Influence of operational changes of clearances in pump channels on the work of the automatic balancing device

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Abstract. Automatic balancing devices are used to balance the axial force in many multistage centrifugal pump designs. Despite the obvious advantages of these devices their failure is one of the main reasons for the complete failure of the pump unit. The axial force acting on the pump rotor can be changed under pump operation in a sufficiently large range and up to several tens of tons. Usually the calculation of automatic balancing device is made under pump's operating parameters. At the same time it is possible the changing of value of the axial force generated by the automatic balancing device on the calculated parameters due to the interdependence of hydrodynamic processes in clearances of the flow part of the pump and in the clearances of the balancing device. The analysis of possible causes of the axial force value changes as a result of unbalanced pressure forces acting on pump impellers and counterbalancing of axial forces arising in face gap balancing device is presented. An improved methodology for calculating the estimation of the axial force arising in the balancing device using the example of industrial multistage centrifugal pump is proposed.

1. Introduction

The axial forces acting on the rotor $T$ are balanced by special automatic balancing devices (Fig. 1) in many designs of high-pressure multistage pumps. This is due to such device is the most effective balancing system, because it acts as thrust bearing with automatic adjustment of the bearing capacity capable of operating in a wide range of axial forces and as the end seal through which almost the total pressure of the pump is throttled [1-4].

However, according to the work [3] hydraulic balancing device malfunction or failure are in third place among the most common causes of failure of the pump unit. Moreover, the construction and the operation of hydraulic balancing device significantly affects the amount of volumetric and mechanical losses, which in turn leads to a decrease in pump efficiency. Thus in some cases the total losses of hydraulic balancing device reach up to 10% of the pump power [1,5]. And this in turn negates any designer efforts to increase the efficiency of the pump.

When the hydraulic balancing device designing, a direct problem is solved: the corresponding axial force and the flow rate values are calculated under the geometric parameters of device.

At the same time the volume losses are directed to a minimum (reduce the size of the face gap). However, it is necessary to prevent excessive reduction of the end gap due to changes in operation of the total axial force acting on the rotor. As shown in [1], these deviations can reach up to 50%. Search
for alternative solution improving some characteristics while worsening others stands at the design stage of the device. At the same time that most of the initial values are probabilistic in nature.

\[ F_p = \frac{p_3}{p_{in}} \]

\[ S_F = \frac{p_{in}}{p_{out}} \]

\[ T = p_{2} p_1 \]

\[ I \]

**Figure 1.** Automatic balancing device.

After the design phase errors in the hydraulic balancing device manufacture and assembly cause the flatness working surfaces of unloading and stationary discs. As a result, the average face gap increases, which entails an increase in losses. Furthermore, during operation of the balancing device, deformations of the rotating and non-rotating discs occur, which in turn affect the static and dynamic characteristics of the device. At present there are designs of automatic unloading devices in which the movable ring follows the initial misalignment of the face gap surfaces and helps to maintain a flat clearance [5-6]. However, to completely avoid the flatness of working surfaces during operation is not possible.

Initial misalignment, power deformations during operation, the gap in the cylindrical throttle and the loss coefficients in the device itself and the pump flow part depend on the accepted tolerances, assembly and manufacturing errors, the properties of the pumped medium, its temperature, and also the pump operating parameters. Used mathematical models for calculating the static and dynamic characteristics of the balancing device do not take into account the fact that some geometrical parameters are random functions. This work is a continuation of previously published works on the influence of random changes autobalancing device parameters on its static, dynamic and flow characteristics. In this paper, estimation of a random change in time of the clearance of a cylindrical throttle on the static characteristics of hydraulic balancing device is done.

