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THE IMPACT OF SELECTED PARAMETERS ON PRESSURE DISTRIBUTION IN THE FACE THROTTLE OF A CENTRIFUGAL PUMP AUTOMATIC BALANCING DEVICE

WPŁYW WYBRANYCH PARAMETRÓW NA ROZKŁAD CIŚNIENIA W SZCZELINIE POPRZECZNEJ TARCZY ODCIĄŻAJĄCEJ POMPY ODŚRODKOWEJ

Key words: non-contact seal, pressure distribution, hydraulic local losses, automatic balancing device, random parameters.

Abstract: Face throttles are a necessary functional element of non-contact face seals and automatic balancing devices of centrifugal pumps of different constructions. To calculate the hydrodynamic forces and moments acting on the rotor and fluid flow through the automatic balancing device, it is necessary to know the pressure distribution in the cylindrical and face throttle when considering all important factors which predetermine fluid flow. The face throttle surfaces are moving, which leads to unsteady fluid flow. The movement of the walls of the face throttle causes an additional circumferential and radial flow, which subsequently leads to the additional hydrodynamic pressure components. The paper analyses viscous incompressible fluid flow in the face throttle of an automatic balancing device taking into account the axial and angular displacements of throttle's surfaces and the inertia component of the fluid. The effect of local hydraulic losses as well as random changes in the coefficients of local hydraulic resistance at the inlet and outlet of the throttle is analysed.

Słowa kluczowe: bezkontaktowe uszczelnienie, rozkład ciśnienia, pompa wirowa, losowe zmiany.

Streszczenie: Szczeliny poprzeczne są niezbędnym elementem funkcjonalnym bezstykowych uszczelnień czołowych oraz tarcz odciążających pomp odśrodkowych o różnej konstrukcji. Konieczna jest znajomość rozkładu ciśnienia w szczelinie poprzecznej z uwzględnieniem wszystkich najważniejszych czynników determinujących przepływ cieczy w celu obliczenia sił hydrodynamicznych oraz natężenia przepływu. Powierzchnie szczeliny są ruchome, co powoduje niestacjonarny przepływ płynu. Ruch ścianek szczeliny powoduje dodatkowy przepływ obwodowy oraz promieniowy, a to w konsekwencji prowadzi do powstania dodatkowego ciśnienia hydrodynamicznego. W pracy wykonano analizę przepływu lepkiego nieściśliwego płynu przez szczelinę poprzeczną tarczy odciążającej pompy odśrodkowej z uwzględnieniem przemieszczeń osiowych i kątowych powierzchni szczeliny. Uwzględniono składowe bezwładności płynu. Przeanalizowano wpływ strat lokalnych oraz losowych zmian współczynników hydraulicznych strat miejscowych na wlocie i wylocie szczeliny poprzecznej na rozkład ciśnienia.

INTRODUCTION

The usage and the quantity of hydraulic machinery are constantly increased along with the demands on its operational parameters. Face throttles are commonly used elements in such machinery, namely, in centrifugal pumps. They are the main functional element of the non-contact mechanical seals, thrust bearings, automatic balancing devices (**Fig. 1**), and seals with floating rings.

The face throttle is formed by two surfaces, which are separated from each other by a thin layer of the pumping medium (**Fig. 1**). The film of liquid is the main part of the sealing system and

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has a significant effect on its performance. The pressure of the liquid film separating the throttle surfaces arises due to hydrostatic (pressure drop across the face throttle) and hydrodynamic (influence of the throttle surfaces mobility) effects. In turn, the pressure forces acting on the surfaces from the liquid side affect the vibration level of the rotor system and the efficiency of the pump [**L. 1,2,16**]. For instance, the total losses of energy in an automatic balancing device reach up to 10% of the pump power **[L. 1, 7]** in some cases. Simplified models are commonly used for engineering calculations of such throttles; however, deformations of surfaces are not taken into account, and inertial forces and the stochastic nature of some geometric parameters are neglected. However, with an increasing demand on the operating parameters of centrifugal pumps, the motion of the fluid flow in the face and cylindrical throttles become more complicated, and the introduced simplifications can generate the significant errors.

Calculations of hydrodynamic characteristics of the face throttle are based on the Navier-Stokes equations under some assumptions. Only in this way can the pressure distribution and flow rate characteristics be obtained analytically. Pressure drop and the geometry of the throttle are the main parameters in those equations. As shown in [**L. 16**], most of the values characterizing the geometry of the running part of the pump are random variables due to tolerances and possible surface wear. In fact, the pressure drop in a face throttle changes under pump operation, and its value depends on many random factors. However, many researchers consider this value as a known constant.

