash plate pump with full hydrostatic slipper bearing.
As a result of the preliminary tests of the piston – cylinder bore mechanism of the swash plate pump operational and structural restrictions, which influence the interfaces between pump's elements, were found. Thereby an additional force analysis was performed, applied relations were obtained, which lead to detection of the developed pump swash plate angle range that provides its operability and improved dynamic and energy characteristics.
The conditions of operability of the developed pump were found. An appliance of the present hydraulic machine allows reducing friction forces and significantly improving characteristics of the whole mechatronic system.

Keywords: swash plate pump; machinery; hydraulics; mechatronics; drive unit

1. Introduction

In mechatronic systems, such characteristics as efficiency and sensitivity are crucial and represent the quality of the whole mechanism. From this point, characteristics of each component of the system become important. Drive unit – and prime mover in particular – is a source of significant power losses.

In hydraulic mechatronic systems, hydraulic machines act as prime movers. To reduce power losses caused by friction, there should be fluid friction mode in every kinematic pair. [1]

An axial piston swash plate pump with improved energy and dynamic characteristics was designed [2]. The present hydraulic machine has reduced energy consumption compared to analogues. As a rule, in that kind of pumps a ball and socket joint that connects a piston body to a slipper, is coaxial with the piston body. In that case transverse force, that is a component of a swash plate reaction force, inclines a piston axis with respect to a cylindrical bore axis, which leads to high contact stresses where the piston contacts edges of the cylindrical bore and, therefore, to higher friction forces and reducing efficiency of the hydraulic machine [3].

2. Literature review

There is an axial piston swash plate hydraulic machine, contains a housing including a swash plate and a cylinder block with pistons in its cylindrical bores. Pistons and bores form displacement chambers, and each piston has a ball and socket joint that links it with a slipper. The piston ball's center lies on an axis, directed along the normal to the plane of the swash plate and through the bearing surface of the cylindrical bore at every rotation angle of the machine shaft. In this way, there is a displacement of the piston ball, which is provided by a separator, mounted in a special guide [4].

The shown solution for reduction of the friction force between pistons and cylindrical bores increases contact area that also means an alignment of the piston and the bore. However, a design of the hydraulic machine becomes more complicated. In case of high load, a chance of seizure of the separator can cause the machine failure. Moreover, the present structure does not provide fluid friction mode for the whole operation cycle of hydraulic motor and pump.

To reduce piston transverse forces in the swash plate axial piston machines of fixed displacement type Mohamed Elashmawy proposed circumferential cam contour design. In this machine the piston slipper is replaced by a ball that is rotatably mounted within a ball socket formed at the piston end. The ball runs on a circumferential contour groove formed on the swash plate surface. The sliding friction between swash plate and the slipper is replaced by a rolling friction between ball and circumferential runway groove [5].

Although theoretical computations show positive results and obvious advantages of this development, the essence of replacement of the slipper by the ball in the groove is close to the invention of H. Molly, where reaction of a metal body prevents the piston from inclination. Problems of friction and seizure become even more probable as there is no hydrostatic effect to prevent metal surfaces contacting. Less mobility of the piston outer end leads to the same troubles as in the case of H. Molly's design.

To improve fluid film between a piston and a cylinder load-carrying ability Pelosi [3] designed a barreled shaped piston, which also helped to reduce piston micro-motions. However, profits of this solution were proved only with high shaft speed, when hydrodynamic effects have big influence, and there is no confirmation of advantages of this piston on a whole operating range.

The present hydraulic machine contains piston flat hinge with a slipper axis, directed along the normal to the plane of the swash plate. On the piston surface there are grooves around the piston body lying in a plane parallel to the plane of the swash plate, and one of the grooves has a lubrication orifice, providing operating fluid under pressure from the displacement chamber. An intersection point of the axes of the piston body and the slipper is located between these two grooves. In addition, these grooves stay inside the cylindrical bore during the whole operating cycle.

Preliminary theoretical research of the piston – cylindrical bore mechanism of the swash plate hydraulic machine, such as kinematic and force analysis, allowed to predict significant reduction of the friction force due to compensation of the torque caused by the reaction of the swash plate. This result is achieved by the displaced piston – slipper joint, and, therefore, the point of the swash plate reaction application is displaced, in that way, vectors of reaction in the cylindrical bore become coaxial as the total vector of reaction in the cylindrical bore counterbalances the transverse component of the reaction of the swash plate. The flat hinge determines the piston position concerning the swash plate tilt angle.

