UDC 62-822

Kuzmin A. O., Popov V. V., Stazhkov S. M.

# Hydrodynamic processes in the piston and cylinder unit of axial-piston hydraulic machines

The purpose of the research was to analyse the kinematics of the piston mechanism of an axial-piston hydraulic machine with an adjustable-angle cam plate. The kinematic analysis resulted in establishing various types of relative motion of the piston in the guide bushing, writing and solving Reynolds equation with respect to velocities. A sweep method was used to construct a pressure field in the working fluid layer between the piston and the guide bushing. Pressure fields are constructed for several cases of kinematics of the piston mechanism.

*Keywords:* axial-piston hydraulic machines, hydromechanics, hydrodynamics, mechanical engineering, hydraulic drive.

## Introduction

Axial-piston hydraulic machines are used in various hydraulic drive applications, which require high forces, and therefore high pressures of hydraulic fluid. At present, the most widely used in the Russia's industrial sector are axial-piston hydraulic machines with an inclined cylinder block. The most preferable type in terms of weight-and-dimensional characteristics, manufacturability and easy operation are axial-piston hydraulic machines (APHM) with an adjustable-angle cam plate.

The main disadvantage of APHM with an adjustable-angle cam plate is a relatively large dead zone forming at shaft breakaway or reversing. The dead zone is significantly affected by the magnitude of friction forces in kinematic pairs, first of all, friction force in the piston – guide bushing pair. In the footplate – backplate pair, and to some extent in the footplate – piston pair, the friction forces can be reduced by applying a special hydrostatic unloading design, which is rather hard to achieve in the piston – guide bushing pair.

To develop recommendations for reducing friction forces in the kinematic pairs of APHM with an adjustable-angle cam plate, an in-depth analysis of hydromechanical processes in gaps is required. In this respect, a relevant problem is that of constructing a consistent mathematical model describing those processes. A formula for transition from mixed to liquid friction in the piston – guide bushing pair is given in the paper [1]. The first model to simultaneously consider the temperature, pressure, loads acting on the piston mechanism elements and working fluid leakage is presented in the paper [2]. A model accounting for the elastohydrodynamic theory is discussed in the paper [3]. Considered in the paper [4] is a possible change of the angle of piston inclination in the bushing and its impact on the oil film bearing capacity.

Obviously, friction forces in the said kinematic pair can be noticeably reduced due to establishing liquid friction conditions between the piston and the guide bushing. The low-speed motion is not considered in papers [1–4], as it was presumed that piston rotation speed in the bushing was equal to the hydraulic machine shaft speed. The paper [4] considers the case for describing transition to liquid friction in one plane. Given the above, none of the existing models can be regarded as adequate for the case of low speeds.

The purpose of the study is to describe the kinematics of piston relative motion and to construct a pressure field for further determination of the transition to liquid friction conditions.

## Piston kinematics

The character of piston motion depends on the relationship between friction forces in kinematic pairs. Despite the fact that, according to L. N. Reshetov's classification [5], this mechanism is referred to as self-aligning, it has one too many degrees of freedom. Fig. 1 shows that rotary motion conditioned by the movement over the adjustable-angle cam plate surface and by rotation of the cylinder block can take place in both the piston – guide bushing and the piston – footplate pairs. At breakaway, friction forces on the outer edge of the piston-and-cylinder unit are so high that the spherical head of the piston rotates

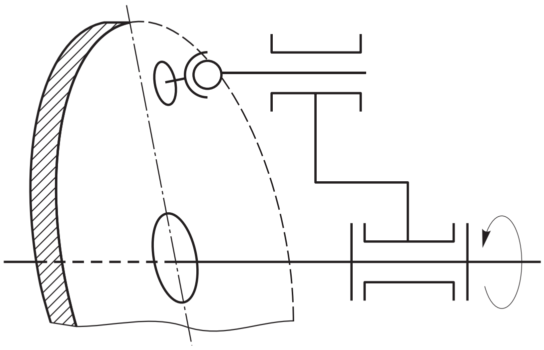

**Fig. 1.** Kinematic diagram of piston mechanism

in the footplate, while the piston can only reciprocate relative to the outer edge of the bushing. In a situation when the piston will have to rotate relative to this part of the guide bushing, the abutment between footplate and adjustable-angle cam plate may open, which will result in machine failure [6].

Further on, with an increase of speed, working fluid pressure in the gap between piston and bushing will reduce friction forces to such an extent that friction forces acting in the piston – footplate pair become predominant, and the cylindrical section of the piston makes a full revolution relative to the guide bushing per one working cycle.