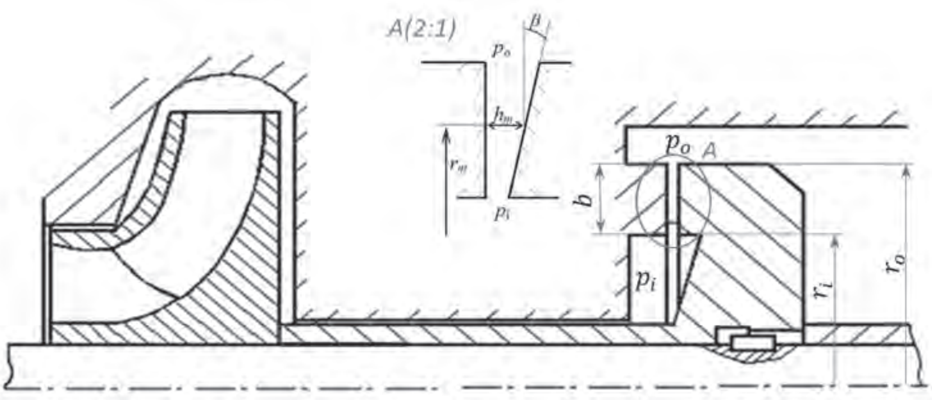

Fig. 1. Simplified model of automatic balancing device
Rys. 1. Uproszczony model tarczy odciążającej

LITERATURE REVIEW

The static calculation of the automatic balancing device is reduced to determining the dependence of the axial unloading force on the face clearance value [**L. 1–3**]. The basic requirement concerning the reliable operation of a face throttle is to maintain the value of the radial clearance within the limits determined at the design stage. This is difficult to ensure because of different disturbances affecting the face throttle operation. To determine static characteristics of the automatic balancing device, Navier-Stokes equations describing the hydrodynamic processes in the cylindrical and face throttles of the device are used, taking into account a number of assumptions typical to these flows [**L. 3–6**]. While studying the flow in the face throttle, a flat shape of the face gap is assumed, which determines the linear pressure distribution along its length [**L. 3**]. The actual profiles of cylindrical and face throttles are quite complex. The non-flatness of cylindrical and face throttles is a result of the axis eccentricity and misalignment of the shaft, as well as possible force deformations [**L. 1, 2, 11**]. In practice, the profile of the face throttle is in form of diffuser, which is due to the deformations of the balancing device disk. This leads to a variation in the pressure distribution (concave pressure diagram) and in a decrease of the resulting unloading axial force. The pressure distribution with considering an effect of the motion of unparalleled surfaces, their waviness and different roughness, is presented in [**L. 7, 8, 9**].

Usually, the isothermal and quasi-stationary flow is considered under pressure distribution calculations in cylindrical and face throttles of an automatic balancing device [**L. 3, 4, 8**]. Calculations of resulting hydrodynamic characteristics should be carried out for different flow regimes in the throttles also. The laminar flow model is applicable for relatively small gaps, while the effect of inertia forces is not taken into account [**L. 1, 3**]. However, some non-contact seals operate under high sliding speeds and pressure drops [**L. 1, 2, 10**]. Namely, the flow in the face throttle of automatic balancing device of a multistage centrifugal pump can be turbulent. In this case, the given above assumptions can significantly change both the static characteristics of the device and affect the dynamics of the pump as well. Analysis and the influence of some components of inertia forces on the hydrodynamic characteristics of the face throttle of an automatic balancing device is given in [**L. 1, 10, 12**]. Pressure distribution in a face throttle depends on inlet and outlet hydraulic losses, the effect of which is considered in works [**L. 1, 2, 9, 13**]. Generally, a rectangular shape of the inlet and outlet is calculated, and the values of the corresponding coefficients are determined empirically. The lack of requirements to a profile of an inlet cross-section of throttles leads to different values of resistance coefficients in practice. It means, that an incorrect determination of the calculated values of forces and flow rates, which are the main static characteristics of the considered device, can take place. A calculation model of the automatic balancing device includes the hydrodynamic models of the face and cylindrical throttle; they are largely determined by the geometry of the throttles. The eccentricity, deformations, local hydraulic resistances, roughness of the surfaces and the clearance of the face throttle are random values under pump operation [**L.15, 16**]. This determines the pressure distribution as a function of some random values and, consequently, the axial force and flow rate. Therefore, the calculation model of the face throttle should include the possible random variations of the above factors.

CALCULATION OF PRESSURE DISTRIBUTION IN FACE THROTTLE

The geometry of the throttle includes the small taper β along the radius and possible axial and angular movements of its surfaces. The calculated model of the face throttle is presented on **Fig. 2**.