3. Preliminary experiment

A special test rig was designed to prove the points of the theoretical substantiation of the developed hydraulic machine. The pistons work in antiphase – in pump and motor mode alternately. The cyclogrammes on Figure 1 show a lengthwise component of the friction force in piston – cylindrical bore mechanism, written at different pressure values.

An operating cycle starts from a position matching with the beginning of the pump mode and lasts at low speed in pump and motor mode. This kind of operating is the most problematic as stability of the slipper increases with its speed [6] and the lift generated by the slipper also grows with the rotational speed of the machine shaft [7]. At the swash plate tilt angle of 15° in motor mode, varying pressure full operating pressure range and assuming the piston position from 0° to 135° concerning the starting position, capsize of the slipper was not recorded, though the friction force increases significantly at 270°, which agrees with the decreasing pressing force of the hydrostatic bearing [8].

At the turn of the piston – cylindrical bore mechanism of 135 ° to 180°, which matches the motor mode, the capsize of the slipper was recorded. That fact causes violation of the structure and kinematics of the developed hydraulic machine.

The goal of the present work is to describe this process and to find the lower limit of the swash plate tilt angle providing the developed axial piston hydraulic machine operability and experimental proof of theoretical computations.

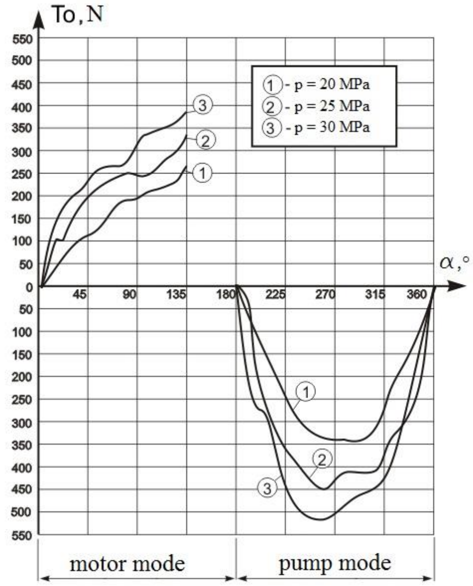

Figure 1. Friction in the cylindrical bore cyclogramme with 15° tilt angle.

4. Theoretical study

The point of the joint between the slipper and the swash plate disclosure is in developed piston design features. For piston axis incline regarding to cylindrical bore axis elimination piston flat hinge has to be displaced to generate torque, opposite to the torque caused by the swash plate tilt angle, in that way, the piston does not rotate around its own axis. Such kinematics causes piston rotation around the cylindrical bore axis and slipper sliding over the swash plate. In case of prevalence of friction forces influence in these kinematic pairs over the reaction of the swash plate effects, that keeps the piston in the operating position, the piston and its slipper are starting to rotate with the cylindrical bore, disclosing the slipper – swash plate joint.

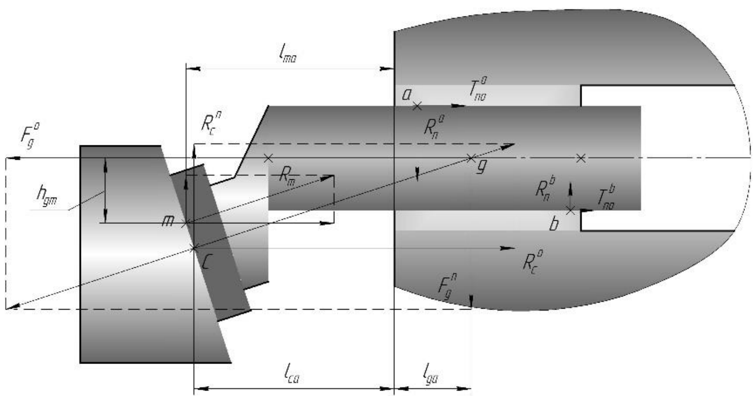

Figure 2. Force model of the piston-cylindrical bore mechanism.