However, in the piston-and-cylinder unit, friction forces act in two areas: on the outer and inner edges. So, when there is no relative piston slip on the outer edge, this friction force is counteracted by friction force in the piston – footplate pair and by friction force on the inner edge. In this way, it can be possible that relative tangential velocity on the outer edge of the bushing is not equal to zero but still remains lower than relative tangential velocity of the piston on the inner edge (Fig. 2). In other words, the point with zero relative tangential velocity may take positions in a span between the surface and centre of the piston cylindrical part cross-section. In the first case, when the point with zero relative tangential velocity is located on the surface of piston cylindrical part, piston surface rolls over the bushing surface on the outer edge, and in the second case, piston surface slides on both boundaries with equal velocity.

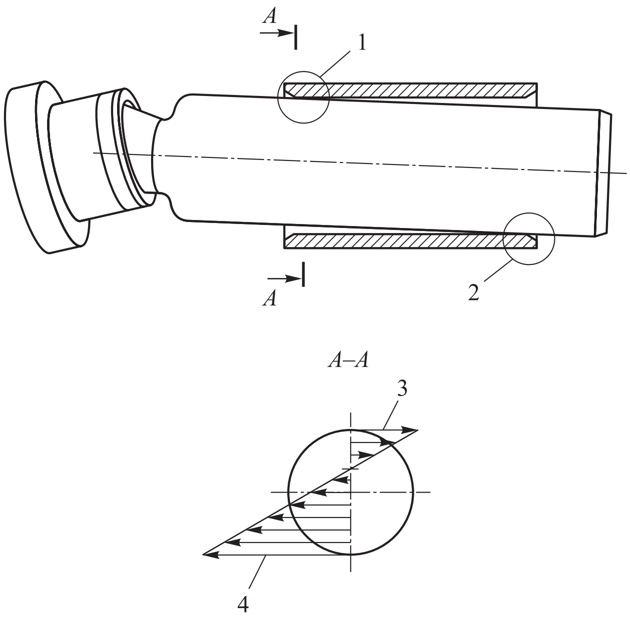


**Fig. 2.** Relative tangential velocity of piston on the outer edge:
1, 2 – contact on the outer and inner edges, respectively;
3, 4 – relative tangential velocity on the outer and inner edges, respectively

We should determine relative velocities of piston surface relative to the bushing in the minimum-gap areas, since it is in those areas that a hydrodynamic wedge is formed. The minimum-gap areas (see Fig. 2) are those located on the bushing edges. Let us consider the dependence of tangentially and normally directed velocities vs. zero-velocity point position. This point may take positions from the piston’s cross-section centre to its surface (Fig. 3).

Distance $L$ is determined by the cosine theorem, and the tangential and normal velocity components are calculated by the sine theorem. Then

$$v = w\left(r - l\right)\sin\varphi; \qquad (1)$$

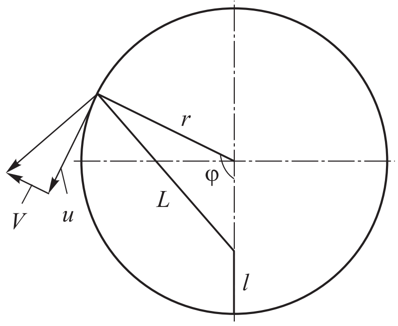


**Fig. 3.** Diagram of velocities of arbitrary point on the surface:
$V$ – normal velocity component relative to bushing

$$u = wL\sqrt{1-\left(\frac{(r-l)\sin\varphi}{L}\right)^2}, \qquad (2)$$

where $v$ – normal velocity component of arbitrary point on the surface;

$w$ – angular rotation speed of hydraulic machine shaft;

$r$ – piston radius;

$l$ – distance from surface to the centre of rotation;

$\varphi$ – angular position of arbitrary point on the surface;

$v$ – tangential velocity component of arbitrary point on the surface;

$L$ – distance from the centre of rotation (point with zero relative tangential velocity) to arbitrary point on the surface.

**Reynolds equations**

Having determined the velocity of points on the surface, we can solve the Reynolds equation for pressure relative to velocities:

$$\begin{cases} \dfrac{\partial p}{\partial x} = \mu\dfrac{\partial^2 U}{\partial y^2}; \\ \dfrac{\partial p}{\partial y} = 0; \\ \dfrac{\partial p}{\partial z} = \mu\dfrac{\partial^2 W}{\partial y^2}; \\ \dfrac{\partial U}{\partial x} + \dfrac{\partial V}{\partial y} + \dfrac{\partial W}{\partial z} = 0. \end{cases} \qquad (3)$$

Here, $p$ – pressure in working fluid layer;

$\mu$ – viscosity coefficient;

$U$ – tangential velocity component relative to bushing;

$W$ – axial velocity component relative to bushing.

