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MODELLING THE INFLUENCE OF CYLINDRICAL THROTTLE EROSION WEAR ON CHARACTERISTICS OF AUTOMATIC BALANCING DEVICE

MODELOWANIE WPŁYWU EROZJI CYLINDRYCZNEJ PRZEPUSTNICY NA WŁAŚCIWOŚCI AUTOMATYCZNEGO URZĄDZENIA WYWAŻAJĄCEGO

Key words: erosive wear, annular seal, automatic balancing device, centrifugal pump, static characteristic.

Abstract Complicated system of forces acts on the rotor of multi-stage centrifugal pump under its exploitation. Axial force is the biggest one for pumps with one-side inlet wheels. This force in many constructions of pumps is balanced by special automatic balancing device (ABD). This device consists of an unloading disk rigidly fixed on the shaft and a non-rotating supporting disk. The corresponding surfaces of these elements form cylindrical and face throttles arranged in series divided with the unloading chamber. Automatic operation of such devices is caused by the interdependence of hydrodynamic characteristics in cylindrical and face throttles. The middle gap value of cylindrical throttle is one of the main characteristics determining the pressure in the chamber and the ABD flow-rate value. The middle gap can increase due to erosive wear under pump operating, and it can result in negative impact on ABD operating. In this work, the intensity of erosion wear is considered as a function of middle velocity of flow and the coefficient of material wear. The middle velocity of flow in the cylindrical throttle is obtained by solving equations of turbulent fluid flow motion jointly with continuity equation. It allows one to determine the changes of the middle gap of cylindrical throttle in time and to estimate the influence of such changes on static and flow-rate characteristics of the ABD.

Słowa kluczowe: erozyjne zużycie, bezkontaktowe uszczelnienie, tarcia odciążająca, pompa wirowa, statyczna charakterystyka.

Streszczenie Podczas pracy wielostopniowej pompy na jej wirnik oddziałuje złożony układ sił. W pompach posiadających wirnik z jednostronnym wlotem największą wartość uzyskuje siła osiowa. W większości spotykanych rozwiązań równoważenie tej siły odbywa się za pomocom specjalnego elementu konstrukcyjnego – tarczy odciążającej. Tarcza obraca się razem z wałem współpracując ze stałymi pierścieniami. Przestrzeń między tymi elementami tworzy układ dławiący. Jeśli wystąpi przemieszczenie wirnika w stronę wlotu, to wzrastające ciśnienie wywoła powstanie siły odsuwającej tarczę. Istotne znaczenie ma rozkład ciśnienia w układzie dławiącym. Występująca zdolność samoregulacji tarczy odciążającej zależy od charakterystyki hydrodynamicznej układu dławiącego. Grubość szczeliny cylindrycznego dławika może powiększać się podczas pracy pompy w wyniku erozyjnego zużycia powierzchni uszczelnienia. W opracowaniu intensywność zużycia uzależniono od średniej prędkości w szczelinie oraz współczynnika zużycia materiału. Wartość średniej prędkości uzyskano w wyniku rozwiązania układu równań ruchu cieczy w szczelinie oraz równania ciągłości. Takie podejście pozwoliło wyznaczyć grubość szczeliny dławika w funkcji czasu i oszacować jej wpływ na charakterystyki tarczy odciążającej.

INTRODUCTION

The development of power, metallurgical, petroleum, and other branches of industry is impossible without the application of new types of pumps with wide range of heads and pressures. For this purpose, sectional centrifugal pumps are most commonly used **[L. 1–3]**. The typical construction of centrifugal pump consists of many different rotating and immobile elements interacting through the pumping medium. That is why development of a pump calculation model is very complicated even while taking into account modern

