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ANALYSIS OF THE HYDROSTATIC LOAD OF THE CYLINDER BLOCK IN AN AXIAL PISTON PUMP

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Summary. The paper offers an analysis of the hydrostatic load of the cylinder block – valve plate system in an axial piston pump. Calculation models of hydrostatic pressing and repelling forces operating between the cylinder block and the valve plate are presented. The degree of relief and trajectories of the gravity centre of the resultant hydrostatic forces applied to the cylinder block are investigated.

Key words: hydrostatic forces, cylinder block relief, axial piston pump

INTRODUCTION

Axial piston pumps are applied in a number of industries. It is so, because these pumps can operate with high pressures and high powers, displaying at the same time high efficiency coefficients defined as the ratio of power to mass or volume. This type of displacement machines is usually employed in the drive systems of rather complex devices having greater requirements with respect to efficiency and effectiveness. It is therefore necessary to further improve the exploitation parameters of these machines by modernizing their design [Osiecki 1998].

Designing the valve plate-cylinder block system is one of the most challenging constructional tasks involved in the process of developing the machine. The construction of the valve plate is determined by several crucial parameters, including high efficiency, high pressure, low noise, and high durability.

Typically two types of valve plates are applied in axial piston pumps: flat and spherical ones. In spite of certain disadvantages, the flat type is more popular as its production is simpler [Stryczek 1995].

AIM OF THE PAPER

The aim of the paper is to analyze the hydrostatic forces operating in the gap between the cylinder block and the valve plate. There are pressing forces acting on the bottoms of the cylinders in the cylinder block and the hydrostatic repelling forces pushing the cylinder block apart from the valve plate. The analysis utilizes computer simulations.

ANALYSIS OF HYDROSTATIC PRESSING FORCES

As the cylinder block rotates the number of active pistons under pressure varies. This number is determined by the following formula:

$$i_{max} = z/2 + 0.5$$
$$i_{min} = z/2 - 0.5 \qquad (1)$$

where:

z – the total number of pistons in a pump.

a)

b)

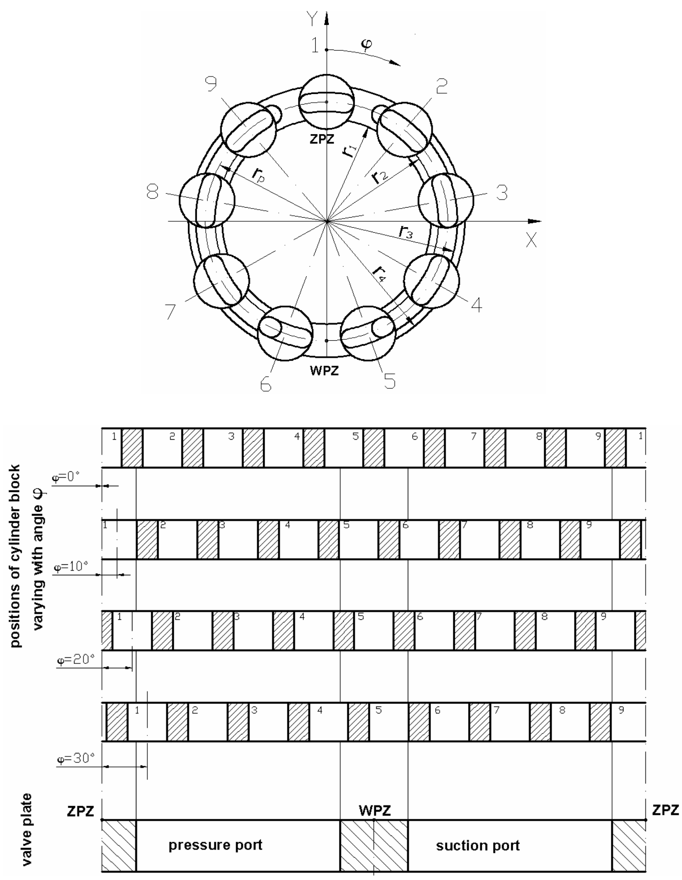

Fig. 1. The cylinder block–valve plate system: a) co-operation of the cylinder block with the valve plate, b) extended view of the cylinder block along the radius r_p for the rotation angle φ = 0°, 10°, 20° and 30° with respect to the valve plate

Fig. 1 presents the cylinder block-valve plate system with variable positions of the cylinder block with respect to the valve plate.