Flow in the channel is primarily caused by pressure drop ($p_o - p_i$) along the face throttle length. Simultaneously, local hydrodynamic losses are considered in the boundary conditions for pressure at the inlet and outlet (Δp_1, Δp_{1o}). The flow in the throttle is also influenced by possible inertia forces caused by velocity changing. Boundary conditions for velocities include the rotation of one of the surfaces, which is fixed on the rotor ($\upsilon_\varphi = \omega r$) and the possible axial ($\dot{z}$) and angular (γ) movements.

Dynamic viscosity μ and density ρ of liquid are assumed to be constant. Due to small value of middle gap (20–80 μm) of the face throttle in comparison with its length, the flow velocity component changes in the axial direction and its derivatives are not considered under calculation.

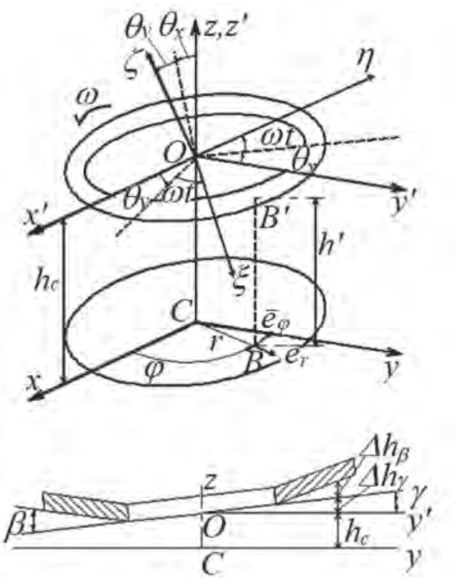

Fig. 2. Calculation model of the face throttle
Rys. 2. Obliczeniowy model szczeliny poprzecznej

The averaged equation of a viscous incompressible liquid motion (averaged Navier-Stokes equation)

$$\frac{dp}{dr} - g_r = -\mu \frac{C Re_r^{1-n}}{8h^2} \overline{\upsilon}_r \tag{1}$$

and the continuity equation are considered to determine the pressure distribution in face throttle:

$$\frac{1}{r}\frac{\partial}{\partial r}(v_r r) + \frac{1}{r}\frac{\partial}{\partial \varphi}(v_\varphi) + \frac{\partial}{\partial z}(v_z) = 0 \tag{2}$$

where (**Fig. 2**)
$p = p(r, z, \varphi, t)$ – pressure in a face throttle gap;

$$g_r = \frac{\rho}{h}\int_{h_1}^{h_2}\left(\frac{\partial v_r}{\partial t} + v_r\frac{\partial v_r}{\partial r} + \frac{v_\varphi}{r}\frac{\partial v_r}{\partial \varphi} + v_z\frac{\partial v_r}{\partial z} - \frac{v_\varphi^2}{r}\right)dz$$

– averaged components of inertia forces;
Re_r – Reynolds number;
$H = h_c + z + r(\theta_x \sin\varphi - \theta_y \cos_\varphi) + (r - r_i)\beta.$

The values of the constants C and n, depending on the flow regime, are given in [**L. 5**].

Velocity distribution along the height of the face throttle gap was taken in the form of a power dependence, where the power coefficient is determined by the flow mode [**L. 5, 6**]:

$$v_r = v_{r\,max}\left[4\frac{z}{h}\left(1-\frac{z}{h}\right)\right]^m. \tag{3}$$

Angular component:

$$v_\varphi = \begin{cases} \kappa v_\varphi\left(2\frac{z}{h}\right)^m, \text{ at } 0 \le z \le \frac{h}{2}, \\ v_\varphi\left[1-(1-\kappa)\left[2\left(1-\frac{z}{h}\right)\right]^m\right], \text{at } \frac{h}{2} < z \le h, \end{cases} \tag{4}$$

where

$$\kappa = \frac{\left(V_\varphi\right)_{z=h/2}}{\omega r} \le 0.5.$$

For the turbulent flow mode: $m = 0$ and $\overline{v_r^2} = \overline{v}_r^2$, $\overline{V_\theta^2} = 0.3\left(V_{\varphi 1} - V_{\varphi 2}\right)^2 - V_{\varphi 1}V_{\varphi 2}$.

