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The effect of the deformation of the face throttle surface on the operation of the balancing disk of a multistage centrifugal pump

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Abstract. This paper presents selected results of numerical and analytical analysis of a traditional design of an axial force balancing system in a multistage centrifugal pump. The value of the axial force generated in the face throttle of such a system is directly influenced by the geometry of the throttles: length and height. In the process of operating a multistage pump, the surfaces that form the face throttle are deformed due to high values of pressure or temperature. Changing the geometry results in a change in the pressure distribution, and thus in the value of the generated axial force and hydraulic losses. The analytical approach provides a simplified theoretical estimation of disc deflection and force balance, while the numerical simulations capture the detailed pressure distribution and corresponding structural response for different face gap values. The practical value of this research lies in providing a deeper understanding of the relationship between balance disc deformation and axial force. The results enable more accurate selection of disc geometry and face gap value, helping to minimize residual thrust on bearings and hydraulic losses on the automatic balancing device.

Key words: annual throttle, face throttle, axial force, multistage pump, FSI

1. INTRODUCTION

Multistage centrifugal pumps are widely used in many applications, and they play an extremely important role in the chemical, mineral, and petroleum industry. The design of such pumps is very complicated, as consist of many stationary and rotating components. The geometry of the running part affects the main hydrodynamic characteristics such as head, flow rate, efficiency, and dynamic characteristics of the pump. Rotation of the pump rotor creates a complex pressure field in the running part of the pump. As a result, radial and axial forces act on the rotor during pump operation. The axial force is the biggest by its value and, depending on the design and operation parameters, can be up to several hundreds of kilonewtons (for multistage feed pumps for example) [1-3]. Thus, the axial force plays a critical role in the stability and life-time of multistage pumps.

This force is caused by the lack of symmetry of the impeller construction in the plane, which is perpendicular to the rotor axis. The outer surface of the main disc 2 is larger than the opposite surface of the cover disc 1 (Fig. 1). It leads to the axial pressure force $\boldsymbol{T}$, which is directed to the inlet. The more stages are in a pump construction, the greater the axial force generated.

Two types of balancing devices are mainly used to balance the axial force in multistage pumps with impellers facing the same direction of suction (single-suction). There are balance drums (balance pistons) and balancing discs. Both of them have their advantages and disadvantages. The balancing drum cannot fully compensate for the axial force, especially if it is not constant and therefore axial bearings are required. In addition, the grooved surface of the drum can cause a cavitation phenomenon and erosion of the drum material [4]. It can result in higher volumetric losses and bigger loads on the bearings.

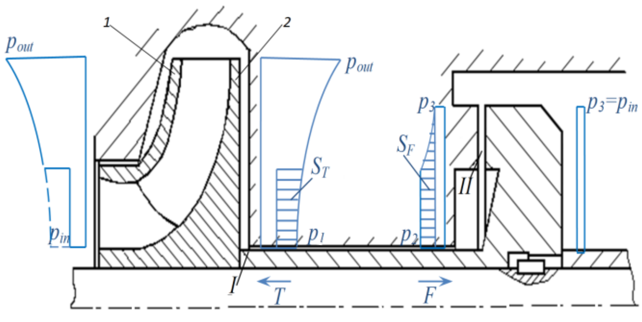

Fig.1. Axial force balancing assembly of multistage pump with balance disc

The balance disc, as a design element of the pump, allows for complete balancing of the axial force under varying operating conditions because it exhibits the ability to self-regulating [1, 3, 5]. Unlike other design solutions for balancing the axial force, no additional assemblies/components or fluids are required for the stable operation of the balance disc, it also

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performs the role of the end seal on which the total pressure of the pump is throttled. A properly designed balancing assembly allows monitoring of the change in the axial force acting on the pump rotor due to the feedback between the axial force and the width of the face throttle. Namely, as the axial force acting on the pump's rotor increases, rotor moves to the left (suction). This leads to a decrease in the width of the face throttle ***II***, which in turn increases the resistance to fluid flow through the gap, thereby increasing the pressure in the chamber of the balancing assembly. Due to the increase in pressure in the chamber, the resultant pressure force, which balances the axial force, also increases.