Having solved equation (3) with account of additional terms that were not considered in [1–3] (because the motion with piston surface rolling over guide bushing outer edge partially or fully was not discussed there), we have:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right) =$$

$$= 12\mu v + 6\mu u\frac{\partial h}{\partial x} + 6h\mu\frac{\partial u}{\partial x} +$$

$$+18\mu v\left(\frac{\partial h}{\partial x}\right)^2 + 6\mu h\frac{\partial v}{\partial x}\frac{\partial h}{\partial x} + 6\mu hv\frac{\partial^2 h}{\partial x^2} + 6\mu w\frac{\partial h}{\partial z} +$$

$$+18\mu v\left(\frac{\partial h}{\partial z}\right)^2 + 6\mu hv\frac{\partial^2 h}{\partial z^2}, \qquad (4)$$

where $x$ – axis lying on the bushing cross-section circumference (Fig. 4);

$h$ – height of gap between piston and bushing;

$z$ – axis lying on the bushing axis.

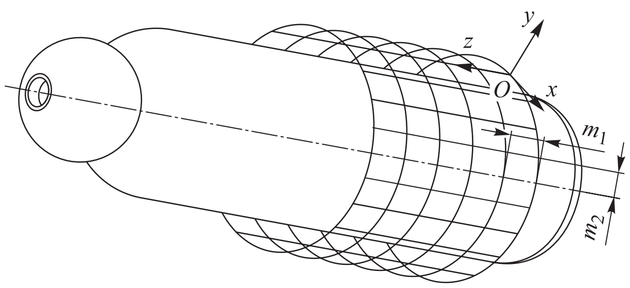


**Fig. 4.** Grid schematic:
$m_1$ – grid step along piston axis;
$m_2$ – grid size along piston circumference

**Results of Reynolds equations solution**

To obtain consistent results, equation (4) was solved by the sweep method [7], which may yield sufficiently accurate results. The method is fairly simple. Pressure fields in the guide bushing were obtained. Fig. 5 shows distribution of the working fluid pressure on the guide bushing outer edge for different positions of the centre of rotation across piston

| ISSN 2542-0542 Journal of “Almaz – Antey” Air and Space Defence Corporation | No. 4, 2017

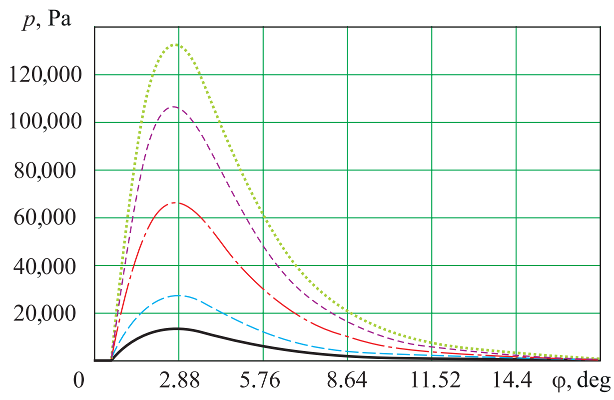


**Fig. 5.** Pressure distribution with zero-velocity point located at a distance from piston cross-section centre equal to: —— – $0.1r$; – – – – $0.2r$; -·-·- – $0.5r$; - - - - – $0.8r$; ······ – $r$

cross-section diameter. In order to determine the influence of rotary motion on the hydrodynamic wedge formation, situations with zero velocity of piston translational movement were considered.

It can be seen from the obtained curves that when the centre of rotation is offset from the piston surface, multiple increase in the the peak pressure value occurs. It means that with friction forces on the outer edge of the bushing reduced to values comparable with the sum of friction force on the inner edge of the bushing and spherical joint of the piston, pressure rises abruptly, which is caused by rotary motion of the piston. For the case of full roll-over, the maximum peak pressure value was about 250 Pa. In this way, even for the centre of rotation offset by 0.1 piston radius, the hydrodynamic force rises significantly. On the whole, the values for full revolution of the piston (see Fig. 5) match those obtained by the authors [1–3].

## Conclusion

For a standard design of APHM with an adjustable-angle cam plate, there is a number of possible kinematic schemes of the piston mechanism. These schemes differ from one another in that the point not rotating relative to the guide bushing, with no account for piston and bushing eccentricity, may take positions between piston centre and a point on the surface in which piston contacts the outer edge of the guide bushing.

In said cases, relative velocities of a point on the piston surface will be varying. In this respect, there are different ways of representing Reynolds equation for solving it with respect to velocities. Hence, the values of pressure peaks produced by piston rotary motion in the bushing also differ from one another.

Based on the obtained values of working fluid pressure peaks on the outer edge of the guide bushing, we may conclude that with an offset of said centre of rotation, the peak pressure values rise substantially, which reduces friction force in the piston-and-cylinder unit when withdrawing from the roll-over implemented at extra high values of friction forces in the piston-and-cylinder unit.

