HYDRODYNAMIC PROCESSES IN PISTON–BORE INTERFACE OF AXIAL PISTON SWASH PLATE MACHINE.

Anton Kuzmin, Valeriy Popov & Sergey Stazhkov

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Abstract

Kinematic analysis and numerical calculations for working fluid pressure field of piston – cylindrical bore interface is presented in this article. Possible types of piston motion relative to cylindrical bore are examined. Results of this analysis are the basis for Reynolds equation generation, that describes hydromechanical processes in piston – bore interface. In the present article, a simulation model of hydromechanical processes considering the studied kinematics is shown. The piston mechanism operation at low speed is examined and pressure fields of working liquid in the gap of the piston – bore gap are obtained. This simulation model is the basis for determination of constructive parameters and operating conditions that are leading to establishment of fluid friction regime in piston – cylindrical bore interface.

Keywords: axial piston machine; swash plate machine; hydrodynamics; friction; piston – cylinder interface

1. Introduction

Axial piston swash plate hydraulic machines (swash plate machines) are important components of different actuators. These hydraulic machines are remarkable for its design simplicity and manufacturability compared with other power hydraulic machines such as axial piston bent–axis hydraulic machines. Among other advantages it has less mass, smaller overall dimensions and high efficiency. The present machines are used as pumps and motors as well, and besides swash plate machines are often used in both modes by rotation. It is also frequently required for these machines to be able to reverse its shaft rotation.

To improve a quality of swash plate machines design and to meet the requirements been said, it is expediently to reduce friction forces between a piston and a cylindrical bore. Due to the construct of the swash plate machine, a location of a hydrostatic bearing of the piston slipper on a swash plate in particular, an axis of a cylindrical part of the piston is inclined relative to an axis of the cylindrical bore. Therefore, friction forces values can be reduced by providing an alignment of the axis of the elements. One of possible solutions of the present problem is in using hydrodynamic counteracting force that is caused by piston motion relative to the bore. This force establishes a moving cylinder concentric to a stable cylinder, as reduction of fluid layer thickness causes a local pressure peak.

This article is focused on obtaining of a mathematical model of hydrodynamic processes in the piston pair. This model will allow to describe a process of transition from mixed to liquid lubrication. The aim of the present work is to find numerical values of a pressure field in the gap between the piston and the cylindrical bore. Results of this investigation will be further confirmed experimentally.

2. Literature review

In his works Slyozkin [1] described hydrodynamical parameters and calculation dependencies of an operating fluid in a lubricating layer between two concentric cylinders. In these studies, the pressure field is calculated using Reynolds equations for the lubricating layer and solving the obtained equation system relative to velocities. In more detail, the case of concentric cylinders was examined in research by Korovchinsky [2] using an example of a sliding bearing concerning its real construct.

In his research N.D. Manring [3] presented a force analysis of the piston–bore kinematic pair and a dependency for minimum fluid-film thickness. U. Wiecek and M. Ivantysynova [4] demonstrated a model of a gap flow in the piston pair. Y. Fang and M. Shirakashi [5] described conditions of metal contact in the piston–bore interface and a way to express characteristics of mixed lubrication. M. Pelosi [6] in his work solved a problem of piston–cylindrical bore interaction using elasto-hydrodynamic theory that allowed obtaining more accurate results of the fluid film in loaded conditions. However, this model is calculated for high machine shaft rotation frequency (100 rad/s). In this case, there is fully hydrodynamic lubrication and the piston rotates relative to the interior surface of the cylindrical bore with a frequency equal to the shaft rotation frequency. Investigations by B. Xu, J. Zhang, H. Yang and B. Zhang [7] included radial piston motion to show its influence on the whole carrying ability of the lubricating oil film. J. Jiang and K. Wang added pump flow to this model [8].

Pressure, causing transition from mixed to fully hydrodynamic lubrication, is built up through relative velocities of the piston and the bore. For hydraulic machine shaft rotation frequencies about 0,1–1 rad/s presented pressure is not enough for creating conditions for transition to liquid friction. In this regard, friction forces can take values, enough to define kinematics of the piston mechanism in the following way: rotation happens in spherical hinge between the piston and the slipper, while cylindrical part of the piston partly rolls along the interior surface of the bore. This situation should also be taken into account while describing hydrodynamic processes in the gap.