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numerical abilities. A complicated system of forces acts on the rotor of multi-stage centrifugal pump under its exploitation. There are inertia forces of unbalanced masses, hydrodynamic forces in small throttles of running part, in seals and bearings, and axial forces of pressure on disks of driving wheels. Axial force is the biggest one, and its value can be measured by tens or even by hundreds kilonewtons. This force is caused by the lack of symmetry of the driving wheel in the plane which is perpendicular to the rotor axe **[L. 1]**. The area of outer surface of the main disk is bigger than the opposite surface of cover disk (**Fig. 1**). It leads to the increase of axial pressure force, which is directed to the inlet. This axial force, in many constructions of high-pressure pumps, is balanced by special automatic balancing device (ABD). ABD systems have a quite simple construction. They consist of a cylindrical throttle, a chamber, and a face throttle (**Fig. 1**). The increase of axial force acting on a rotor results in a decrease of throttle face gap value. Thus, conductivity of face throttle (4) (**Fig. 1**) decreases and pressure p_2 in chamber (3) is increases and pressure p_3 decreases too (usually it is a pressure at pump inlet). It leads to the value of the axial force acting on a rotor bringing it back into primary position **[L. 4]**.

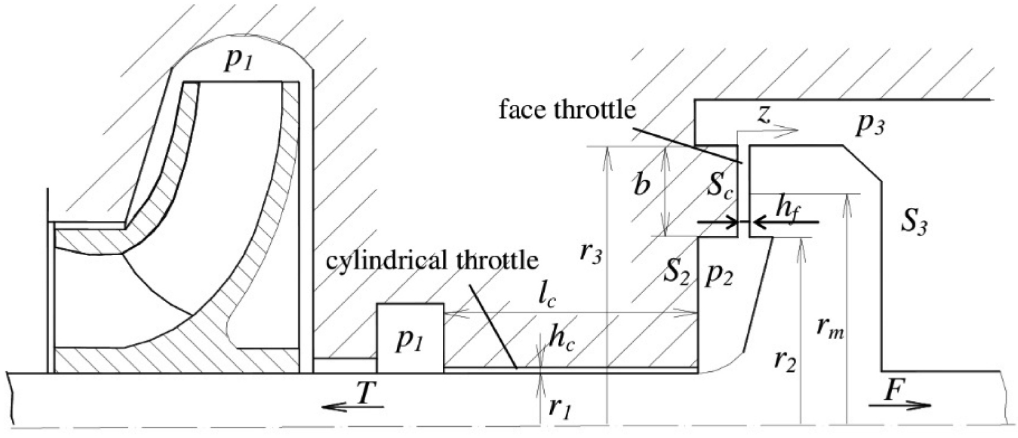

Fig. 1. Calculation model of the ABD
Rys. 1. Schemat obliczeniowy tarczy odciążającej

Some types of centrifugal pumps for steam boilers pump water at a temperature range from 80 to 165 degrees. Structurally, they are horizontal, sectional, multi-stage pumps with the one-sided driving wheels. The running part is made from high-quality cast-iron, corrosion resistant steel, coloured metals, or their alloys. Erosion-corrosion that is defined as a combination of erosive wear and corrosion can lead to wear in a short period of time in pumps made of cast iron. Depending on structural features, such pumps can operate with a liquid in which the number of admixtures does not exceed 0.2%. The size of the mechanical parts is within the limits of 0.05–0.1 mm. Therefore, mechanical erosion wear of sealing surfaces in such flows can also occur. It results in the middle gap value changes in time, and it influences on flow-rate characteristics of cylindrical and face throttles as well as static and dynamic characteristics of the ABD.

LITERATURE REVIEW

Static calculation of the ABD systems is based on getting the dependence of external forces in general on axial force T on face throttle gap value **[L. 4–6]**. The value of the axial force T can deviate from the calculated one significantly during pump operation. Most commonly used schemes of the ABD systems are presented on **Fig. 1**. All researches of the ABD systems try to find a calculation method that could allow one to choose the basic geometric parameters of the ABD system with the guarantee that values of flow-rate through the ABD system as well as the face throttle gap values are in required areas of axial force deviations. The design of the ABD cannot be made without modelling of flow in cylindrical and face throttles, because the flow in these gaps determines the operation of whole the ABD **[L. 6–9]**.

In most earlier research works, hydrodynamic characteristics of face throttles were calculated on the basis of a simplified flat model of a face gap **[L. 4, 8]** with parallel surfaces. But values of hydrodynamic force in this case were significantly smaller than the experimental ones. In works **[L. 7, 9, 10]**, hydrodynamic forces and moments were determined while taking into account the movement of unparalleled surfaces, the waviness of surfaces, and roughness. An unparalleled form of face throttle walls can be caused by the presence of a defect or the non-planarity of surfaces (cone shape, waviness, roughness). The cone shape can be caused by thermal and force deformations and mechanical wear. The waviness of surfaces is due to the deformation of rings under the action of temperature and force deviations or as a result of the thermo-elastic instability of surfaces. A roughness is determined by the material of the machined surfaces.