The balance device with the balance disc is the most heavily loaded unit, and its hydrodynamic characteristics mainly determine the efficiency of the entire pump. Therefore, the basic reliability and efficiency of the pump are largely determined by the performance of the axial force balancing assembly.

Designing a balancing device of a multistage pump involves solving several problems. First, determining the value of the axial force and the range of possible variations in its value. Calculation of axial force is mostly based on semiempirical models. Analytical methods are based on the assumption that, in the areas between the rotor discs, the fluid rotates as a solid with an angular velocity equal to half the rotor angular velocity [1, 3]. The most detailed and widespread analytical approach is carried out by Gülich [3]. Modified analytical model with pressure correction coefficient was developed in [6]. The use of numerical methods allows for obtaining more accurate values of this force but does not always allow determining the range of its possible changes. The critical analysis of the current state of the art in flow inside the sidewall gaps of hydraulic pumps and turbines is given in [7]. Axial force values are also studied experimentally in [1, 8-10], mainly close to the nominal operating point and for normal operating modes. A couple of numerical studies of the axial force were carried out by different researches [11-14] for the past few years. Despite the advanced capabilities of modern commercial and open-source CFD software, predicting pressure distribution and axial force in multistage pumps remains challenging. This is due not only to difficulties in modeling extremely small clearances, turbulence, secondary flows, and precise boundary conditions, but also because each multistage pump exhibits unique flow characteristics. Variations arise from manufacturing and installation tolerances, and operational wear, which alter flow geometry and hydraulic behavior in ways that are difficult to numerically calculate.

Another problem is the determination of the hydrodynamic characteristics of the balancing assembly itself. These characteristics depend as much on the applied geometric values of the assembly as on the value of the axial force obtained at the previous design stage. Analytical models for determining the hydrodynamic characteristics of the automatic balancing device include the hydrodynamic models of the face and cylindrical throttle. Mostly, the two-dimensional Reynolds equation is used for pressure distribution in a circular throttle due to small changes in the flow parameters in circumferential and radial directions [2,14-16]. Different calculation models of the face throttle are presented in [17-19]. In [17,18] the influence of the non-flat shape of the clearance in face throttle and inertia terms on pressure distribution is analyzed. It should be noted that the values of gaps of the cylindrical and face throttles change during the operation of the pump due to tolerances, pressure and temperature deformations, or surface wear. Considering the fact that the tightness and reliability of the balancing device determine the reliability and efficiency of the pump in total, high demands are placed on the accuracy of the calculation of this device. However, the existing analytical methods for calculating the characteristics of such a balancing device, used in engineering practice, are based on simplifications, without which this problem has no solution. These simplifications often lead to rather large coarsening of the obtained results, which can subsequently result in a large deviation in the calculated and operating parameters and, consequently, to an incorrect selection of the geometric parameters of the device at the design stage. Existing numerical design capabilities make it possible to more accurately solve the joint problem of hydrodynamics of fluid flow in the cylindrical and face throttle of balancing device, taking into account possible force and thermal deformations of the throttle surfaces.

Despite some progress in the study of balancing devices, several research gaps remain unsolved. These include the need for fully coupled fluid–structure interaction models, investigation of transient and thermal effects, and a better understanding of surface roughness and wear influence on the axial force value.

There are two commonly used strategies [20, 21] for solving fluid-solid interaction (FSI) problems. First, the so-called one-way FSI in which the information exchange between subprocessors is provided directly only once without any iteration for overall solution in time. Under two-way FSI, the iteration process for overall solution is used and data from one subtask is transferred into another subtask at each time step. Under studying balance disc operation, the main engineering interest is the steady leakage flow and hydraulic force at the design clearance. The disc achieves an equilibrium position during operation, after which displacement is negligible relative to the steady state. Therefore, one-way coupling is sufficient: CFD provides the pressure and leakage at prescribed clearances, while the structural model verifies the value of deformations of the disc. To account for possible axial displacement in calculations, values of face throttle with a constant step are used. Two-way FSI is extremely computationally expensive and may be required when investigating transient instabilities, vibrations, or wear dynamics, which is beyond the scope of this work.