3. Mathematical model

For operating fluid pressure field in the piston–cylindrical bore interface calculation, Reynolds equations in the lubricating layer are applied.

$$\left\{ \begin{array}{l} \frac{\partial p}{\partial x} = \mu \frac{\partial^2 U}{\partial y^2} \\ \frac{\partial p}{\partial y} = 0 \\ \frac{\partial p}{\partial z} = \mu \frac{\partial^2 W}{\partial y^2} \\ \frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + \frac{\partial W}{\partial z} = 0 \end{array} \right. \quad (1)$$

In works by N.A. Slyozkin [1], M. Pelosi [6] Y.Fang and M. Shirakashi [5] and other authors presented equation system was integrated without considering members of second order of smallness and members including the factor $\partial u/\partial x$. However, while formulating more accurate simulation models, considering deformations of mating surfaces of the elements of the piston mechanism and operating fluid viscosity change, it is also worth considering the following components. For high shaft rotation speed – a possible change of an angle between the axes of the piston and the slipper by the impact of counteracting force caused by operating liquid pressure. For low rotation speed – a circular motion, different from full sliding of the piston surface along the interior surface of the bore.

Integrating the system and using mathematical transformations, the following result is obtained:

$$\begin{aligned} \frac{\partial}{\partial x} \left(h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(h^3 \frac{\partial p}{\partial z} \right) = & 12\mu v + 6\mu u \frac{\partial h}{\partial x} + 6\mu h \frac{\partial u}{\partial x} + 18\mu v \left(\frac{\partial h}{\partial x} \right)^2 + 6\mu h \frac{\partial v}{\partial x} \frac{\partial h}{\partial x} + 6\mu h v \frac{\partial^2 h}{\partial x^2} + \\ & + 6\mu w \frac{\partial h}{\partial z} + 18\mu v \left(\frac{\partial h}{\partial z} \right)^2 + 6\mu h v \frac{\partial^2 h}{\partial z^2} \end{aligned} \quad (2)$$

p – pressure, μ – viscosity coefficient, U – velocity component, tangential to the bore surface, V – velocity component, perpendicular to the bore surface W – velocity of piston motion along the bore axis (considering the piston incline), h – fluid film thickness, v – velocity component, perpendicular to the piston surface, u – velocity component, tangential to

the piston surface, w – velocity of piston motion along its axis (not considering the piston incline), ω – the machine shaft rotation frequency, φ – angular coordinate of a point on the piston surface.

Kinematics of the piston mechanism is defined by ratio of friction forces in kinematic pairs. For the piston operating throughout a duty cycle there are two pairs of rotary motion: the spherical hinge formed by the piston and the slipper, piston–cylindrical bore (Fig.1). This scheme has excess motion, as the rotary motion is caused only by a cylinder block rotation around its axis.

Fig. 1. Kinematic scheme of the piston mechanism.

At the impact of pressure, caused by the velocity of the relative piston to cylindrical bore motion, friction forces in piston–bore interface are significantly reducing, and rotation happens in this kinematic pair. At piston velocities that do not provide pressure, enough for significant friction forces reduction, rotation happens in the piston–slipper kinematic pair.

At the breakaway, friction forces in the piston–bore interface are so high, that in an artificial case of kinematics with rotation in piston–bore pair a joint opening between the slipper and the swash plate occurs [9]. Such a situation ruins the hydraulic machine operability.

In the swash plate machine progress, a case is possible, in which friction forces in the piston–bore prevent piston from rotating: either sliding or rolling along the bore surface (Fig.2). For this situation, the difference between two types of kinematics is visually shown: the piston sliding along the bore surface with a rotary speed, equal to the rotary speed of the shaft, and the piston local rolling along the bore.

Fig. 2. The two types of piston surface rotation

In the present article, only these two cases are described. The intermediate cases will be examined in further works. The final forms of the Reynolds equation for the full piston sliding is:

$$\frac{\partial}{\partial x} \left(h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(h^3 \frac{\partial p}{\partial z} \right) = 6\mu u \frac{\partial h}{\partial x} + 6\mu w \frac{\partial h}{\partial z} \quad (3)$$

Pelosi also examined this case in his simulation model [6]. For the other type of motion formula (2) should be applied, while

$$\begin{cases} v = 2\omega r \cdot \sin\left(\frac{\varphi}{2}\right) \cdot \cos\left(\frac{\varphi}{2}\right) \\ u = 2\omega r \cdot \left|\sin\left(\frac{\varphi}{2}\right) \cdot \sin\left(\frac{\varphi}{2}\right)\right| \end{cases} \quad (4)$$

For pressure field values calculation the tridiagonal matrix algorithm was used [10]. This method is enough for qualitative assessment of processes, occurring in the fluid film in the piston–cylindrical bore interface. Axes directions and schematic grid image (but not correct – the correct grid is defined for the fluid in the gap) are shown on Fig. 3. For presented problem, the method can be written in the following way:

$$\begin{cases} A \cdot P_{j+1} + B \cdot P_j + C \cdot P_{j-1} = F_j \\ A = \frac{h^3}{m_1^2} + \frac{\partial h^3}{\partial z} \cdot \frac{1}{2m_1} \\ C = \frac{h^3}{m_1^2} - \frac{\partial h^3}{\partial z} \cdot \frac{1}{2m_1} \\ B = 2h^3 \cdot \left(\frac{2}{m_2^2} + \frac{1}{m_1^2}\right) \end{cases} \quad (5)$$