In many works **[L. 6–9]**, the calculation of the hydrodynamic characteristics of the face throttle is made on the assumption of isothermal and quasi-stationary flow in the gap. The laminar flow model is taken due to the small value of the gap, and the inertia forces are neglected as a rule **[L. 7]**.

Black and Jenssen **[L. 10, 11]** presented solutions for the short annular seal hydrodynamic coefficients and estimated correction factors for the long annular seals. In **[L. 12]**, Childs presented an analytical expression for the hydrodynamic force generated in long annular seals. Baskharone et al. **[L. 13]** developed a model for leakage flow based on the bulk-flow model and compared numerical and experimental results for annular seal flows. Kanemori et al. **[L. 14]** tested the influence of misalignment on hydrodynamic coefficients for long annular seals. Brennen **[L. 15]** found that inlet whirl flows reduced the stability of rotors. Suzuki **[L. 16]** examined fluid-induced rotor dynamic forces applied on artificial feed pump shafts and found that the rotor dynamic forces become destabilized in a wide range of positive whirls. Moreover, in **[L. 1]**, the analytical equations of the flow rate and hydrodynamic forces and moments in a shot annular seal were obtained on the bases of the Navie -Stoke's equations solution while taking into account the conical shape of the gap and the misalignment of rotor and sleeve axes. Later, in the work **[L. 17]**, the influence of inertia forces and random deviations of the geometrical parameters of the annular seal on flow rate and hydrodynamic characteristics were examined.

The analytical model of the ABD should be developed on the basis of models presented above of the cylindrical throttle (annular seal) and the face throttle (face non-contact seal).

During pump operation, the annular seals gaps at the impellers and the ABD gradually increase due to sealing surfaces wear. Such significant wear is caused by occasional contact, erosion, corrosion due to high values of flow velocity, and abrasion due to solid in the pumping medium. Therefore, a calculation model of ABD should include the influence of possible changes of cylindrical throttle value. In this research, a semi-empirical model is developed including experimental determined values of material erosion wear.

METHODOLOGY OF CALCULATION OF EROSIVE WEAR OF CYLINDRICAL THROTTLE'S SEALING SURFACES

Based on experimental data presented in **[L. 18]**, the intensity of erosion wear γ in small throttles of feed pumps can be taken as a function of middle flow velocity w using the following equation:

$$\gamma = k_e w^n \tag{1}$$

where k_e– an erosive wear coefficient for axial flow characterising material and operational conditions; n – a coefficient depending on middle flow velocity. According to **[L. 18]**, the value of n for annular flow can be determined as follows:

$$n = \begin{cases} 2, \ w < 60\dfrac{m}{s} \\ 3, \ w \geq 60\dfrac{m}{s} \end{cases}$$

Intensity of the sealing surfaces wear depends on material characteristics (in general hardness and toughness of the material), liquid characteristics (viscosity and density), flow velocity, which in turn depends on the size and shape of the throttle, the roughness of the surface, and, in the case of hydro-abrasive wear, from solid particle concentration, the size and shape of solid particles, and the solid particle impact angle. In this work, the erosion wear of ABD annular seal is calculated on the basis of the model (1). The coefficient of material wear k_e can be determined from statistical data obtained under exploitation of such throttles or from service life tests of the material wear. In **[L. 18]**, the values of the coefficient of erosive wear in condensate are presented for the three most widely used in pump building materials.