After averaging (2) by the height of the throttle

$$\frac{\partial}{\partial r}\left(\overline{v}_r hr\right) + ru_1 + r^2 u_2 = 0, \tag{5}$$

where

$$u_1 = \dot{z},\ u_2 = \left(\dot{\theta}_x + \frac{\omega}{2}\theta_y\right)\sin\varphi - \left(\dot{\theta}_y - \frac{\omega}{2}\theta_x\right)\cos\varphi.$$

Integration (5) by radius under boundary conditions for velocities while considering of one surface's rotation and possible angular and axial oscillations give us the following:

$$q = \overline{v}_r hr = q_* + q_z\left(r, u_1\right) + q_\gamma\left(r, u_2\right), \tag{6}$$

where q_*= const – constant component of the flow rate,
q_z (r, u_1) – flow rate component caused by axial oscillation of throttle's surfaces,
q_γ (r, u_2) – flow rate component caused by angular oscillation of throttle's surfaces.

After the substitution of (6) into (1), the pressure distribution in the face throttle can be obtained as a sum:

$$p = p_s + p_{cz} + p_{c\gamma} + p_g, \tag{7}$$

where
$p_s = p_s(\Delta p)$ – hydrostatic component of the pressure;
$p_{cz} = p_{cz}(z), p_{c\gamma} = p_{c\gamma}\left(\omega, \dot{\theta}_x, \dot{\theta}_y\right)$ – hydrodynamic components of pressure caused by axial and angular oscillations of the movable surface accordingly;
$p_g = p_g\left(\dot{z}, \omega, \dot{\theta}_x, \dot{\theta}_y, \ddot{z}, \omega^2, \ddot{\theta}_x, \ddot{\theta}_y\right)$ – pressure component caused by flow inertia.

The hydrostatic pressure component determines the hydrostatic force components, which, in turn, have an effect on the balancing axial force's value:

$$p_s = \frac{p_o + p_i}{2} + \frac{p_o - p_i}{2}\left[y + \frac{\Lambda + 3\theta_m}{2}\left(1 - y^2\right)\right] + \\ + \frac{p_o - p_i}{2}\left[\frac{\Lambda^2 + 3\theta_m\Lambda + 6\theta_m^2}{3}\left(y^3 - y\right)\right], \tag{8}$$

where
$p_i = p_{i0} - \Delta p_{1i}$ for $r = r_i$,
$p_o = p_{o0} - \Delta p_{1o}$, for $r = r_o$,
$y = \frac{r - r_m}{0{,}5b}$, $b = r_0 - r_i$, $r_m = \frac{r_0 + r_i}{2}$, $r = r_m\left(1 + \Lambda y\right)$, $\Lambda = \frac{b}{0.5r_m}$,
$\theta_m = \frac{\left(\theta_x sin\varphi - \theta_y cos\varphi + \beta\right)b}{2h_m}$, $h_m = h_c + \frac{b\beta}{2}$,
$\Delta p_{1i} = \zeta_{1i}\frac{\rho\overline{v}_i^2}{2}$, $\Delta p_{1o} = \zeta_{1o}\frac{\rho\overline{v}_o^2}{2}$ – local hydraulic losses at the inlet and outlet of the throttle, ζ_{1oi} and ζ_{1i} – coefficients of local resistances for r_o and r_i. According to [**L. 5, 13**]: $\zeta_{1i} = 1.5$, $\zeta_{1oi} = 0.2\ \zeta_{1i}$.

The pressure component caused by axial oscillations of throttle surface ($\dot{z}$) is as follows:

$$p_{cz} = -k_r\mu\frac{b^2}{8h_m^3}\left(1 - \frac{\Lambda + 6\theta_m}{3}y\right)\left(1 - y^2\right)\dot{z}; \tag{9}$$

and pressure component, caused by angular oscillations ($\dot{\theta}_x$, $\dot{\theta}_y$):

$$p_{c\gamma} = -k_r\mu\frac{b^2 r_m}{8h_m^3}\left(1 - 2\theta_m y\right)\left(1 - y^2\right) \\ \left[\left(\dot{\theta}_x + \frac{\omega}{2}\theta_y\right)\sin\phi - \left(\dot{\theta}_y - \frac{\omega}{2}\theta_x\right)\cos\phi\right]. \tag{10}$$

Pressure component caused by flow inertia ($\dot{q}$):

$$p_{g1} = -\rho \frac{p_o - p_i}{k_r \mu} \frac{3h_m}{2} \left[\dot{z} + r_m \left(\dot{\theta}_x \sin\phi - \dot{\theta}_y \cos\phi \right) \right] \theta_m \left(1 - \frac{2\Lambda + 5\theta_m}{3} y \right) \left(1 - y^2 \right) - \rho \frac{b^2}{8h_m} \ddot{z} \cdot$$
$$\cdot \left(1 - \frac{\Lambda + 2\theta_m}{3} y \right) \left(1 - y^2 \right) - \rho \frac{b^2}{8h_m} r_m \left[\left(\ddot{\theta}_x + \frac{\omega}{2} \dot{\theta}_y \right) \sin\phi - \left(\ddot{\theta}_y - \frac{\omega}{2} \dot{\theta}_x \right) \cos\phi \right] \left(1 - \frac{2\theta_m}{3} y \right) \left(1 - y^2 \right), \tag{11}$$