$$R_n^b = \frac{\left[R_m^n l_{ma} - R_m^o\left(h_{gm} + 0.5d_n\right) + F_g^n l_{ga}\right]}{l_{ab}} + \frac{\left[0.5F_g^o d_n + R_c^n l_{ca} - R_c^o\left(h_{gc} + 0.5d_n\right) - T_{no}^b\right]}{l_{ab}} \quad (1)$$

$$R_n^b = \frac{\left[R_m^n (l_{ma} + l_{ab}) - R_m^o\left(h_{gm} - 0.5d_n\right) - F_g^n\left(l_{ab} - l_{ga}\right)\right]}{l_{ab}} + \frac{\left[-0.5F_g^o d_n + R_c^n (l_{ca} + l_{ab}) - R_c^o\left(h_{gc} - 0.5d_n\right) + T_{no}^a\right]}{l_{ab}} \quad (2)$$

while $T_{no}^a = R_n^a f d_n$, $T_{no}^b = R_n^b f d_n$,
$R_n^a -$ reaction of the external edge of the cylindrical bore,
$R_n^b -$ reaction of the internal edge of the cylindrical bore,
$R_c^o -$ lengthwise component of the reaction of the swash plate,
$R_c^n -$ transverse component of the reaction of the swash plate,
$m -$ the point of contact between the swash plate and the slipper,
$c -$ bearing center of the piston mechanism,

$g -$ the point of intersection of the axis, lying along the normal of the swash plate and through the c point, and the axis of the piston
$F_g^n -$ transverse component of the force, acting from the piston to the swash plate, applied at the g point,
$F_g^o -$ lengthwise component of the force, acting from the piston to the swash plate, applied at the g point,
$R_m -$ reaction of the direct contact between the slipper and the swash plate,
$R_m^o -$ lengthwise component of the swash plate reaction,
$R_m^n -$ transverse component of the swash plate reaction,
$a -$ the point of R_n^a application,
$b -$ the point of R_n^b application,
$l_{ab} -$ axial distance between a and b points,
$l_{ma} -$ axial distance between a and m points,
$l_{ca} -$ axial distance between a and c points,
$h_{gm} -$ transverse distance between g and m points,
$h_{gc} -$ transverse distance between g и c points,
$d_n -$ the piston diameter,
$l_{ga} -$ axial distance between a and g points,
$T_{no}^a -$ lengthwise component of the piston – cylindrical bore friction force in a point,
$T_{no}^b -$ lengthwise component of the piston – cylindrical bore friction force in b point,
$f -$ coefficient of sliding friction.

As a result a relationship for total reaction in piston – cylindrical bore was obtained:

$$R_n^{\Sigma} = R_n^a + R_n^b \quad (3)$$

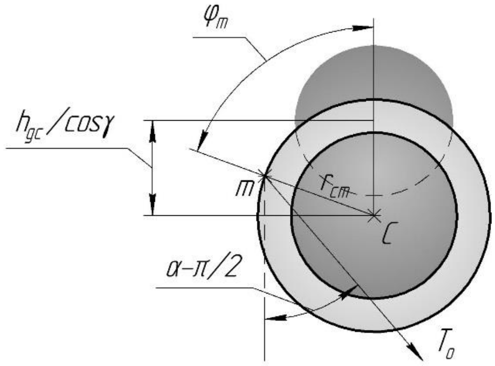

Figure 3. Force model from the swash plate surface.

Present version of the piston mechanism kinematic condition is necessity of the piston rotation relatively to the cylindrical bore with frequency equal to the frequency of the hydraulic machine shaft rotation. This condition is true only if the torque caused by the transverse component of the swash plate reaction in the point of direct contact with the slipper exceeds torques caused by friction forces in kinematic pairs of the piston – cylindrical bore mechanism.