As a reference, corrosion-resistant high-temperature steel 321 (US), 321S51(England), was taken. The reference coefficient of erosion wear is as follows:

$$k_{er} = \begin{cases} 7.12 \cdot 10^{-6}, w < 60\dfrac{m}{s} \\ 3.26 \cdot 10^{-8}, w \geq 60\dfrac{m}{s} \end{cases}$$

The middle velocity of pumping flow in an annular throttle can be calculated on the basis of the continuity equation and Navier-Stoke's equations solution. The equation of liquid flow motion in the gap of annular throttle under prevailing axial flow can be written as follows:

$$\frac{\partial p}{\partial y} = 0, \frac{\partial p}{\partial z} + \varrho_i = -k\frac{\mu}{h^2}\overline{w} \tag{2}$$

where: p – a pressure in the gap of annular throttle, ϱ_i – an inertia component of the hydrodynamic force, k – a modified coefficient of the friction, μ – a dynamic viscosity of the pumping liquid, h – gap's value, $\overline{w}$ – an averaged through the gap value of axial velocity.

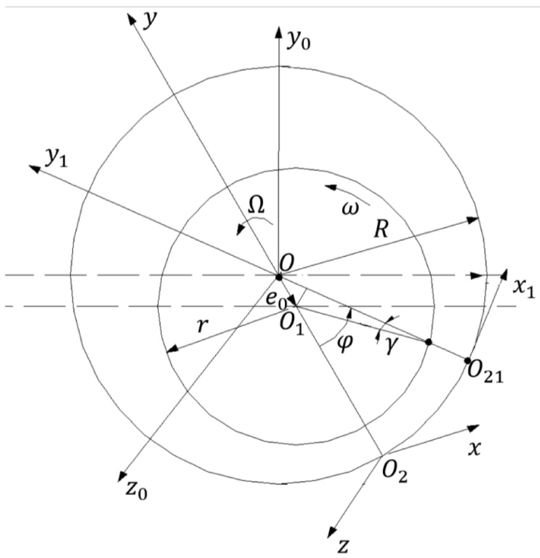

Fig. 2. Calculation model of cylindrical throttle
Rys. 2. Obliczeniowy model pierścieniowej szczeliny

Due to absence of the axial velocities of the sealing surfaces and under the assumption of the prevailing axial flow, the averaged continuity equation can be as follows:

$$\frac{\partial}{\partial z}\left(\overline{w}h\right) = -\frac{\partial}{\partial x}\left(\overline{u}_c h\right) + u\left(h\right)\frac{\partial h}{\partial x} - v\left(h\right) + v(0) \quad (3)$$

where z, x – axial and tangential coordinates, $\overline{u}_c = 0.5\kappa\left(u\left(0\right) + u(h)\right) = 0.5\kappa\omega r$, $u(h)$, $v(h)$, $v(0)$ – velocities of the sealing surfaces (on rotating and stationary wall) (**Fig. 2**). Here, κ is a coefficient depending on the initial rotation of the flow at the throttle inlet: $\kappa = 1$ if there is no rotation of flow at the inlet, $\kappa > 1$ if the flow at the inlet has initial circumferential velocity, and $\kappa < 1$ otherwise.

After integrating (3), $\overline{w}h = q = q_* - 0.5l\left(\Delta\omega\, y sin\varphi + \dot{y} cos\varphi\right)$ where q_* – elementary flow rate in the middle at the throttle length section, $y = -e_0$, e_0 – eccentricity. An elementary flow rate q_* can be defined through the pressure gradient from (2). Therefore, in general, the axial component of the velocity in annular throttle can be presented as follows:

$$w = w_p + w_d + w_g \quad (4)$$

Where w_p – main component caused by pressure drop in throttle with immobile surfaces, w_d – component caused by radial and angular oscillations of surfaces and w_g – component caused by inertia characteristics of the flow. The value of the inertia component of velocity w_g can be significant for some conditions and is verified in **[L. 18]**, but, in this research, it is not considered. Therefore, the value of circumferential velocity is basically determined by the rotation frequency of rotor and the radius of inlet impeller.

$$u = \omega r + u_p \quad (5)$$

Where u_p is caused by spreading of pumping liquid in circumferential direction under radial and angular oscillations of the rotor. As shown by experimental research presented in **[L. 1, 8]**, the circumferential velocity value for turbulent flows can be obtained from the following dependences:

$$\begin{aligned} u &= u\left(0\right) + 0.5\omega r\left(2y_1/h\right)^{1/7}, \quad 0 \le y_1 \le 0.5h \\ u &= u\left(h\right) - 0.5\omega r\left[2\left(1 - y_1/h\right)\right]^{1/7}, \quad 0.5h < y_1 \le h \end{aligned} \quad (6)$$

Here, y_1 is a local coordinate (**Fig. 2**).