2. Analysis of literature and problem statement

To date an effective mathematical method has been developed for calculating the static and dynamic characteristics of automatic unloading devices in a deterministic setting. Thus, in [7] a method of the static characteristic calculation is given - the dependence of the face gap value on the balanced force acting on the rotor. It should be noted that in the majority of works devoted to the calculation characteristics of the balancing device do not take into account the fact that some geometrical parameters are random functions. This work is a continuation of previously published works on the influence of random changes autobalancing device parameters on its static, dynamic and flow characteristics. In this paper, estimation of a random change in time of the clearance of a cylindrical throttle on the static characteristics of hydraulic balancing device is done.
system was carried out in [1, 7, 16]. In this case, the adjustable value is the axial position of the rotor, and the external impacts are the axial force acting on the rotor $T$ and the pressure at the inlet to the cylindrical throttle $p_1$. Analysis of the dynamic state of the balancing device system is carried out according to linearized equations in the field of small changes in the equilibrium face gap [1, 16]. In previous works, the effect of a random change in the parameters of the annular seal on the efficiency of the pump, as well as a random change in the coefficients of local hydraulic losses on the static and dynamic characteristics of the balancing device are evaluated. In this paper, engineering calculation method of the static characteristic of hydraulic balancing device, taking into account a random change in the average radial clearance in a cylindrical throttle, a random change in the coefficients of local hydraulic losses in both a cylindrical and face throttle, as well as the initial misalignment of a rotating and non-rotating discs is given.

3. Probabilistic calculation method of static characteristic of automatic balancing device

In static calculation of automatic balancing device the dependence of the face gap of the balancing device on external impact, in particular, on the axial force $T$ which value during operation may significantly deviate from the calculated one, is studied.

The purpose of the static calculation is to select such basic geometric parameters which allow to obtain the values of the face gap and flow within acceptable limits in a given range of variation of the balanced force.

From the fluid continuity equations, the fluid flow through a cylindrical throttle $Q_c$ is equal to the flow through the face throttle $Q_f$ of the device:

$$Q_c = Q_f.$$  \hspace{1cm} (1)

From the solution of the flow motion in cylindrical throttle with taking into account the eccentricity of rotor and sleeve axes can be find:

$$Q_c = g_c \sqrt{p_1 - p_{20}} = 2\pi r q_{p0} \left(1 + \frac{3}{4} \left(1 - \frac{\Delta \xi}{\xi_c} \right) \epsilon \xi_c^2 \right)$$  \hspace{1cm} (2)

where

$q_{p0} = \left(\frac{4\Delta p H^3}{\rho \xi_c^2}\right)^{1/2}$ – flow-rate through the concentric cylindrical throttle, $\Delta p$ – throttled pressure drop.

$H$ - radial gap in the middle section along the throttle length, $\xi_c = \Delta \xi + \frac{c \xi}{2H} - \Theta \xi_m$ – common coefficient of resistance in conical throttle, $\Theta$ – relative angel of sleeve taper, $\Delta \xi = \xi_{11} - \xi_{12}, \xi_m = \xi_{11} + \xi_{12}, \xi_{11} - \xi_{12}$ – coefficients of local hydraulic resistances at the inlet, $\xi_{12}$ – at the outlet of the cylindrical throttle.

Calculation of flow rate characteristics of face throttle is made taking as an assumption isothermal and quasi stationary flow in the gap. Laminar model of flow is taken. Inertia forces are neglected. So, after solution of flow motion problem in face throttle, the flow rate value:

$$Q_f = g_f \sqrt{p_2 - p_3},$$  \hspace{1cm} (3)

where

$g_f = 2\pi r h f \sqrt{\frac{2}{\rho f}}$ – conductivity of face throttle, $\xi_f = \xi_{fl} + \xi_{ft} - \xi_{fo}$ – coefficient of total losses in face throttle, $\xi_{fl}$ and $\xi_{fo}$ – coefficients of local losses in the inlet and the outlet of the face throttle, $\xi_{ft}$ – coefficient of losses on the length, $\lambda$ - coefficient of friction resistance for the corresponding regime of fluid flow.