The results of computer simulation investigating the number of active pistons under high pressure are depicted in Figure 2a. The surface of the cylinder block bottom was assumed as A_d = 0.00010432828 m^2 and pressure as p_t = 32 MPa and the resultant hydrostatic force applied to the cylinder block was obtained as shown in Fig. 2b.

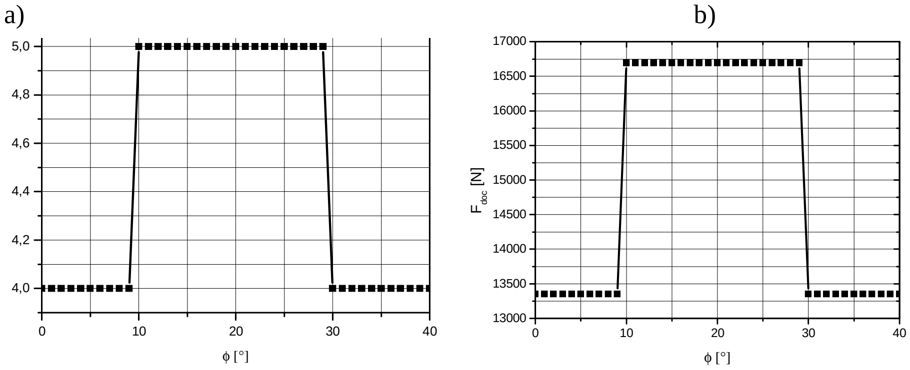

Fig. 2. The four and five-piston zone in the cylinder block – valve plate system depending on the cylinder block rotation angle φ: a) number of active pistons in the pressure zone, b) values of the resultant hydrostatic forces

As the cylinder block rotates, the cylinders and the forces operating therein change their location. The resultant hydrostatic force moves along a certain trajectory. The coordinates of the gravity centre for the resultant hydrostatic force were obtained as follows:

$$Xc = \frac{R\sum_{i=1}^{k} F_i \cdot sin\,\varphi_i}{\sum_{i=1}^{k} F_i} \tag{2}$$

$$Yc = \frac{R\sum_{i=1}^{k} F_i \cdot cos\,\varphi_i}{\sum_{i=1}^{k} F_i} \tag{3}$$

where:

F_i – force operating in the i-th cylinder,
R – radius of the gravity centre of the active cylinder bottom surface,
i – consecutive number of cylinder,
k – current state of active cylinders in the pressure zone [(z+1)/2 or (z-1)/2],
φ_i – angle assigned to the i-th cylinder

If the forces F_i are equal in the cylinders under consideration equations (2) and (3) become

$$X'c = \frac{R}{k} \cdot \sum_{i=1}^{k} sin\,\varphi_i \tag{4}$$

$$Y'c = \frac{R}{k} \cdot \sum_{i=1}^{k} cos\,\varphi_i \tag{5}$$

Fig. 3 presents the trajectory of the local gravity centre of the resultant force derived from the forces applied to the cylinder bottoms in the pressure zone. Fig. 3 shows discrete transition from the four-piston zone to the five-piston zone.