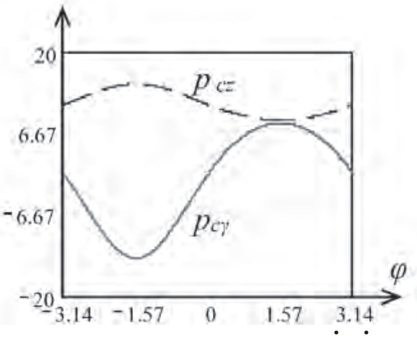

Fig. 3. Distribution of hydrodynamic pressure components caused by axial ($\dot{z}$) and angular ($\dot{\theta}_x$, $\dot{\theta}_y$) oscillations

Rys. 3. Rozkład ciśnienia hydrodynamicznego wywołanego osiowymi ($\dot{z}$) oraz kątowymi ($\dot{\theta}_x$, $\dot{\theta}_y$) drganiami

The hydrodynamic pressure component expressed by equations (9) and (10) are asymmetric with respect to the axis of rotation. It generates the axial force that damps the oscillations. The inclination of the face throttle surfaces ($\gamma_m^2 = \theta_x^2 + \theta_y^2$) causes an unequal distribution of hydrodynamic pressure around the circumference; therefore, a moment occurs, the direction of which depends on the taper of the throttle.

Total pressure losses in face throttle (**Fig. 4a**):

$$\Delta p_0 = p_{o0} - p_{i0} = \zeta_{1o} \frac{\rho \bar{V}_o^2}{2} + \zeta_{2m} \frac{\rho \bar{V}_m^2}{2} - \zeta_{1i} \frac{\rho \bar{V}_i^2}{2} = \zeta_t \frac{\rho \bar{V}_m^2}{2} \tag{12}$$

where $\zeta_{2m} = \frac{\lambda_0 b}{2h_m} = \left[\frac{\mu}{2\rho |\bar{v}_m| h_m} \right]^n \frac{Cb}{2h_m}$ – coefficient of hydraulic losses along the length of the throttle, ζ_t – coefficient of total hydraulic losses in face throttle.

Local hydraulic losses at the inlet and outlet are functions of the averaged values of inlet/outlet velocity, accordingly. The total radial flow rate should be known to determinate these velocities' values. A constant component of the total flow-rate q_* can be obtained when integrating (1) under boundary conditions for pressure, considering local hydraulic losses. Therefore, on the basis of (1) and (5), total radial flow-rate includes the following: the component of flow rate (q_s) caused by pressure drop, the component of flow rate (q_g) caused by inertia forces and two components (q_{cz}) and (q_{cy}) caused by axial and angular oscillations accordingly:

$$q = q_* \left[1 - (1 + \Lambda y)^2 \bar{u}_{cz} - (1 + \Lambda y)^3 \bar{u}_{c\gamma} \right] = q_s + q_{cz} + q_{c\gamma} + q_g, \tag{13}$$

where $q_* = \bar{v}_{m*} h_m r_m$ – component of the flow rate, which is independent on the radius; $\bar{v}_{m*}$ – constant component of the radial velocity at the middle radius r_m; $\bar{u}_{cz} = \frac{r_m}{2h_{m0}\bar{v}_{m*}} u_1$; $\bar{u}_{c\gamma} = \frac{r_m^2}{3h_{m0}\bar{v}_{m*}} u_2$.

The hydrostatic component of pressure while considering local losses (**Fig. 4b**) is as follows:

$$p_s = \frac{p_{0i} + p_{0o}}{2} - \frac{\Delta p}{2} \frac{\zeta_i' + \zeta_o'}{\zeta_t'} + \frac{\Delta p}{2} \frac{\zeta_{2m}}{\zeta_t'} \left[y + \frac{(2-n)\Lambda + 3\bar{\beta}^2}{2} (1 - y^2) \right] -$$
$$- \frac{\Delta p}{2} \left[\frac{(2-n)(3-n)\Lambda^2 + 6(2-n)\bar{\beta}\Lambda + 12\bar{\beta}^2}{6} y (1 - y^2) \right], \tag{14}$$

where

$$\zeta_i' = \frac{\zeta_{1i}}{(1+\Lambda^2)(1+\bar{\beta}^2)}, \zeta_o' = \frac{\zeta_{1o}}{(1+\Lambda^2)(1+\bar{\beta}^2)}.$$

