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Impact of exploitation parameters on the hydrostatic relief of the cylinder block in an axial piston pump

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S u m m a r y. The paper presents results of numerical analysis of the variable overpressure-peak induced hydrostatic relief force operating on the cylinder block in axial piston pumps. The hydrostatic relief force of the cylinder block was obtained from the pressure distribution on the valve plate in a variable height gap by means of the numerical Gauss cubature method. The analysis of the overpressure-peak induced hydrostatic relief force was performed as a function of the angular velocity of the cylinder block and of the dynamic viscosity coefficient of oil.

K e y w o r d s: Hydrostatic relief force of the cylinder block, variable-height gap, piston pump.

INTRODUCTION

Multi-piston pumps are applied in a number of industry branches. They operate with high pressures and high powers and are therefore characterized by high power efficiency coefficients, defined as the ratio of power to mass or volume. Because of that such pumps are mostly used in devices which require highly efficient and effective drives [9,14,16,17,18].

When an axial multi-piston pump is operating there appears a small gap between the cylinder block and the valve plate. The gap is filled with oil and due to the slanting position of the cylinder block, the gap height becomes variable (Fig. 1) [4,5,10,11].

The studies of hydrostatic pressure distribution in a variable-height gap have shown that an overpressure peak appears there [6,22].

The paper employs the numerical method of determining the hydrostatic relief force of the cylinder block taking into consideration the overpressure peak occurring in the gap at its lowest height.

PRESSURE DISTRIBUTION OF OIL FILM IN A VARIABLE-HEIGHT GAP AND THE REYNOLDS EQUATION

The pressure distribution of oil film in a variable-height gap on the valve plate in an axial multi-piston

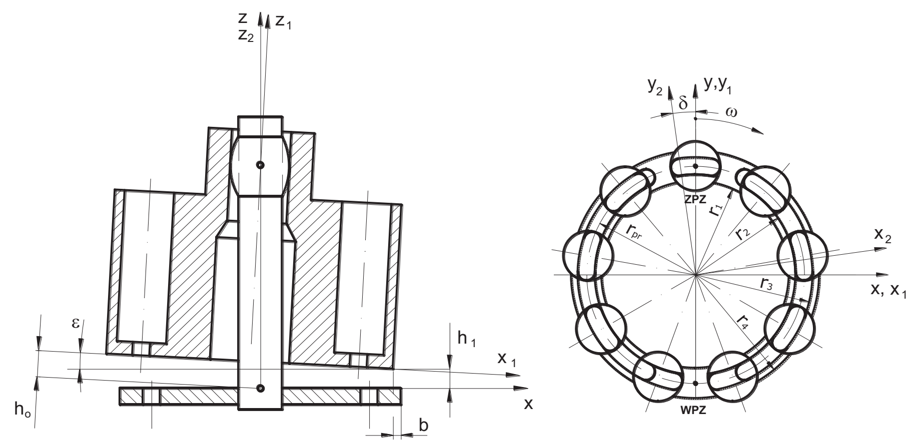

Fig. 1. Regular position of the cylinder block with respect to the valve plate

pump can be described by means of the Reynolds equation (1) [14,12,3]:

$$\frac{\partial}{\partial x}\left(\frac{h^3\rho}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{h^3\rho}{\eta}\frac{\partial p}{\partial y}\right)= \\ =6\frac{\partial}{\partial x}\left(\rho uh\right)+6\frac{\partial}{\partial y}\left(\rho vh\right)+12\rho w, \qquad (1)$$

where: p – the pressure in the gap, h – the gap height, ρ – the working fluid density, η – the dynamic viscosity coefficient, u,v,w – the components of rotational velocity by the prescribed angular velocity ω and the radius vector r of the cylinder block with respect to the directions of the coordinate axes x, y, z.

The solution of the Reynolds equation (1) is valid if the following assumptions are met:
– the flow in the front gap is laminar,
– fluid friction occurs between the adjacent surfaces,
– the lubricant is an incompressible Newtonian fluid,
– the pressure is constant in the direction orthogonal to the surface,
– the adjacent surfaces are rigid.