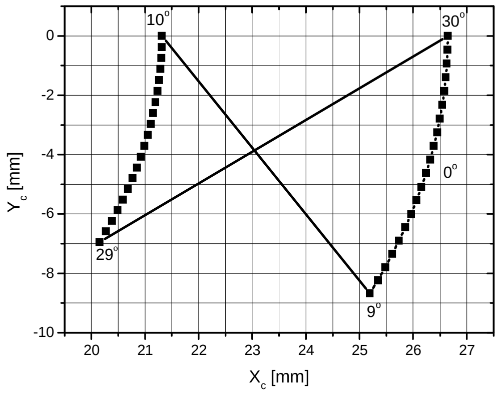

Fig. 3. Trajectory of the gravity centre of the resultant pressing force applied to the cylinder block in the pressure zone

ANALYSIS OF THE HYDROSTATIC FORCES PUSHING THE CYLINDER BLOCK AWAY FROM THE VALVE PLATE

The forces pressing the cylinder block towards the valve plate are counteracted by the force pushing them apart (the repelling force), which consists of the aerodynamic lift resulting from oil pressures occurring in the pressure port and the suction port as well as the aerodynamic lift resulting from oil pressure occurring in the gap between the sealing surfaces.

Figure 4 presents a diagram of a frontal ring gap with parallel walls occurring between the rotating cylinder block and the valve plate [Ivantysyn and Ivantysynova 2001].

The investigation of the liquid flow in the gap are based on the following assumptions:

– the flow is laminar,
– the co-operating surfaces are rigid,
– the gap is of small height and is completely filled with oil,
– the tangent stress in the liquid is subject to the Newton law,
– the liquid is incompressible and of constant viscosity,
– the liquid particles directly adjacent to the rotating surfaces preserve the liquid velocity,
– the liquid inertia forces are negligible.

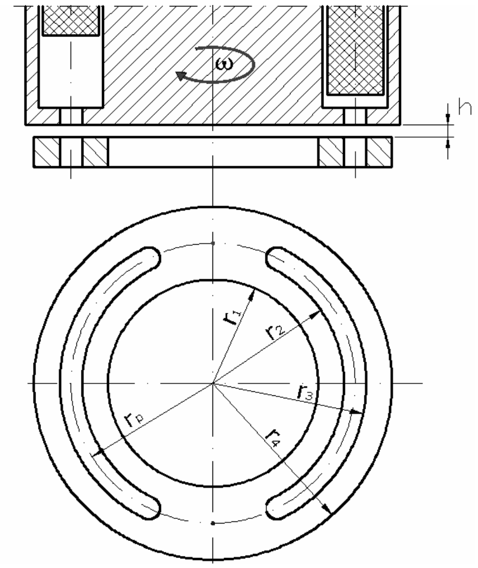

Fig. 4. Diagram of a ring gap with parallel walls

During a revolution of the cylinder block the angular scope of the pressure, suction, and transition zone pressure occurring in the upper and lower bridge varies. Figure 5 presents a diagram of the pressure distribution for the starting position of the cylinder block.

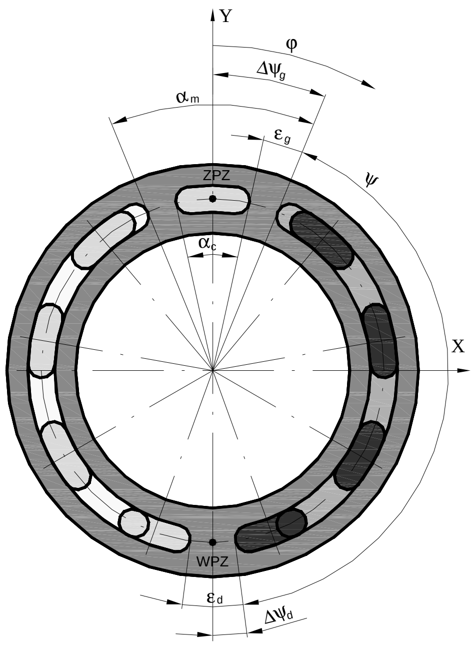

Fig. 5. Alignment of the cylinder block with respect to the valve plate in the starting position

The angular scopes of the distribution of the pressure ψ, upper transition zone pressure ε_g, as well as the lower transition zone pressure ε_d for different stages of the cylinder block revolution were determined on the basis of computer simulations and are shown in Figures 6 and 7.