Therefore, the resulting velocity in annular throttle can be calculated as follows:

$$V = \sqrt{\left(w_p + w_d\right)^2 + u^2} \quad (7)$$

The velocity of wear can be considered as increasing in time in the middle radial gap of annular throttle $\gamma = \frac{dh}{dt}$.

Velocity of flow in concentric annular throttle with immobile surfaces is $w = w_p = \frac{\Delta pH^2}{12\mu l}$ – for laminar flow and $w_p = 10\left(\frac{\Delta pH}{\rho l}\right)^{\frac{1}{2}}$ for fully developed turbulent flow without taking into account cylindrical throttle sealing surfaces motion.

Therefore, from (1), The following can be obtained:

$$\frac{dh}{dt} = k_e\left(\frac{\Delta pH^2}{12\mu l}\right)^3 \text{ and } \frac{dh}{dt} = k_e\left(10\left(\frac{\Delta pH}{\rho l}\right)^{\frac{1}{2}}\right)^3 \quad (8)$$

Equation (8) was solved by the Runge-Kutta method with 4th order, and the obtained results were approximated by proper dependence. In the case of a lack of sealing surface motion, this dependence is as follows:

$$H = u_0 e^{(u_1 t)}$$

where $u_0 = 2.46 \cdot 10^{-4}$, $u_1 = 5.9 \cdot 10^{-5}$ for reference steel under presser drop 1MPa.

Figure 3 presents the dependences of the middle gap value of cylindrical throttle during the time of pump operation for reference steel under different pressure drop and rotation frequency values ($\overline{w} < 60 m/s$).

Under a 1 MPa pressure drop, the possible value of middle gap of cylindrical throttle is 57% higher than the initial (design) value. The movement of the sealing

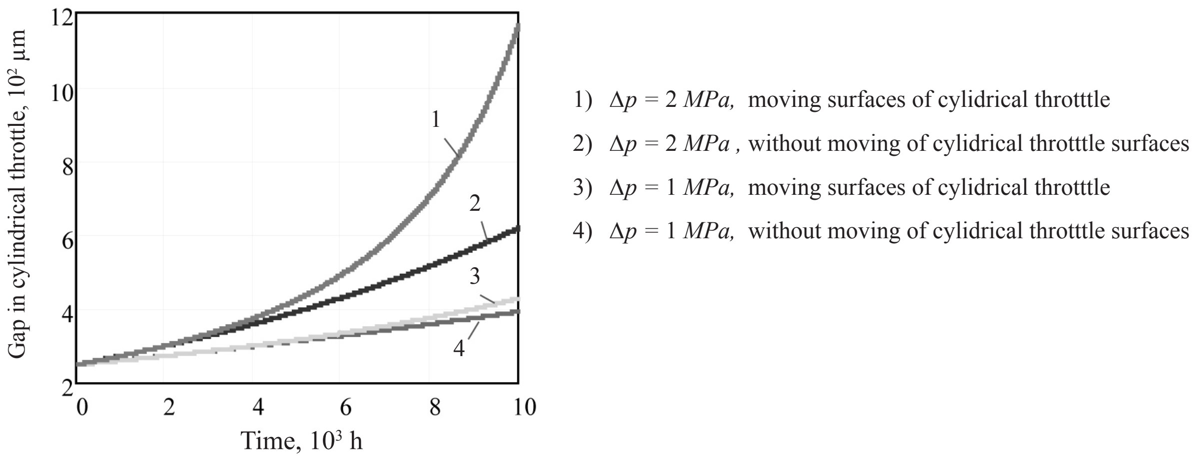

Fig. 3. Dependences of middle gap value of annular throttle on time of pump operation
Rys. 3. Zależność średniej wysokości pierścieniowej szczeliny od czasu pracy pompy

surfaces (generally rotation of the shaft) increases the intensity of wear, and, in this case, the value of the middle gap of cylindrical throttle is up to 71% higher than the initial one. The same tendency can be observed under a 2 MPa pressure drop: 146% and 360%, accordingly. Therefore, the movement of the sealing surfaces and the increasing of the pressure drop on cylindrical throttle lead to the increase in the wear intensity and, consequently, the increase in the mean gap value. The conductivity of the cylindrical throttle g_1 depends on its geometry, in particular, it is inversely proportional to the value of the radial gap h. As shown below, the gap increases over time (**Fig. 3**). Thus, the hydraulic resistance of the gap decreases in time and the conductivity of the cylindrical throttle g_1 also decreases.