The task of solving equation (1) analytically is quite complicated, particularly for surfaces of more complex shape. Therefore, the equation was solved numerically using the finite element method, integrated into the computer programme Reynolds developed by the authors of the present paper. The programme offers the possibility of examining the pressure distribution in a variable-height gap on the valve plate, for varying geometrical and exploitation parameters, such as the inclination angle of the cylinder block, its angular velocity, the dynamic viscosity coefficient and the minimal gap height. The programme makes it possible to read input data and to save results in files compatible with the software NuscaS [15].

The programme operates in the following stages:
– reading the finite elements mesh for the valve plate (Fig.2),
– assembling the system of equations [8,20],
– determining the boundary conditions,
– solving the system of equations [2,13],
– saving the results.

The main advantages of the programme are applying such data structures, which do not burden the operational memory with zero elements of sparse matrices in the system of equations and applying the Conjugate Gradient method with Jacobi preconditioning [1] for solving the equations, which significantly fosters the convergence of results.

The examination of the oil film pressure distribution in a variable-height gap on the valve plate proved the existence of overpressure peaks next to the smallest gap height. Fig. 3 presents an example of pressure distribution on the valve plate and the graph representing the variation of oil film pressure next to the smallest gap height at the radius r = 0.0366 m. The pressure distribution was examined for the minimal gap heigh $h_1 = 0.3 \times 10^{-6}$ m, the angular velocity of the cylinder block ω = 157 rad/s, the dynamic viscosity coefficient of oil η = 0,0253 Pas and the angle δ= 0,785 rad of the position of the minimal gap height h_1 with respect to the axis x. The characteristic radiuses of the valve plate are r_1 = 0.0284 m, r_2 = 0.0304 m, r_3=0.0356 m and r_4 = 0.0376 m, respectively.

OBTAINING THE HYDROSTATIC RELIEF FORCE OF THE CYLINDER BLOCK

On the basis of the software Reynolds for calculating pressure distribution on the valve plate, a method was developed for obtaining the hydrostatic relief force, based on the numerical Gauss cubature method [7].

In this method each of the triangular finite elements of the mesh is brought to a standardized right-angled triangle with the nodes (0,0), (1,0) and (0,1) (Fig.4).

The coordinates of the finite element can be obtained in the following way from the standardized triangle:

$$x = x_1 + (x_2 - x_1)\zeta + (x_3 - x_1)\gamma, \qquad (2)$$

$$y = y_1 + (y_2 - y_1)\zeta + (y_3 - y_1)\gamma, \qquad (3)$$

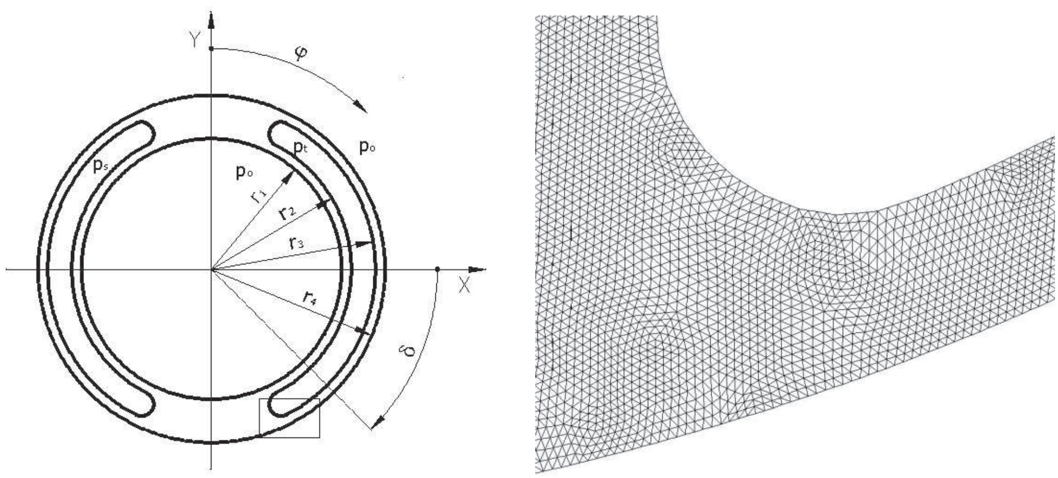

Fig. 2. Computational domain of the valve plate and part of the finite element mesh

where:
(x1,y1) → (0,0); (x2,y2) → (1,0); (x3,y3) → (0,1).