It is assumed that the angular span α_c of the cylinder ports, the angular spans of the upper and lower bridges α_m and the angular spans $\Delta\varphi$ between the pistons are characteristic values of the pump model under consideration. .

The position of the pressure zone ψ in the upper location with respect to the coordinate y is determined by the angular parameter $\Delta\psi_g$, and in the lower location by the parameter $\Delta\psi_d$ (Fig. 5).

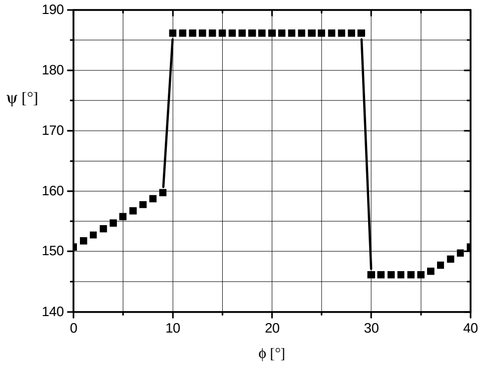

Fig. 6. Angular scope ψ of the high pressure depending on the cylinder block rotation angle φ

a) b)

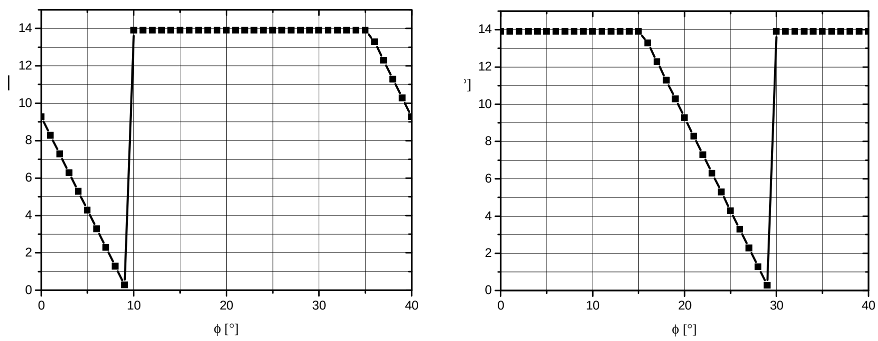

Fig. 7. Angular scopes of the transition zones depending on the cylinder block rotation angle φ: a) upper ε_g, b) lower ε_d

In order to investigate the hydrostatic relief forces depending on the cylinder block revolution, a simulation calculating model was developed. The model operates according to the following procedure:

– determining the geometrical and exploitation parameters of the cylinder block-valve plate system (r_1, r_2, r_3, r_4, α_m, α_c, $\Delta\varphi$, p_t and p_s),

– determining the area where the main zone pressure ψ, the upper transition zone pressure ε_g and the lower transition zone pressure ε_d operate,

– modeling the pressure distribution in the main zone and in the transition zones,

– calculating the hydrostatic relief forces acting on the cylinder block in the main zone and in the transition zones,

– determining the trajectory of the resultant hydrostatic force motion depending on the cylinder block revolution.

In the calculation model the pressure distribution in the main zone ψ from the pressure p_t outside and towards the valve plate was assumed to be a logarithmic function. In the upper transition zone ε_g and the lower transition zone ε_d the change from pressure to suction in the circumferential direction as well as from and towards the valve plate in the radial direction was assumed to be linear.

After substituting the geometrical and exploitation parameters of a Polish axial piston pump into the formulas (q = 32 cm^3, p_{nom} = 32 MPa and n_{nom} = 1500 rev/min) the resultant hydrostatic relief force was obtained by comparing it to the hydrostatic pressing force (Fig. 8a). Figure 9b presents the cylinder block relief coefficient depending on the cylinder block revolution.

According to the data found in the available literature the cylinder block relief coefficient K varies around 90–94% [Baszta 1971], 92–98% [Turza 2005] or 85–95% depending on the calculation model assumed [Guillon 1966, Turza 2005].