THE INFLUENCE OF THE EROSIVE WEAR OF CYLINDRICAL THROTTLE'S SEALING SURFACES ON ABD CHARACTERISTICS

In a static calculation, it is necessary to determine the dependence of the face gap value on the axial force **[L. 5, 9]**. It can be found from the condition of axial equilibrium of the unloading disc (**Fig. 1**):

$$T = F + F_d \tag{9}$$

where $F = S_2 p_{20} + S_2 \Delta p_b \psi_2 + F_m$ – resulting hydrodynamic axial force of pressure, acting on disk of ABD, which consists of the pressure force p_2 and the pressure force in the face gap, $F_d = k_d (\Delta - h_{m0})$, k_d – the stiffness of the springs of the pushing device, Δ – their preliminary compression, h_{m0} – value of the face gap.

The dependence of pressure p_2 in the chamber of ABD on the face gap value can be calculated from the balance of flow-rate of liquid through the cylindrical Q_1 and face Q_2 throttles as follows:

$$Q_1 = Q_3, \tag{10}$$

where

$$Q_1 = g_1 \sqrt{p_1 - p_2} \text{ and } Q_2 = g_2 \sqrt{p_2 - p_3}$$

where g_1, g_2 – conductivities of cylindrical and face throttle respectively. The conductivity of the cylindrical throttle can be calculated as $g_1 = 2\pi r_1 h \sqrt{2/\rho\zeta_c}$, where $\zeta_c = \zeta_{ic} + \zeta_{2c} - \zeta_{oc}$ – coefficient of full losses in cylindrical throttle, ζ_{ic} and ζ_{oc} – coefficients of local losses at inlet and outlet of cylindrical throttle, and $\zeta_{2c} = \dfrac{\lambda l_1}{2h_1}$ – coefficient of losses on the length of cylindrical throttle. Note that the gap value of the cylindrical throttle increases over time (**Fig. 3**).

Similarly, $g_2 = 2\pi r_m h_m \sqrt{2/\rho\zeta_f}$, where $\zeta_f = \zeta_{if} + \zeta_{2f} - \zeta_{of}$ – coefficient of full losses in face throttle, ζ_{if} and ζ_{if} – coefficients of local losses at inlet and outlet of face throttle, and $\zeta_{2f} = \dfrac{\lambda b}{2h_m}$ – coefficient of losses on the length of face throttle. A possible change in the value of the face gap h_m determines the regulation of the magnitude of the axial force acting on the pump rotor T. This dependency obtained by solving Equations 9 and 10 is a static characteristic of the ABD (**Fig. 4**).

The conductivity of the face throttle g_2 depends on the value of the face gap h_f and causes the pressure in the discharge chamber p_2. Pressure in chamber p_2 is obtained from the following equations (10):

$$p_2 = \left(p_1 + p_3 \frac{\zeta_A}{\zeta_f} \frac{r_m{}^2 h_f{}^2}{r_1^2 h^2} \right) \Big/ \left(1 + \frac{\zeta_A}{\zeta_f} \frac{r_m{}^2 h_f{}^2}{r_1^2 h^2} \right) \tag{11}$$

The dependence of the axial force on the pressure in chamber p_2 is obtained from (9):

$$p_2 = \left(p_1 + p_3 \frac{\zeta_A}{\zeta_f} \frac{r_m^{\,2} h_f^{\,2}}{r_1^2 h^2} \right) / \left(1 + \frac{\zeta_A}{\zeta_f} \frac{r_m^{\,2} h_f^{\,2}}{r_1^2 h^2} \right) \quad (12)$$

Expressions (11) and (12) are an implicit static characteristic of the ABD, namely, the dependence of the axial force on the value of the face gap $F(h_f)$ (**Fig. 4**). This relationship determines the value of the face gap at a certain axial working force. This allows us to further calculate the possible leakage of the ABD and pump efficiency.