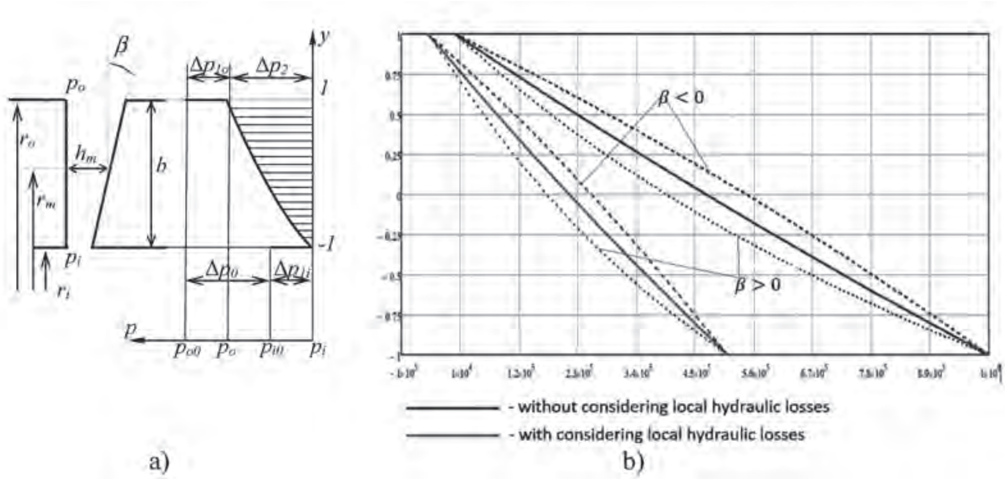

Fig. 4. Pressure distribution (hydrostatic component) in face throttle considering the local hydraulic losses (Δ_p = 1 MPa)

Rys. 4. Rozkład ciśnienia (hydrostatyczna składowa) w szczelinie poprzecznej z uwzględnieniem hydraulicznych strat miejscowych (Δ_p = 1 MPa)

Hydrostatic pressure component (14) determines the force and moment values caused by main pressure drop along the face throttle. The taper of the throttle (β) affects the distribution of pressure along the radius, namely, pressure increases in the convergent throttle (**Fig. 4**) and decreases under the diffuser shape of the throttle section. Local losses at inlet decrease the pressure value up to 50%, and it results in decreasing the axial force. Inclination of the face throttle surfaces ($\gamma^2 = \theta_x^2 + \theta_y^2$) causes an unequal distribution of hydrostatic pressure around the circumference, and, therefore, a moment occurs, the direction of which depends on the taper of the throttle.

INFLUENCE OF RANDOM CHANGES IN LOCAL RESISTANT COEFFICIENTS ON PRESSURE DISTRIBUTION IN A FACE THROTTLE

According to (14), the hydrostatic pressure component is function of geometrical parameters and pressure drop:

$$p_s = f_1(\Delta p, \zeta_{1i}, \zeta_{10}, b, h_m)$$

Pressure drop Δp and length b are assumed to be deterministic values as well as the throttle's height h_m under this research.

Values of the local resistant coefficients ζ_{1o} and ζ_{1i} should be determined empirically, but these values can change under pump operation and even under installation. In major researches, its values are considered constant. Therefore, one of the aims of this research is to calculate the influence of the random changes of local resistant coefficients values on pressure distribution in face throttle. Normal distributions for ζ_{1o} and ζ_{1i} were taken for modelling the random changes of these coefficients with parameters: $m_{\zeta_{1o}}$, $\sigma_{\zeta_{1o}}$ and $m_{\zeta_{1i}}$, $\sigma_{\zeta_{1i}}$ correspondingly. The empirical values are taken ($m_{\zeta_{1o}} = 1.5, m_{\zeta_{1i}} = 0.3$) as the mean values of the hydraulic losses' coefficients at the inlet and outlet. Under such assumptions, the mean values of pressure along the radius can be expressed as follows:

$$m_{p_s} = \int_{\zeta_{1omin}}^{\zeta_{1omax}} \int_{\zeta_{iomin}}^{\zeta_{1omax}} f_1(y, \Delta p, \zeta_{1i}, \zeta_{10}) f_2(\zeta_{1i}) f_3(\zeta_{10}) d\zeta_{1i} d\zeta_{10} \tag{15}$$

and the dispersion as follows:

$$D_{p_s} = \int_{\zeta_{1omin}}^{\zeta_{1omax}} \int_{\zeta_{iomin}}^{\zeta_{1omax}} \left(f_1(y, \Delta p, \zeta_{1i}, \zeta_{10}) - m_{p_s}\right)^2 f_2(\zeta_{1i}) f_3(\zeta_{10}) d\zeta_{1i} d\zeta_{10} \tag{16}$$