The current investigations indicate that the coefficient K is about 85–97% (Fig. 8b).

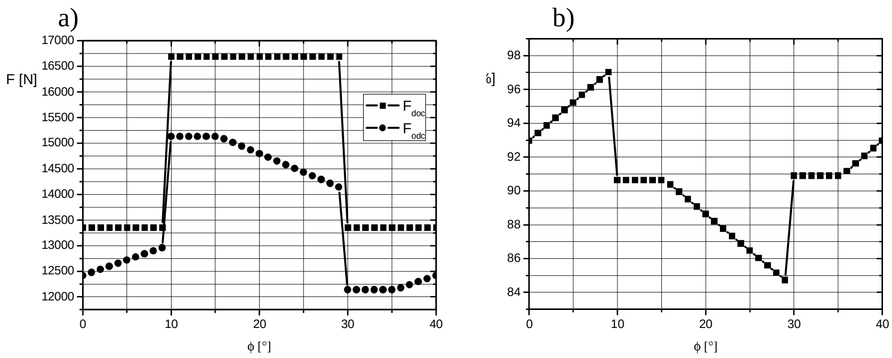

Fig. 8. Cylinder block load depending on the revolution stage: a) values of the pressing forces and of relief forces, b) values of the cylinder block relief coefficient

When the cylinder block is rotating the resultant hydrostatic relief force is moving along a certain trajectory. Figure 9 presents the calculated trajectory of the gravity centre of the resultant hydrostatic relief force in comparison to the trajectory of the gravity centre of the resultant hydrostatic pressing force.

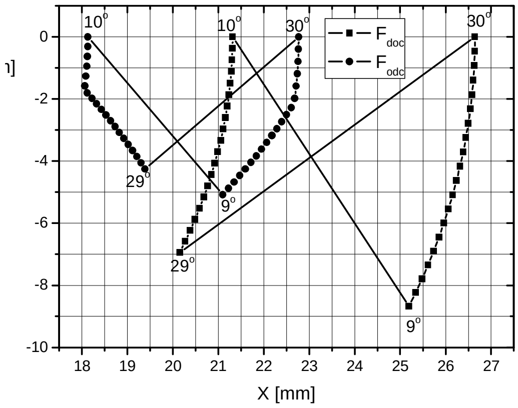

Fig. 9. Trajectories of the gravity centres of the resultant hydrostatic pressing force F_{doc} and of the resultant hydrostatic relief force F_{odc}

It follows from Figure 9 that the trajectories do not match, which may result in the cylinder block assuming a slanted position so that it is necessary to correct the construction of the cylinder block-valve plate system.

CONCLUSIONS

On the basis of the investigations discussed in the present paper the following conclusions can be arrived at:

1. The calculation method described makes it possible to determine the degree of hydrostatic relief of the cylinder block depending on its revolution angle.
2. The trajectory obtained of the gravity centre of the hydrostatic relief forces acting on the cylinder block enables an assessment of the faults (if any) in the cylinder block-valve plate system construction and provides methods of its modification.

REFERENCES

Baszta M. T. 1971: Maszinostroitielnaja gidrawlika. Maszinostrojenie, Moskwa.

Guillon M. 1966: Teoria i obliczanie układów hydraulicznych, WNT Warszawa.

Ivantysyn J., Ivantysynova M. 2001: Hydrostatic Pumps and Motors. Akademia Books International, New Delhi.

Osiecki A. 1998: Hydrostatyczny napęd maszyn. WNT, Warszawa.

Stryczek S. 1995: Napęd hydrostatyczny, tom 1, WNT, Warszawa.

Turza J. 2005: Stanovenie geometrických rozmerov rotačného rozvodu axiálneho piestového hydrostatického prevodníka. Acta Hydraulica et Pneumatica (Slovak Society for Hydraulics and Pneumatics), 2, s. 69–75.