INVESTIGATION OF THE OPERATION OF THE DEVELOPED PISTON MECHANISM OF THE AXIAL-PISTON HYDRAULIC SWASH PLATE MACHINE IN THE STRAINING AND CREEPING SPEEDS MODES

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Abstract

This article is devoted to the research and improvement of axial-piston hydraulic swash plate machines, widely used in power drives for various purposes. The paper presents a refined force analysis of the piston mechanism. Special attention is paid to the research of the friction forces in the "slipper - swash plate" pair. In the straining and "creeping speeds" modes these friction forces, as well as friction forces in the piston pair, can have a significant impact on the dead band of the hydraulic machine. The results of experimental researches of the piston mechanism are presented, which suggest a decrease in friction forces during straining and "creeping speeds" modes in the "slipper - swash plate" pair for the developed piston compared to a standard piston.

Keywords: axial-piston hydraulic swash plate machine; dead band; force analysis; friction force in the "slipper - swash plate" pair; drive.

1. Analysis of friction forces in a pair of "slipper-swash plate" in the straining and "creeping speeds" modes

The axial-piston hydraulic swash plate machine is widely used in energy-intensive drives for various purposes. It has high weight and size characteristics, simplicity of design and, as a result, higher reliability and service life, as well as low cost. However, these hydraulic machines are inferior to axial piston bent axis hydraulic machines in terms of such a parameter as the control range. Such difference is caused by the relatively large dead band when starting and reversing the operation of the hydraulic machine. In this regard, it is relevant to conduct research on axial-piston hydraulic swash plate machines and develop constructive and technological recommendations for the improvement and use of this type of hydraulic machines in high-precision drives. An indirect confirmation of the relevance of these researches is the recent increase in number of publications devoted to this topic.

Many researches are devoted to the power analysis of a hydraulic machine. In particular, Marek Woźniak, Tomasz Szydlowski and Krzysztof Siczek in their work [1] researched the nature of the movement of the slipper. They found that with a reciprocating sliding motion along the axis of the piston, the thickness of the lubricant between the slipper and the swash plate quickly reached a minimum and stabilized.

That is, hydrostatic unloading is realized. Xiangxu Meng, Chang Ge and Hongxi Liang in their research [2] show that the position of the slipper has a significant influence on the pressure distribution, and also propose a grooved sealing pad design. Heon-Sul Jeong and Hyoung-Eui Kim in their article [3] obtained two formulas for estimating the instantaneous frictional force of the piston and the average loss of the friction moment of the piston, taking into account the effects of mixed / boundary lubrication. As a result of further work [4], they presented a model of a hydraulic machine, which makes it possible to evaluate its performance based on the size of the hydraulic machine.

However, in the publications presented above, in the course of power analysis, as a rule, the modes of starting and "creeping velocities" are not considered. In this paper, experimental researches of the friction forces in the piston assembly axial-piston hydraulic swash plate machine in the straining and "creeping speed" modes were conducted. It was assumed, that higher friction forces in the "piston and cylinder" and "piston-slipper" interface lead to the tilt of the slipper. Therefore, the direct contact of its edge and the surface of the swash plate leads to a decrease in the lifting hydrostatic force, an increase in contact stresses and, as a consequence, a significant increase in the friction force in a given kinematic pair.

2. Experimental researches

To research the piston mechanism in the conditions of straining during the transition from pumping to motor mode and from motor to pumping mode, an experimental stand was used [5], shown in Figure 1. The experimental stand is based on the principle of inversion of movement of swash plate machines. The piston groups are assembled in a stationary cylinder block, and the swash plate is driven by an external drive.

Fig. 1. Experimental stand for the piston mechanisms of axial-piston hydraulic swash plate machines research

- 1 - pumping station;
- 2 - block of the rotating swash plate;
- 3 - stationary cylinder block;
- 4 - lever;
- 5 - piston groups;
- 6 - pressure sensor;
- 7 - weight sensor;
- 8 - analog-to-digital converter;
- 9 - computer.