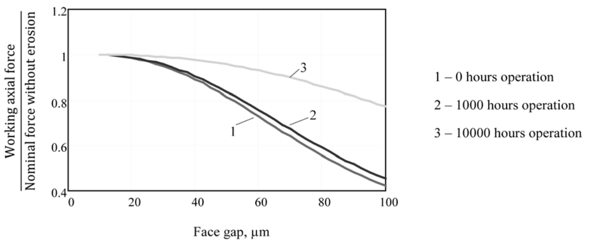

Fig. 4. The static characteristic of the ABD
Rys. 4. Charakterystyka statyczna tarczy odciążającej

Changes in the time of cylindrical throttle gap caused by erosive wear leads to a change of the face throttle gap to provide the constant value of operating axial force. **Figure 4** shows the static characteristics of the ABD for different values of the cylindrical throttle gap, namely, at an initial gap (1) and the gap value after 1000 (2) and 10000 hours (3). It is visible that the characteristic drops. Therefore, the value of face throttle gap of the ABD is bigger to provide the necessary unbalancing force. Moreover, increasing the face gap value leads to an increase in leakage and a decrease of pump efficiency. More informative changing the static characteristic of ABD in time while taking into account erosive wear of annular throttle is presented in **Fig. 5**.

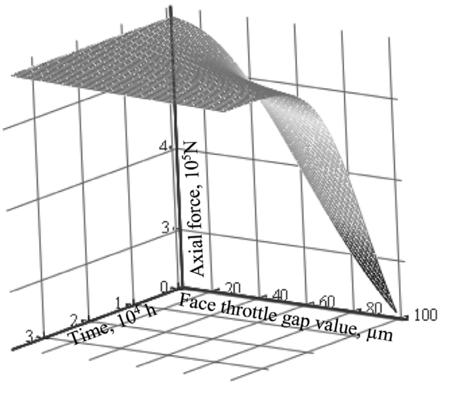

Fig. 5. Static characteristic of ABD while taking into account erosive wear of cylindrical throttle
Rys. 5. Charakterystyka statyczna tarczy odciążającej z uwzględnieniem erozyjnego zużycia pierścieniowej szczeliny

Obviously, the cylindrical throttle's resistance decreases under the increase of its gap value. It results in decreasing p_2 pressure in the chamber of the ABD (**Fig. 1**) to the pressure value at pump outlet (pump head). Therefore, the face gap opens and the ABD does not operate as an automatic device. Moreover, the pressure axial force on the disk of the ABD exceeds the axial force acting on the rotor. As a result, the displacement of rotor to the side reverse inlet leads to inevitable failure of pump.

CONCLUSIONS

The problems of feed-pumps failures under operation and possible methods of their solutions are possible, in relation to these failures resulting in the considerable decrease of the power unit capacity and, in some cases, in stopping of the power unit. The most frequent causes of failures (in order of their frequency) are the ABD, the seal, the rotor, and the bearing, according to the practice of this type of pump operation. Therefore, the ABD is one of the most responsible units of the feed-pump. Moreover, possible changes of operating characteristics of the ABD lead to the possible increase of axial mobility of rotor, which, in turn, determines the failure of other elements, such as seals and bearings.

According to calculations, erosive wear of cylindrical throttle's surfaces results in the middle gap increase and decrease of its resistance. Under the pressure drop value of 1MPa, the middle gap of cylindrical throttle value raises 1.5 times without considering of the movement of sealing surfaces, and 1.7 times when moving. Due to interaction of the hydrodynamic characteristics of cylindrical and face throttle, it leads to ABD characteristic changes. The static characteristic of the ABD drops 13% according to the research values of this study of the middle velocity in the gap and the reference coefficient of material wear. In particular, the increase of cylindrical throttle middle gap value after 10000 hours of operation leads to the loss of axial force self-regulation in a dependence on the face throttle gap value. In this case, the hydrodynamic force acting on the disk of the ABD becomes more than the operating axial force on a rotor. Due to this, the rotor moves in the opposite direction of the axial force. This rotor movement leads to the unregulated increase of the face gap value and, in turn, to the rise of leakages and a drop of pump efficiency. In some cases, it can result in the failure of other elements, such as seals and bearings.