For boundary i components the coefficient accepts the following values:

$$B = 2h^3 \cdot \left(\frac{1}{m_2^2} + \frac{1}{m_1^2}\right) + \frac{h^3}{m_2^2} \pm \frac{\partial h^3}{\partial \varphi} \frac{1}{2m_1} \quad (6)$$

Fig. 3. Schematic grid representation

Coefficients α and β are calculated in the next way:

$$\begin{cases} \alpha_{j+1} = \frac{A_j}{B_j - C_j \alpha_j} \\ \beta_{j+1} = \frac{F_j + C_j \beta_j}{B_j - C_j \alpha_j} \end{cases} \quad (7)$$

For $j=1$: $\alpha=0$, $\beta=P_0$. Boundary conditions: on the inner edge of the cylindrical bore $P_0 = 30$ MPa, on the outer edge $P = 0$.

$$P_j = \alpha_{j+1} \cdot P_{j+1} + \beta_{j+1} \quad (8)$$

j – a number by axis OZ , i – a number by angular coordinate φ , m_1 – step along axis OZ , m_2 – step along angular coordinate φ , F – the right side of equations 2 and 3.

4. Pressure field

As a result, a pressure distribution of the operating fluid in the gap between the piston and the cylindrical bore was obtained. Parameters: piston diameter is 3,15 cm, cylinders centers circle diameter is 14 cm, the swash plate tilt angle is 18° , the gap height between the piston and the cylindrical bore in concentric state is $12,5 \mu\text{m}$, the bronze bushing length is 6,15 cm. On Fig. 4-11 pressure fields on the edges of the cylindrical bore (bronze bushing) for shaft rotation speeds of 0.1 rad/s, 10 rad/s and 100 rad/s are presented.

From the values of the pressure field been obtained it can be concluded, that at maximal speed, coaxial with the bore, the rotation motion type has almost no impact on pressure peaks forming. Calculations can be more accurate while solving a complex system, M. Pelosi [6] model for example, with smaller steps values of pressure peaks reduce, while the order stays approximately the same. When the piston has no speed towards or backwards (it is situated at the dead centers),

pressure peaks values on the outer edge of the bushing for sliding motion repeatedly exceed pressure that is formed by motion without sliding.

However, it is possible to conclude preliminarily, that this pressure is not enough to balance an applied to the machine shaft load. These facts give grounds to assert that no-sliding or rolling type of piston motion is more suitable for breakaway and at low speeds.

Fig. 4, 5. Pressure at piston dead centers and full surface sliding

Fig. 6, 7. Pressure at piston maximal speed and full surface sliding

Fig. 8, 9. Pressure at piston dead centers and no surface sliding

Fig. 10, 11. Pressure at piston maximal speed and no surface sliding

The pressure field values are mostly coincide with results of other researches [4] – [8] for the same kinematics of the mechanism, though the pressure field values for the kinematics that had not been described in these works were also found. These results are sufficient to find a speed value that will cause transition to liquid lubrication.

5. Conclusion and further work

The mathematical model of the hydrodynamic processes in the piston – cylindrical bore pair was designed. This model allows to solve the problem of transition from mixed to liquid friction.

The kinematics of the swash plate machine piston motion relative to the cylindrical bore was analyzed considering different types of relative motion. The first type was described by researchers and was examined in detail. The second type of piston motion relative to the cylindrical bore is the partial rolling along the bore (or the bronze bushing) outer edge. This type of motion had not been described in mathematical models of authors [4] – [8]. Presented kinematic dependencies give data that is necessary to solve the Reynolds equations. In future work transitional between these two types of kinematics will be examined. After a number of experiments a full model of the piston mechanism motion and correlations of the friction forces can be obtained.

For pressure field in the gap between the piston and the bore the tridiagonal matrix method was applied. This decision helped to solve the Reynolds equations with sufficient accuracy without designing a more complicated numerical model. Using this method will give more abilities for investigation of different parameters influence on the pressure built up in the gap and friction forces in future.

Numerical results of pressure values of the operating fluid in points that belong to the piston–cylindrical bore kinematical pair were obtained. Using these results it is possible to define constructive parameters that will cause earlier transition to liquid friction.

The results of this research can be a basis for recommendations on developing of the swash plate machine piston mechanism and reducing friction forces at low speed. This will cause less breakaway torque value for the axial piston swash plate machine and smaller dead band. It is also crucial for machines that work in a regime of frequent shaft rotation direction change.

Obtained values need confirming experimentally in further works. The first step is to conduct a number of experiments at a test rig, fixing a moment of transition from mixed to liquid lubrication. The next step is to apply the research results to a real hydraulic machine.

6. References

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