Initial value of middle gap of cylindrical throttle can be modeled by normal distribution. Parameters of the distribution are known: mean value and standard deviation. Interval of possible values of middle gap is determined by engineering tolerances. For some cases values of tolerances could reach up to 30% from initial value.

Initial value of middle gap of cylindrical throttle can change in time under pump operation due to wear process of sealing surfaces. Wear can be caused by occasional contact, erosion, corrosion and abrasion due to solid inclusions present in the pumping medium. In this work the erosion wear of
An increase in the middle radial gap value of cylindrical throttle by the action of steam-water mixtures can be modeled by an exponential dependence from middle velocity in gap. This dependence has different form for each specific material and pressure drop value. For example, to evaluate changes in the middle radial gap of cylindrical throttle with a nominal gap value of 250 μm the following dependence for steel AISI 321 can be used

\[ H = 2.1 \cdot 10^{-4} \exp(4.507 \cdot 10^{-5}t) + 5.327 \cdot 10^{-13}t^2. \] \hfill (4)

The eccentricity of the cylindrical gap depends on many random factors. The boundaries of its change are determined by the values of the radial middle gap value. Since for the relative movement of the rotor all directions are equivalent and can only be positive, its probability density can be described by the truncated Rayleigh law \cite{17}

\[ f(e) = \frac{7.882}{H^2} e \exp\left(-\frac{3.858e^2}{H^2}\right). \]

The normal distribution law was taken according to the central limit theorem of probability theory for other random variables: \( f_i(x_i) = \frac{1}{\sqrt{2\pi}\sigma_i} \exp\left(-\frac{(x_i-\langle x_i \rangle)^2}{2\sigma_i^2}\right) \). Mean values of each distribution are set as deterministic values of corresponding variables: \( \langle h_1 \rangle = h_1(t), \langle \xi_{f1} \rangle = \langle \xi_{c1} \rangle = 1.5, \langle \xi_{f0} \rangle = \langle \xi_{c0} \rangle = 0.2\langle \xi_{f1} \rangle \), and standard deviations are calculated on the bases of empirical rule \( \sigma_i = \frac{\Delta_i}{3} \).

Probability density of pressure in automatic balancing device chamber can be find from equations (1)-3

\[ f_{p2}(p_2) = f(h_1, \xi_{c0}, e, \xi_f, t) = f_1(h_1(t))f_2(\xi_c)f_3(\xi_f)f_4(e|h_1) \] \hfill (5)

where \( f_i(x_i) \) - probability density of corresponding random parameters: \( x_1 = h_1(t), x_2 = \xi_c, x_3 = \xi_f, f_4(e|h) \) - probability density of eccentricity under condition \( h(t) = h(t*) = \text{const} \), \( t* \) - operation time of the pump.

The equation of axial balance:

\[ T = F, \] \hfill (6)

where \( F = p_2S_c + F_c - p_2(S_c + S_e) \) - resulting axial force acting on rotor. The pressure force in automatic balancing chamber \( (p_2S_c) \) and pressure force in face throttle with taking into account the possible deformation of surfaces \( F_c \) are the components of this force:

\[ F_c = S_c = \left[p_2 + p_2 - p_2\left(\frac{2\Lambda + 3\beta}{3 \xi_f} \frac{\xi_{f1} + \xi_{f0}}{\xi_f}\right)\right], \]

where \( \Lambda = \frac{b}{2r_f} \), \( \beta = \frac{b}{2r_f} \) - relative taper of face throttle.

From equations (1) and (3), a deterministic dependence of the axial force on the value of the end gap can be obtained

\[ F = p_2\left[S_e + \frac{S_c}{2} \left(1 - \frac{2\Lambda + 3\beta}{3} \frac{\xi_{f1}}{\xi_f} \frac{\xi_{f1} + \xi_{f0}}{\xi_f}\right)\right] - p_3\left[S_e + \frac{S_c}{2} \left(1 - \frac{2\Lambda + 3\beta}{3} \frac{\xi_{f1}}{\xi_f} \frac{\xi_{f1} + \xi_{f0}}{\xi_f}\right)\right]. \] \hfill (7)

Static calculation of ABD systems bases on getting the dependence of face throttle gap value on external forces in general on axial force \( T \) (6). This relationship determines the value of the face gap at axial operating force value and allows to calculate the possible values of hydrodynamic balancing.
force in face throttle under different initial random values of middle gap in cylindrical throttle and random values of resistance coefficients.