It must be noted that the quality of feed-water has a substantial influence on the capacity and life of the feed-pump. The quality of feed-water is provided by the water preparation and cleaning systems of power plants. However, installing filters cannot always prevent the impact of solid parts in the pump, and these impacts can cause increase wear of impeller seals, bearings, and the surfaces of ABD. Therefore, the next problem of the influence of the presence of solid parts on the intensity of wear must be solved. It allows one to develop a method of ABD reliability calculation. In the present research, the first approach to the calculation of the critical value of face middle gap is made and the obtained results can be used to calculate the possible life time of the ABD.

REFERENCES

1. Matrsinkovski V., Szevczenko S.: Nasosy atomnych elektrostancji: rasczot, konstruirowanie, eksploatacjia. Monografia, 2018, Sumy, Universitetskaja kniga.
2. Pak P., Belousov A., Timszyn A.: Nasosy atomnyh elektrostancii, 1989, Moscow, Energoatomizdat.
3. Pak P., Belousov A., Belousov A.: Nasosnoje oborudowanie atomnych stancii, 2003, Moscow, Energoatomizdat.
4. Czegurko L.: Razgruzochnye ustrojstva pitatelnyh nasosov teplovyh elektrostancij, 1978, Moskva: Energija.
5. Czegurko L.: Centrobeznyje energeticzeskije nasosy, ich neispravnosti i metody ustranienia, 2002, Czeliabinsk.
6. Korczak A.: Badania układów równoważących napór osiowy w wielostopniowych pompach odśrodkowych, 2005, Wyd. Politechniki Śląskiej, Gliwice.
7. Kundera Cz.: Aktywne uszczelnianie drgających elementów wirujących. Monografia nr 15, 1999, Kielce, Wydawnictwo Politechniki Świętokrzyskiej.
8. Jedral W.: Turbulentny przepływ cieczy w hydraulicznie gładkich szczelinach poprzecznych, 1981, Archiwum budowy maszyn, T. XXVIII, №1.
9. Zueva N.: Issledovanie hydrodynamiki teczenia zydkosti v torcovom drossele s ekscentriczno raspolozennymi stenkami kanala, Udoskonaluvannia turboustanovok metodami matematycznogo i fizycznogo modeluvannia, 2003, Kharkiv, Instytut problem maszynobuduvannia im A. Pidgornogo NAN Ukrainy, pp. 607–610.
10. Black H.F.: Effects of hydraulic forces in annular pressure seals on the vibrations of centrifugal pump rotors, Journal of Mechanical Engineering Science, 11 (2) (1969), pp. 206–113.
11. Black H.F., Jenssen D.N.: Dynamic hybrid properties of annular pressure seals, Journal of Mechanical Engineering, 184 (1970), pp. 92–100.
12. Childs D.W.: Turbomachinery rotordynamics: phenomena, modelling, and analysis, Wiley (1993), pp. 275–277.
13. Baskharone E.A., Daniel A.S.: Rotor dynamic effects of the shroud-to-housing leakage flow in centrifugal pumps, Journal of Fluids Engineering, 116 (1994), pp. 558–563.
14. Kanemori Y., Iwatsubo T.: Experimental study of dynamic fluid forces and moments for long annular seal, ASME Journal of Tribology, 114 (1992), pp. 773–778.
15. Finnie I.: Some observations on the erosion of ductile metals, Wear, 1972, 19, pp. 81–90.
16. Hashish M.: Proc. 7th Int. Conf. on 'Erosion by liquid and solid impact', Cambridge, UK, 1988, Cavendish Laboratory, Paper 66.
17. Tarasevych Y., Savchenko I., Sovenko N., Savchenko A.: Research of influence of random change of annular seal parameters on efficiency of centrifugal pump. Eastern-European Journal of Enterprise Technologies, vol. 6, №7 (84), (2016), pp. 37–42.
18. Ratner A., Zelenski V.: Erozia materialov teploenergeticzeskogo oborudovania, 1966, Moscow, Energia.