During the standardization the system of coordinates is changed and it is necessary to calculate the Jacobian of the transformation. In the case of triangular elements it is:

$$|J| = \begin{vmatrix} \dfrac{\partial x}{\partial \zeta} & \dfrac{\partial x}{\partial \gamma} \\ \dfrac{\partial y}{\partial \zeta} & \dfrac{\partial y}{\partial \gamma} \end{vmatrix} = \begin{vmatrix} (x_2 - x_1) & (x_3 - x_1) \\ (y_2 - y_1) & (y_3 - y_1) \end{vmatrix} = 2|D|, \quad (4)$$

where:
D – the area of a triangular element.

The value of pressure for the standardized element can be obtained from:

$$p(\zeta,\gamma) = |J|\, p(x,y), \quad (5)$$

where:
p(x,y) – the interpolated value of pressure obtained from the nodal values of the finite triangular element and the shape function [21].

The ultimate formula for the relief force from a triangular finite element is:

$$F_{ele} = \int_0^1 d\zeta \int_0^{1-\gamma} p(\zeta,\gamma)\, d\gamma = \frac{1}{2}\sum_{i=1}^{n} p(\zeta_i,\gamma_i)\, w_i, \quad (6)$$

where:
ζ_i, γ_i – the Gaussian coordinates,
w_i – the weighs of the Gaussian points,
n – the number of the Gaussian points.

The overpressure-peak induced hydrostatic relief force occurring in the neighborhood of the smallest gap height was calculated according to the algorithm below (Fig. 5).

RESULTS OF COMPUTATIONS

The method described above was used for analyzing the overpressure peak-induced hydrostatic relief force of the cylinder block as a function of the angular velocity ω of the cylinder block and the dynamic viscosity coefficient η of oil.

The following input parameters were assumed in the analysis:
– in the pressure port the pressure p_t = 32 MPa,
– in the suction port the pressure p_s = 0.1 MPa,

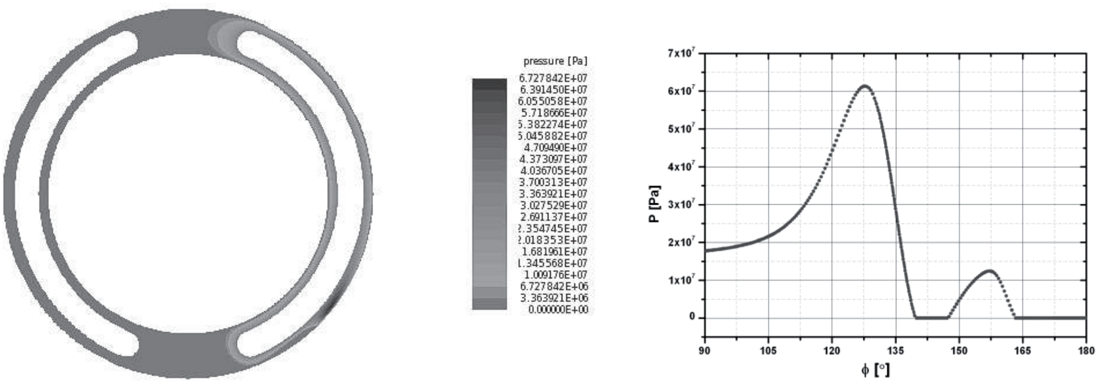

Fig. 3. Oil film pressure distribution on the valve plate and pressure variation next to the smallest gap height as a function of the rotation angle of the cylinder block