We plan to eventually conduct an experiment in order to prove the kinematics and the values of friction forces in the piston mechanism, as well as to further develop the mathematical model for determining velocity at which transition to liquid friction conditions occurs in the piston – guide bushing pair.

## Bibliography

**1.** *Wieczorek U., Ivantysynova M.* Computer Aided Optimization of Bearing and Sealing Gaps in Hydrostatic Machines—The Simulation Tool Caspar // International Journal of Fluid Power. 2002. Vol. 3. Iss. 1. P. 7–20.
DOI 10.1080/14399776.2002.10781124

**2.** *Pelosi M., Ivantysynova M.* A Geometric Multigrid Solver for the Piston – Cylinder Interface of Axial Piston Machines // Tribology Transactions. 2012. Vol. 55. No. 2. P. 163–174.
DOI 10.1080/10402004.2011.639049

**3.** *Xu Bing, Zhang Junhui, Yang Huayoung, Zhang Bin.* Investigation on the radial micro-motion about piston of axial piston pump // Chinese Journal of Mechanical Engineering. 2013. Vol. 26. Iss. 2. P. 325–333.

**4.** *Manring N. D.* The relative motion between the ball guide and slipper retainer within an axial-piston swash-plate type hydrostatic pump // Journal of Dynamic Systems, Measurement and Control. 1999. Vol. 121. P. 531–537.

**5.** *Reshetov L. N.* Samoustanavlivayushchiesya mekhanismy. M.: Mashinostroeniye, 1972. 265 s. (Russian)
**6.** *Kuzmin A., Popov V., Stazhkov S.* Advanced axial piston swash plate pump parameters recommendations // Proceedings of the 27th DAAAM International Symposium. Vienna, Austria: DAAAM International, 2016. P. 0556–0561.
**7.** *Samarskiy A. A., Nikolaev E. S.* Metody resheniya setochnykh uravneniy. M.: Nauka, 1978. 532 s. (Russian)

**Submitted on 30.11.2017**

**Kuzmin Anton Olegovich** – Design Engineer of the 2nd category, 32nd Department, Design Bureau of Special Mechanical Engineering, Joint Stock Company, Saint Petersburg, Post-graduate student, Baltic State Technical University "VOENMEH" named after D. F. Ustinov, Saint Petersburg.
Science research interests: hydraulic drive, mechanical engineering, axial-piston hydraulic machines, electrohydraulic drive.

**Popov Valeriiy Vladimirovich** – Candidate of Engineering Sciences, Associate Professor, Department of Drive Systems, Mechatronics and Robotics, Baltic State Technical University "VOENMEH" named after D. F. Ustinov, Saint Petersburg.
Science research interests: hydrodynamic processes in groove seals, hydraulic drive.

**Stazhkov Sergey Mikhaylovich** – Doctor of Engineering Sciences, Professor, Head of the Department of Drive Systems, Mechatronics and Robotics, Baltic State Technical University "VOENMEH" named after D. F. Ustinov, Saint Petersburg.
Science research interests: hydraulic drive, mechanical engineering, axial-piston hydraulic machines, mechatronics, robotics.

## Гидродинамические процессы в поршневой паре аксиально-поршневых гидромашин

Проведен анализ кинематики поршневого механизма аксиально-поршневой гидромашины с наклонным диском, в ходе которого были установлены различные типы относительного движения поршня в направляющей втулке. На основе кинематического анализа записано и решено уравнение Рейнольдса относительно скоростей. Применен метод прогонки для построения поля давления в слое рабочей жидкости между поршнем и направляющей втулкой. Построены поля давления для нескольких случаев кинематики поршневого механизма.

*Ключевые слова:* аксиально-поршневые гидромашины, гидромеханика, гидродинамика, машиностроение, гидравлический привод.

**Кузьмин Антон Олегович** – инженер-конструктор второй категории 32-го отдела АО «КБСМ», аспирант Балтийского государственного технического университета «ВОЕНМЕХ» им. Д. Ф. Устинова», г. Санкт-Петербург.
Область научных интересов: гидропривод, машиностроение, аксиально-поршневые гидромашины, электрогидравлический привод.

**Попов Валерий Владимирович** – кандидат технических наук, доцент, доцент кафедры «Системы приводов, мехатроника и робототехника» Балтийского государственного технического университета «ВОЕНМЕХ» им. Д. Ф. Устинова», г. Санкт-Петербург.
Область научных интересов: гидродинамические процессы в щелевых уплотнениях, гидропривод.

**Стажков Сергей Михайлович** – доктор технических наук, профессор, заведующий кафедрой «Системы приводов, мехатроника и робототехника» Балтийского государственного технического университета «ВОЕНМЕХ» им. Д. Ф. Устинова», г. Санкт-Петербург.
Область научных интересов: гидропривод, машиностроение, аксиально-поршневые гидромашины, мехатроника, робототехника.