$$M_{R_m} > M_{T_o} + M_{T_n} \quad (4)$$

where $M_{R_m} = \Psi \frac{\pi d_n^2}{4} p \tan\gamma \, r_{cm} \sin(\varphi_m)$,
$M_{T_o} = f\Psi \frac{\pi d_n^2}{4} p \, sin\left(\alpha - \frac{\pi}{2}\right) \cdot (h_{gc} - r_{cm}\cos(\varphi_m) \cdot cos\gamma) + f\Psi \frac{\pi d_n^2}{4} p \, cos\left(\alpha - \frac{\pi}{2}\right) \cos\gamma \cdot r_{cm} \, sin(\varphi_m) -$ the friction force between the slipper and the swash plate torque,
$M_{R_m} -$ the torque caused by the reaction R_m,
$\Psi -$ slipper pressing to the swash plate surface coefficient,
$p -$ operating liquid pressure,
$\gamma -$ the swash plate tilt angle,
$r_{cm} -$ m point relatively to c point position radius,
$\varphi_m -$ m point angular coordinate,
$\alpha -$ cylinder rotation angular coordinate,
$M_{T_n} = R_n^{\Sigma} f \, \frac{d_n}{2} -$ the friction forces between the piston and the cylindrical bore torque.

For this case, considering the fact that the friction forces counteract movements of the slipper, the slipper – swash plate joint operation condition can be calculated as:

$$\tan\gamma > f\left(\frac{R_n^{\Sigma}}{F_p\Psi}\cdot\frac{d_n}{2r_{cm}} + |cos\alpha|\frac{h_{gc}}{r_{cm}} + |sin\alpha|\cdot\cos\gamma\right) \quad (5)$$

In this way, the relationship for the minimum swash plate tilt angle, which provides the operability for the developed hydraulic machine, was obtained. The calculation according to this formula showed, that the tilt angle, which saves the slipper from the capsize, must be not less than 16°.

5. Experimental

To verify technical implementation of the present constructive option of the piston mechanism the bearing between the slipper and the swash plate, providing the tilt angle of 18°, was made. Wherein, although the vector of the swash plate reaction went beyond the cylindrical bore, the operability of the piston mechanism was provided throughout the whole operation cycle, the friction force between the piston and the cylindrical bore was significantly reduced.

Figure 3 shows the cyclogramme of the lengthwise component in the piston – cylindrical bore interface with different shaft rotation frequency, while the piston mechanism with the displaced flat hinge.

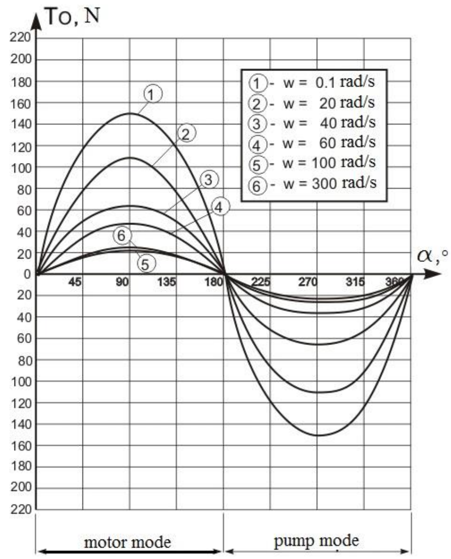

Figure 3. Friction in the cylindrical bore cyclogramme with 18° tilt angle

The developed axial piston swash plate hydraulic machine friction forces in piston – cylindrical bore interface reduced in more than 10 times comparing to the standard design. Better difference was obtained at high pressure values (30-45 Mpa).

6. Results

Firstly of all it was found, that the developed piston – cylindrical bore mechanism has restrictions of its operability. Secondly, boundaries of mechanism operability were defined. The third point is that a method of the operability boundaries computation was formed. The next item is that the experimental verification of the operability range, calculated with the use of the present method, was conducted. The fifth point is that friction forces in the developed swash plate hydraulic machine were significantly reduced.

7. Conclusion and further work

In summary, the present work experimentally substantiates the existence of the limit swash plate tilt angle, which provides the operability of the advanced piston mechanism, and describes the way to find certain values of this limit. This information makes application of the developed piston mechanism possible. Preliminary results of friction forces reduction were reported and discussed in an article by Larchikov, Stazhkov, Yurov [10].

Results of this work allow to design a working sample of the advanced axial piston swash plate hydraulic machine. With this hydraulic machine a hydraulic mechatronic system based on delivery control with improved sensitivity and efficiency can be created.

It is planned to conduct complementary research for elaboration of the permissible swash plate tilt angle. Further work also includes research of the flat hinge as its constructive design may influence its capacity [9].

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