The scheme of the stand is shown in figure 2.

Fig. 2. Scheme of the experiment

A load moment, constant in direction and magnitude, was applied to the lever by a calibrated weight. Hydrostatic pressure acts on the piston, located in the middle position between the dead centers, creating a moment directed against the load moment. During the experiment, an increase in the hydrostatic pressure was carried out, and, consequently, an increase in the torque of the piston mechanism. When the value of the load torque was exceeded, the direction of the resulting torque on the lever changed (transition from pumping to motor mode). With a decrease in torque, a change took place in the opposite direction (transition from a motor to a pumping mod). As a result, the value of the dead band was measured during the transition from one mode to another. Figure 3 shows the researched samples of pistons of standard and developed design [6].

Fig. 3. Standard and developed piston mechanism

Figure 4 shows a graph of the transition from pumping to motor mode (resulting counterclockwise torque). The resulting torque is recorded by a weight sensor and, with the help of secondary equipment, is displayed on a computer. The part of the graph that lies below the abscissa axis corresponds to the pumping mode, above the abscissa axis - to the motor mode. A pronounced dead band is obvious from these graphs. In this case, the dead band of the modified piston mechanism is significantly less than for the standard one.

Fig. 4. The overall moment of force on the swash plate during the transition to the motor mode of operation

Figure 5 shows a graph of the transition from motor to pump mode. When the pressure in the sub-piston cavity decreases, the moment created by the piston mechanism decreases, while the moment from the load on the lever remains constant. Similar to the previous series of experiments, the curve values reflect the same operating modes of the hydraulic machine. At the same time, an increase in the value of the dead band was revealed for both the standard and the modified piston mechanism.

Fig. 5. The overall moment of force on the swash plate during the transition to the pumping mode of operation

A significant difference in magnitude between the dead bands during the transition from pumping to motor mode and from motor to pumping mode can be caused by the action of major and multidirectional friction forces in a pair of "slipper - swash plate". From a comparison of the graphs, it follows that the dead band for the developed piston mechanism changes less significantly than for the standard one. The difference in axial frictional forces between the pumping and motor modes of operation in the «piston-cylinder» pair for the developed piston mechanism in the straining and «creeping speeds» modes at a pressure of 10 MPa is clearly shown in [7] and is no more than 22%. Therefore, the bigger part of the dead band difference is due to friction in the "slipper - swash plate" pair.

Fig. 7. Diagram of forces in the developed piston mechanism taking into account the friction forces between the slipper and the swash plate F_S^M for motor mode (upper figure) and F_S^P for pumping mode (lower figure)

- F_p - force of pressure of the working fluid on the end surface of the piston
 - R - reaction force of the swash plate
 - R_o - axial component of the reaction force of the swash plate
 - R_n - transverse component of the reaction force of the swash plate
 - R_a - reaction force of the outer part of the guide sleeve
 - R_b - reaction force of the internal component of the guide sleeve
 - T_a - friction force of the outer part of the guide sleeve
 - T_b - reaction force of the inner part of the guide sleeve
 - γ - angle of inclination of the swash plate
 - C - center of the hinge
 - d - piston diameter
 - r - piston radius
 - r_c - axial displacement of the piston hinge
 - O - geometric center of the guide sleeve
 - F_S - friction force between the slipper and the swash plate
 - R^M - reaction force taking into account F_S^M in motor operation
 - R^P - reaction force taking into account F_S^P in pumping operation mode
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To determine the friction forces in the slipper and the coefficient of friction between the slipper and the swash plate, it should be assumed that the friction forces in the motor and pumping modes are equal. This assumption can be valid only conditionally, since the angle of inclination of the slipper relative to the plane of the swash plate in pumping and motor modes is unknown. The equation of moments for the piston pair from [8] for the motor mode and the standard piston design:

$$(F_p - T) \cdot \operatorname{tg} \gamma \cdot (l_{CA} + l_{AO}) - R_B \cdot f \cdot r + R_A \cdot f \cdot r - R_A \cdot l_{AO} - R_B \cdot l_{AO} - F_s \cdot \cos \gamma \cdot (l_{CA} + l_{AO}) = 0 \quad (1)$$

$$2R_A \cdot l_{AO} = (F_p - T) \cdot \operatorname{tg} \gamma \cdot f \cdot r + (F_p - T) \cdot \operatorname{tg} \gamma \cdot (l_{CA} + l_{AB}) - F_s \cdot \cos \gamma \cdot (l_{CA} + l_{AO}) \quad (2)$$

$$R_A = \frac{(F_p - T) \cdot \operatorname{tg} \gamma \cdot ((l_{CA} + l_{AB}) + f \cdot r) - F_s \cdot \cos \gamma \cdot (l_{CA} + l_{AO})}{2l_{AO}} \quad (3)$$

$$R_B = \frac{(F_p - T) \cdot \operatorname{tg} \gamma \cdot (l_{CA} + f \cdot r) - F_s \cdot \cos \gamma \cdot (l_{CA} + l_{AO})}{2l_{AO}} \quad (4)$$

$$T^0 \cdot F_p \cdot \frac{l_{AB}}{f} = (F_p - T^0 \cdot F_p) \cdot (\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) + 2f \cdot r)) - F_s \cdot \cos \gamma \cdot (l_{CA} + l_{AO}) \quad (5)$$

$$T_M^0 = \frac{(\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) + 2f \cdot r)) - \frac{F_s}{F_p} \cdot \cos \gamma \cdot (l_{CA} + l_{AO})}{\frac{l_{AB}}{f} + (\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) + 2f \cdot r))} \quad (6)$$

For pumping mode:

$$T_H^0 = \frac{(\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) - 2f \cdot r)) + \frac{F_s}{F_p} \cdot \cos \gamma \cdot (l_{CA} + l_{AO})}{\frac{l_{AB}}{f} - (\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) - 2f \cdot r))} \quad (7)$$

Equations 6 and 7 represent the effect of the friction force in the «slipper-swash plate» pair on the friction force in the «piston-guide sleeve» pair.

$$\begin{aligned} (p_{3H}^M - p_{3H}^H) \cdot S_{\Pi} = \\ F_p^H \cdot \frac{(\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) - 2f \cdot r))}{\frac{l_{AB}}{f} - (\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) - 2f \cdot r))} - \\ F_p^M \cdot \frac{(\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) + 2f \cdot r))}{\frac{l_{AB}}{f} + (\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) + 2f \cdot r))} + \\ k_S \cdot \left(\begin{aligned} &F_p^H \cdot \frac{\cos \gamma \cdot (l_{CA} + l_{AO})}{\frac{l_{AB}}{f} - (\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) - 2f \cdot r))} + \\ &+ F_p^M \cdot \frac{\cos \gamma \cdot (l_{CA} + l_{AO})}{\frac{l_{AB}}{f} + (\operatorname{tg} \gamma \cdot ((2l_{CA} + l_{AB}) + 2f \cdot r))} + \cos \gamma \cdot (F_p^M + F_p^H) \end{aligned} \right) \end{aligned} \quad (8)$$

p_{3H}^H - pressure of transition to pumping mode;

p_{3H}^M - pressure of transition to motor mode;

k_S - coefficient of friction in the «slipper-swash plate» pair.

From these dependencies it follows, that for a standard piston mechanism the pressure of the transition to the pumping mode is 7.34 MPa, the pressure of the transition to the motor mode is 11.72 MPa. Then the coefficient of friction between the slipper and the swash plate will be **0.21**, which corresponds to a pronounced boundary mode of friction with direct contact between the edge of the slipper and the swash plate.