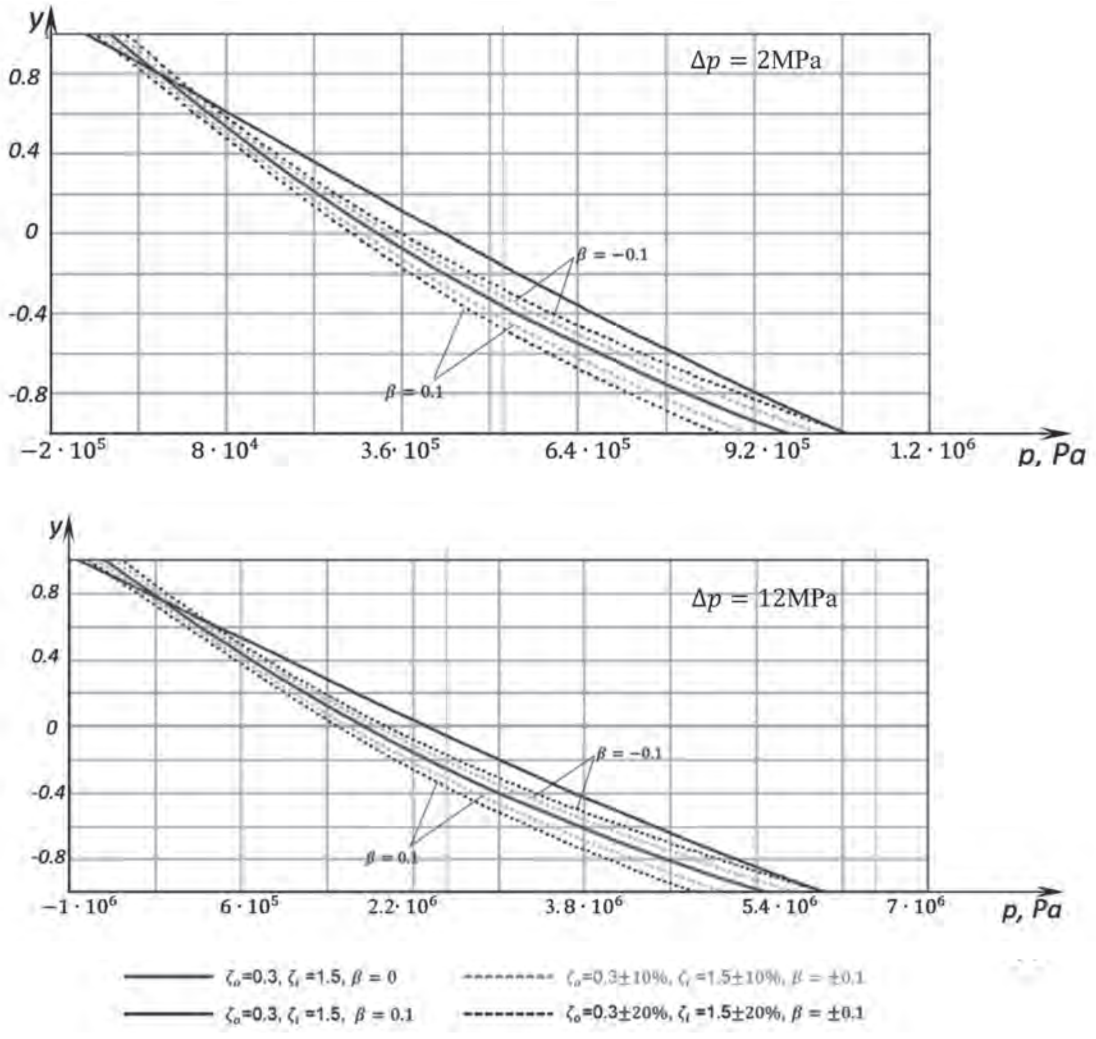

Fig. 5. Pressure distribution in face throttle along the radius with random changes of local hydraulic losses coefficients

Rys. 5. Rozkład ciśnienia wzdłuż szczeliny poprzecznej z uwzględnieniem losowej zmianywspółczynników hydraulicznych strat miejscowych