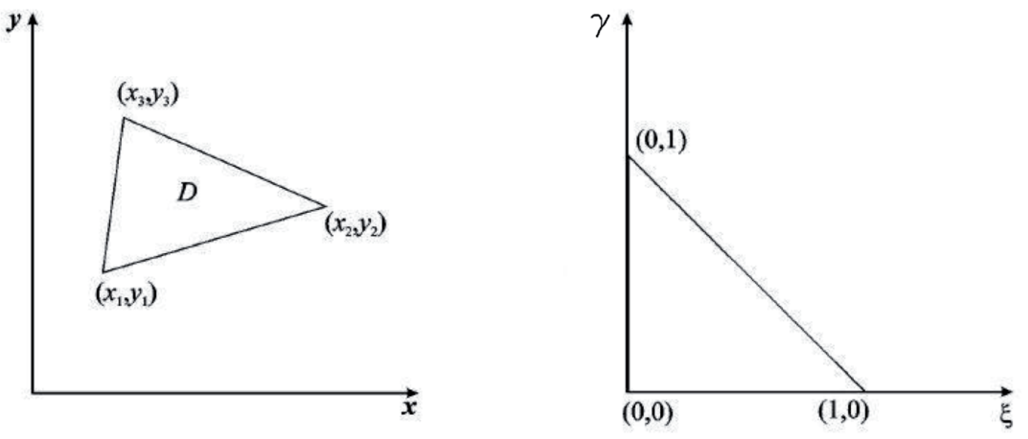

Fig. 4. Transformation of the finite triangular element to the standardized right-angled triangle

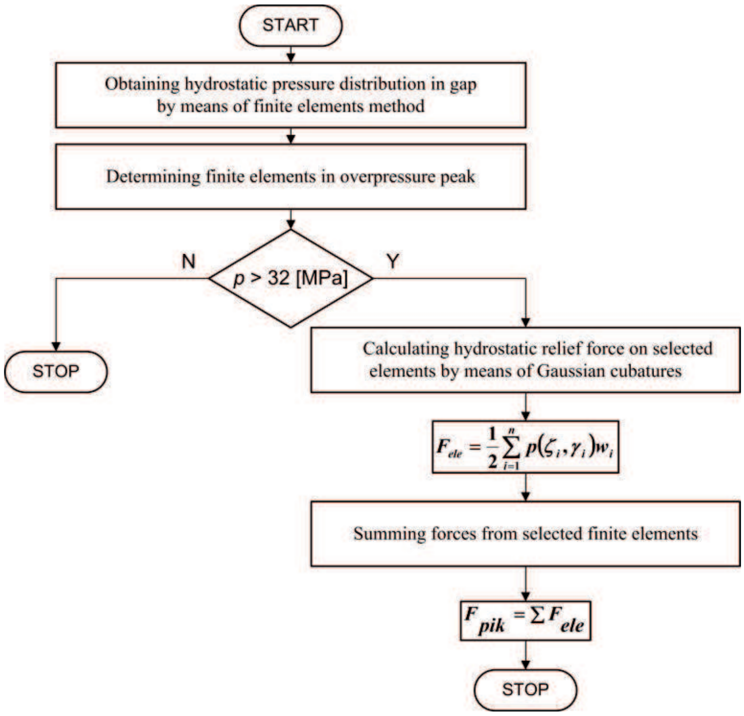

Fig. 5. Algorithm for calculating the overpressure-peak induced hydrostatic relief force occurring on the valve plate of the multi-piston axial pump

- outside and inside the valve plate the pressure p_o = 0 MPa,
- the angular velocity of the cylinder block ω = 157 rad/s
- the dynamic viscosity of the oil μ = 0.0253 Pas,
- the angle of the smallest height of the gap with respect to the axis x δ = 0.785 rad,
- the inclination angle of the cylinder block with respect of the valve plate ε = 0.000523 rad,
- the minimal gap height $h_1 = 0.3 \times 10^{-6}$m
- the characteristic radiuses of the valve plate are r_1 = 0.0284 m, r_2 = 0.0304 m, r_3 = 0.0356 m, and r_4 = 0.0376 m.

Fig. 6 presents the variation in the overpressure-peak induced hydrostatic relief force depending on the variable angular velocity ω of the cylinder block. It follows from the analysis that an increase in the angular velocity causes an increase in the value of the hydrostatic relief force, with the increasing overpressure peak and with the constant smallest gap height. In practice, however, an increase in the hydrostatic relief force is accompanied by an increase in the gap height, and consequently, decay of the overpressure peak.