For the developed piston mechanism, the transition pressure to the motor mode was 11.01 MPa, and to the pumping one - 8.46 MPa. Accordingly, the coefficient of friction in the «slipper–swash plate» pair was **0.12**.

4. Conclusions

In the power analysis of a hydraulic machine, as a rule, friction forces in the "slipper-swash plate" pair are not taken into account, implying the implementation of a liquid friction regime due to the presence of partial or complete hydrostatic unloading. In the course of theoretical and experimental researches, the assumption was confirmed about the need to take into account the friction forces in the "slipper-swash plate" pair in the straining and "creeping speeds" modes. In the course of experimental researches, a decrease in the value of the coefficient of friction in the developed piston mechanism in comparison with the standard one by 1.9 times was obtained. Such an improvement in piston characteristics makes it possible to predict a significant decrease in the value of the dead band of the hydraulic machine and, as a consequence, the expansion of its control range. In the future, our team plans to conduct a series of experiments during which the swash plate will work out the input sinusoidal signal. This will allow simulating the tracking mode of the hydraulic drive.

5. References

- [1] Marek Woźniak, Tomasz Szydlowski, and Krzysztof Siczek (2020). The Effect of Dynamics of the Swash Plate–Slippers–Piston Assembly on Friction, *Mechanics and Mechanical Engineering* 2020, Vol. 24, pp. 42-55
- [2] Xiangxu Meng, Chang Ge, Hongxi Liang (2020). Lubrication characteristics of the slipper–swash-plate interface in a swash-plate-type axial piston pump, *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science* 235(4)
- [3] Heon-Sul Jeong, Hyoung-Eui Kim (2004). On the Instantaneous and Average Piston Friction of Swash Plate Type Hydraulic Axial Piston Machines, *KSME International Journal*, Vol. 18 No. 10, pp. 1700-1711
- [4] Heon-Sul Jeong, Hyoung-Eui Kim (2007). A Novel Performance Model Given by the Physical Dimensions of Hydraulic Axial Piston Motors: Experimental Analysis, *Journal of Mechanical Science and Technology*, Vol. 21, pp. 630-641
- [5] Kuzmin, A[nton]; Popov, V[alery] & Stazhkov, S[ergey] (2016). Advanced Axial Piston Swash Plate Pump Parameters Recommendations, *Proceedings of the 27th DAAAM International Symposium*, pp.0556-0561, B. Katalinic (Ed.), Published by DAAAM International, ISBN 978-3-902734-08-2, ISSN 1726-9679, Vienna, Austria
- [6] Stazhkov, S[ergey]; Korobova, I[raida]; Korolev, V[ladimir] & Kuzmin, A[nton] (2019). Hydrodynamic Processes Research of the Positive Displacement Hydraulic Machines Piston Interface, *Proceedings of the 30th DAAAM International Symposium*, pp.0580-0585, B. Katalinic (Ed.), Published by DAAAM International, ISBN 978-3-902734-22-8, ISSN 1726-9679, Vienna, Austria
- [7] Stazhkov, S[ergey]; Kuzmin, A[nton]; Elchinsky, V[iktor]; & Korolev, V[ladimir] (2020). Investigation of the Effect of the Gap on the Friction Forces in the Piston Pair of the Axial Piston Hydraulic Machine, *Proceedings of the 31st DAAAM International Symposium*, pp.0384-0390, B. Katalinic (Ed.), Published by DAAAM International, ISBN 978-3-902734-29-7, ISSN 1726-9679, Vienna, Austria
- [8] Elchinsky, V[iktor]; Kuzmin, A[nton]; Popov, V[alery] & Stazhkov, S[ergey] (2020). Influence of the Design Parameters of the Piston Mechanism on the Dead Band of the Axial-Piston Hydraulic Machine, *16th International Conference on Industrial Manufacturing and Metallurgy, ICIMM 2021, IOP Conf. Series: Materials Science and Engineering*, Ekaterinburg, Russia