4. Numerical solution
Calculation of static characteristic of automatic balancing device is made under such values of main parameters (fig. 1): \( r_1 = 55 \) mm, \( r_2 = 77.5 \) mm, \( r_3 = 100 \) mm, \( h_1 = 200 \) \( \mu \)m, \( l_1 = 200 \) mm, \( p_1 = 35 \) MPa, \( p_3 = 3 \) MPa.

The static characteristic of automatic balancing device with taking into account possible changes of middle radial gap of cylindrical throttle due to erosive wear of sealing surfaces can be calculated on the bases of equations (7) and (4).

![Figure 2](image2.png)

**Figure 2.** The static characteristic of the ABD under different values of cylindrical throttle gap.

Cylindrical throttle resistance decreases under increasing of its gap value. It results in decreasing of \( p_2 \) pressure in the chamber of ABD closer to the pressure value at pump outlet (pump head) and to a change of face throttle gap value under constant value of operating axial force. So, the face gap opens and ABD does not operate as an automatic device. Increasing the face gap value leads to an increase of leakage and decrease of pump efficiency.

Figure 3 shows the pressure distribution in the automatic balancing chamber as a function of the middle gap value of face throttle taking into account a random change in the characteristics of the cylindrical throttle due to manufacturing tolerances and hydraulic loss coefficients.

![Figure 3](image3.png)

**Figure 3.** Dependence of pressure change in chamber of automatic balancing device on value of middle gap in face throttle with taking into account random changes of losses coefficients and gap value in cylindrical throttle.
According to fig. 3 the possible values of pressure $p_2$ in automatic balancing device could be up to 9-11% smaller or higher than the calculated ones with confidential probability 99.73%.

On fig. 4 the probabilistic densities of axial force are shown for the different values of middle gap in throttles.

![Figure 4. Probabilistic densities of axial force for different values of radial gaps of cylindrical and face throttle.](image)

The standard deviations of axial force decrease with middle gap value of cylindrical throttle increase. So the influence of random changes of initial value of middle radial gap in cylindrical throttle due to engineering tolerances is getting smaller.

5. Conclusions

Due to interaction of hydrodynamic characteristics of cylindrical and face throttle the value of middle radial gap of cylindrical throttle determines the automatic balancing device operating characteristics. Deviations of its value due to engineering tolerances can result in change of pressure $p_2$ in chamber up to 9-10% for calculated model of automatic balancing device. It can result in the flow rate change as well as the pump efficiency.

For some designs of the automatic balancing device the deviations of the force acting in the face gap can have non-constructive values since the selection of geometric characteristics is carried out on specific axial force values. Changes in time of middle radial gap of cylindrical throttle values caused by erosive wear leads to a change of face throttle gap value under constant value of operating axial force. For calculated model the static characteristic of ABD drops on 13%. In this case the value of face throttle gap of the ABD will be larger to provide the necessary.

The axial force acting on the rotor of a centrifugal pump is determined by the distribution of pressure in the flow part. When changing the value of the cylindrical gaps on the front and interstage seals, there is an increase in flow rate and as a result a change in the nature of the flow. The pressure diagram along the radius changes from linear to concave which leads to an increase in axial force value. The change in the cylindrical gaps is random therefore the axial force will also be a random variable and in general case a random function. So, under designing an automatic balancing device the inverse problem of estimation the influence of the random nature of axial force on the characteristics of the device should be solved. In particular the face gap which determines the volumetric efficiency of the pump and the reliability of the automatic balancing device should be calculated.

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