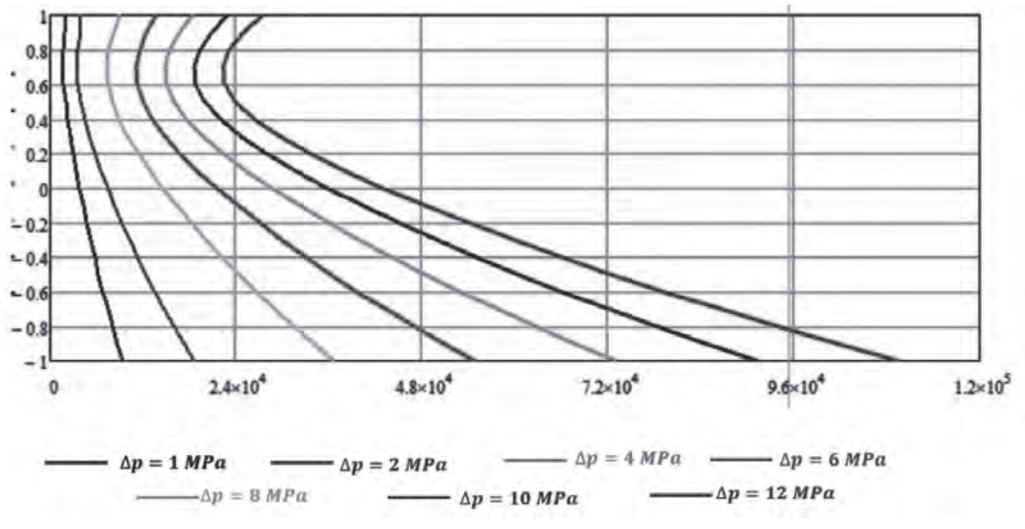

Fig. 6. Pressure standard deviation distribution along the face throttle's length under different pressure drop

Rys. 6. Zmiana wartości standardowego odchylenia ciśnienia wzdłuż szczeliny poprzecznej dla różnych wartości spadku ciśnienia

The pressure distribution is presented in **Fig. 5** under different values of pressure drop and standard deviations of local resistant coefficients ζ_{1o} and ζ_{1i}.

According to **Figure 6**, the value of the pressure standard deviation increases as the throttled pressure rises. The maximum effect of random changes in the values of local hydraulic loss coefficients is observed at the face throttle's inlet: Δp = 2MPa Δp = 12MPa.

CONCLUSIONS

The problem of viscous incompressible fluid flow in the face throttle of automatic balancing device is solved. The pressure distribution is the sum of the components: hydrostatic, hydrodynamic and inertia. Hydrodynamic components of pressure results from the surface motion that includes its rotation, axial and angular oscillations. The inertial component of pressure takes into account the unsteady nature of the liquid flow, i.e. the change in radial velocity with time.

Integration of the hydrostatic pressure distribution (14) over the face throttle area produces a fluid film reaction force. The coefficient of axial stiffness is positive, and the axial force is restored under liquid flow in a convergent throttle. On the contrary, the hydrostatic force's value decreases and the axial stiffness coefficient becomes negative when the fluid flows in the diffuser throttle.

A random change in the local loss coefficients variates the pressure value along the face throttle length. Maximum deviation takes place at the inlet (23.8%). On the basis of the proposed model, the maximum deviations of the pressure values can be determined for various regimes of liquid flow in the face throttle's clearance with a designated confidence level.

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NOMENCLATURE

ρ – fluid density [kg/m^3],
μ – dynamic viscosity [Pa · s],
$p = p(r; z, \varphi, t)$ – pressure in a face throttle [Pa],
h – current value of the face throttle clearance [m],
r_i, r_m, r_o – inlet, middle and outlet radius [m],
h_c – average clearance value without taking into account taper and angular misalignment of a face throttle surfaces [m],
h_m – clearance value for $r = r_m$ [m],
$p_o - p_i$ – pressure drop along the length [Pa],

$Re = \dfrac{2\rho v_m h_m}{\mu}$ – Reynolds number in radial direction [-],

$k_r = \dfrac{C}{8} Re^{1-n}$ – modified friction coefficient [-]

Δp_{1i}, Δp_{1o} – hydrodynamic losses of pressure at inlet and outlet of the face throttle [Pa],
v_r, v_φ, v_z – velocity components in cylindrical coordinate system [m/s],
g_r – averaged components of inertia forces [Pa/m],
ω – angular frequency [rad/s],
$\gamma^2 = \theta_x^2 + \theta_y^2$ – complete angular misalignment of a face throttle surfaces [rad^2],
C, n – coefficients depending on the fluid flow regime [-],
b – a face throttle's length [m],
β – taper angle [rad],
q – elementary flow rate [m^2/s],
p_s – hydrostatic pressure component [Pa],
p_{cz}, $p_{c\gamma}$ – hydrodynamic pressure components caused by axial and angular movements of a face throttle surface [Pa],
p_g – pressure component caused by flow inertia [Pa],
ξ_{1i}, ξ_{1o} – coefficients of local fluid resistance at inlet and outlet [-],
ξ_{2m} – resistance coefficients along the length [-]

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