Fig. 7 presents a change in the overpressure-peak induced hydrostatic relief force depending on the variable dynamic viscosity coefficient η of oil. As can be noted,

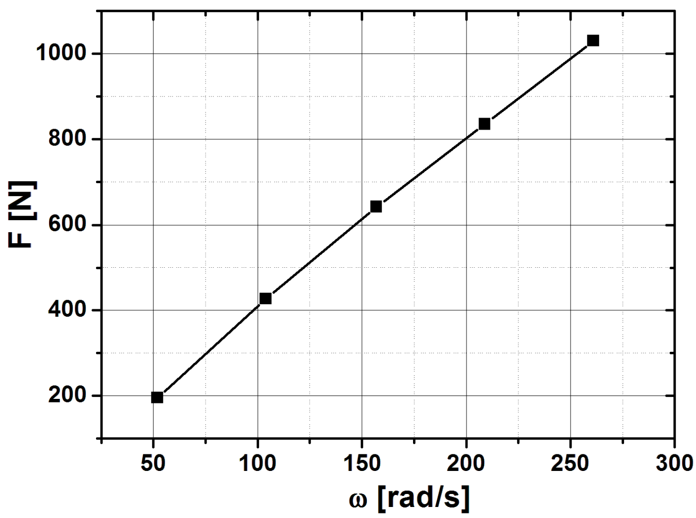

Fig. 6. Overpressure-peak induced hydrostatic relief force as a function of the angular velocity ω of the cylinder block

an increase in the dynamic viscosity coefficient causes an increase in the overpressure-peak induced hydrostatic relief force with the constant smallest gap height.

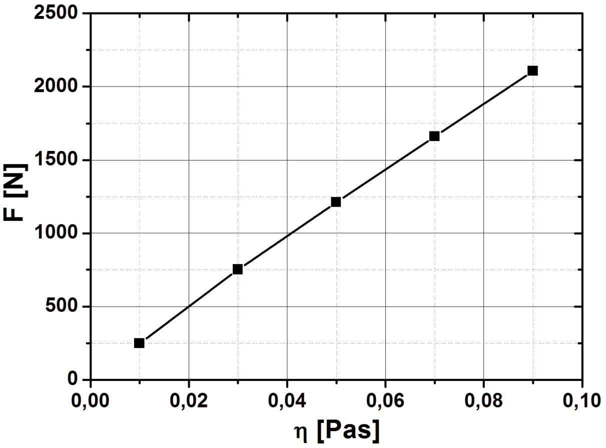

Fig. 7. Overpressure-peak induced hydrostatic relief force of the cylinder block as a function of the dynamic viscosity coefficient η of oil

The simulation experiments show that for the increase in the angular velocity of the cylinder block from 52 to 261 rad/s, the percentage rate of the overpressure-peak induced hydrostatic relief force to the hydrostatic relief force of the cylinder block ranges from 1.25 to 6.49 %. The increase in the oil viscosity from 0.01 to 0.05 Pas causes a relative increase in the overpressure-peak induced hydrostatic relief force with respect to the relief of the cylinder block ranging from 1.59 to 7.57 % with the constant minimal gap height of $h_1 = 0.3 \times 10^{-6}$ m.

CONCLUSIONS

The study leads to the following conclusions:

1. The computational model developed makes it possible to determine the hydrostatic relief force of the cylinder block.
2. Increases in the angular velocity of the cylinder block and in oil viscosity cause increase in the overpressure peak and at the same time increase in the hydrostatic force relieving the cylinder block with the constant minimal height of the gap.
3. In the actual exploitation of hydraulic machines, increase in the overpressure-peak induced hydrostatic relief force causes increase in the minimal gap height and at the same time decreases the overpressure peak, which again decreases the hydrostatic relief force.

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WPŁYW PARAMETRÓW EKSPLOATACYJNYCH NA ODCIĄŻENIE HYDROSTATYCZNE BLOKU CYLINDROWEGO POMPY WIELOTŁOCZKOWEJ

Streszczenie. W pracy przedstawiono wyniki analizy numerycznej siły zmiennego odciążenia hydrostatycznego bloku cylindrowego pompy wielotłoczkowej osiowej wynikającej z powstającego piku nadciśnienia. Siłę odciążenia hydrostatycznego bloku cylindrowego wyznaczano z rozkładu ciśnienia na tarczy rozdzielacza w szczelinie o zmiennej wysokości wykorzystując metodę numeryczną kubatur Gaussa. Analizę siły hydrostatycznej odciążającej pochodzącej od piku nadciśnienia przeprowadzono w zależności od prędkości kątowej bloku cylindrowego i współczynnika lepkości dynamicznej oleju.

Słowa kluczowe: siła odciążenia hydrostatycznego bloku cylindrowego, szczelina o zmiennej wysokości, pompa tłokowa.