A general difficulty in the development of reliable balancing devices is the limited availability of experimental validation data obtained under realistic operating conditions. Since systematic test results are rarely available, engineers must often rely only on simplified analytical models, which remain the main tool for designing such devices. In this context, the present study demonstrates that combining numerical simulations with the analytical approach provides an effective way to improve existing engineering methods and to enhance the reliability of design predictions.

The originality of this study lies in a more detailed analysis of the structural response of the balancing disc under hydraulic load in a multistage centrifugal pump. Previous studies have mainly focused on leakage through balancing discs, it's dynamic characteristics [17], stability and reliability [22], whereas this study concentrates on how the pressure distribution in the device affects the deformation of the disc and the magnitude of the axial force imposed on the rotor. Using one-way FSI simulations for various nominal clearances between surfaces, the study provides new insights into the relationship between deformation and axial force, which is directly relevant to reducing residual axial force on thrust bearings. The results serve not only as reference material for optimizing disc geometry and stiffness, but also as a guide for future experimental verification.

2. METODOLOGY. ANALITYCAL APPROACH

Analytical axial force calculations and simulations in ANSYS CFX were performed for a seven-stage 180-1050 centrifugal pump. According to the data of the geometric and operating parameters of the pump, the total axial force value T = 158.7 kN. This value is calculated using the methodology given in [3].
The calculated model of the balancing device is shown in Fig. 2.

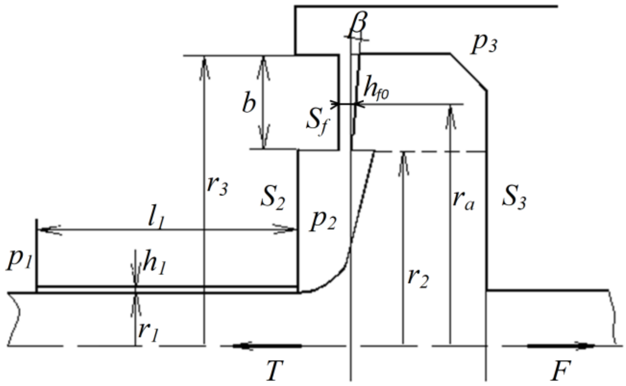


Fig. 2. Calculated model of hydraulic balancing device

The geometric parameters are presented in Table 1.

TABLE 1. Geometric parameters of the calculated model

l_1	0.115 m	r_2	0.09 m
r_1	0.05975 m	r_3	0.115 m
h_1	2.5 ·10^{-3} m	h_{f0}	(6.5; 8; 9.5, 12) ·10^{-5} m

In order to determine the axial force, which is generated in the face throttle and balances the axial force of the pump, it is necessary to obtain the pressure distribution in this throttle. This distribution depends on the flow characteristics of the pump (its running part geometry and operating parameters) as well as on the height and length of the cylindrical throttle. Assuming that the pressure value at the outlet of the last stage of the pump is the pressure value at the inlet to the cylindrical throttle of the balancing device p_1 (Fig. 2), the first boundary condition can be obtained. In most multistage pump designs, the outlet of the face throttle is connected by a pipe to the inlet of the pump. Therefore, the second boundary condition for the pressure value p_3 at the outlet is also known.

The static characteristic of the balancing device is determined by the equilibrium conditions of the axial discharge disc:

$$T = F, \quad (1)$$

where
$F = p_2S_2 + F_f - p_3S_3$ − total axial force of pressure acting on the rotor, which consists of the pressure force in the chamber (p_2S_2) and the pressure force F_f in the face throttle;
S_2, S_3, S_f − areas of the corresponding surfaces (Fig.2).
The value of the pressure p_2 in the discharge chamber can be calculated from the balance of fluid flow Q_c and Q_f through the cylindrical and face throttles, respectively:

$$Q_c = Q_f, \quad (2)$$

which can be represented as

$$Q_c = g_c\sqrt{p_1 - p_2},\ Q_f = g_f\sqrt{p_2 - p_3},$$

where
$g_c = 2\pi r_1 h_1 \sqrt{2(p_1 - p_2)/(\rho\zeta_c)}$,
$g_f = 2\pi r_a h_{f0}\sqrt{2(p_2 - p_3)/(\rho\zeta_f)}$ − conductivity of the cylindrical and face throttle respectively;
$\zeta_f = \zeta_{if} + \zeta_{2f} - \zeta_{of}, \zeta_c = \zeta_{ic} + \zeta_{2c} - \zeta_{oc}$ − total loss factor of face and cylindrical throttle;
ζ_{if} and ζ_{of}, ζ_{ic} and ζ_{oc} − local losses factor at the inlet and outlet of the face and cylindrical throttle;
$\zeta_{2f} = \frac{\lambda_r b}{2h_{f0}}, \zeta_{2c} = \frac{\lambda l_1}{2h_1}$ − loss factor along the length of the face and cylindrical throttle correspondently;
$r_a = 0.5(r_2 + r_3)$ − average radius of the face throttle.
From Eq. (2) the pressure in the discharge chamber p_2 of balancing device

$$p_2 = \left(p_1 + p_3 \frac{\zeta_c}{\zeta_f}\frac{{r_a}^2{h_{f0}}^2}{{r_1}^2{h_1}^2}\right) / \left(1 + \frac{\zeta_c}{\zeta_f}\frac{{r_a}^2{h_{f0}}^2}{{r_1}^2{h_1}^2}\right). \quad (3)$$

To determine the pressure force in the face throttle, the equations of viscous incompressible liquid motion (averaged Navier-Stokes) and the continuity equation are considered:

$$\begin{cases} \frac{dp}{dr} - g_r = -\frac{k_r\mu}{h^2}\bar{v}_r \\ \frac{\partial}{\partial r}(\bar{v}_r hr) + ru_1 = 0 \end{cases} \quad (4)$$

where
$h_f = h_{f0} + z + (r - r_a)\beta$ − the width of the face throttle;
h_{f0} − average width of the face throttle
β − taper of the face surfaces;
z −axial displacement of rotating disc face;
$u_1 = \dot{z} = \frac{dz}{dt}$ − axial velocity of the rotating disc;
g_r − averaged components of inertia forces.
In this work, the isothermal flow is considered. So, the 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 the length of the throttle (15-25 mm), the flow velocity component changes in the axial direction and its derivatives are neglected. The fluid pressure is primarily caused by the static pressure drop, which causes the main pressure flow of the fluid along the radius. For mechanical face seals, the gap and flow rate are small, and the fluid flow regime is laminar. However, for the face throttle of the balancing device, which is characterized by large pressure drops and relatively large width of gaps, the flow is turbulent, which results in local hydraulic losses.
Solution of the Eq. (4) can be obtained by taking into account the boundary conditions for pressure:

$$p|_{r=r_2} = p_{20} - \zeta_{if}\frac{\rho v_{ri}^2}{2}, \; p|_{r=r_3} = p_{30} - \zeta_{of}\frac{\rho v_{r0}^2}{2}.$$

where v_{ri} and v_{r0} − the fluid velocity at the inlet and outlet of the face throttle.
After integrating the pressure distribution along the area surface of the face throttle, the hydrostatic component of the axial force can be presented in a form:

$$F = p_2 S_2 - p_3 S_3 + \\ +(p_2 - p_3)\frac{S_f}{2}\left(1 - \frac{2\Lambda + 3\bar{\beta}}{3}\frac{\zeta_{2f}}{\zeta_f} - \frac{\zeta_{if}+\zeta_{of}}{\zeta_f}\right), \quad (5)$$

where
$\Lambda = \frac{b}{2r_a}, \bar{\beta} = \frac{b\beta}{2h_{f0}}$ − dimensionless values.
Analyzing Eq. 5, it should be noted that taking local hydraulic losses into account leads to a reduction in axial force in face throttle.
The deformation of an initially flat disc under the action of a balancing force F applied to the disc can be calculated by assuming it is evenly distributed over the effective area of the disc. In this case, the distributed load:

$$q = \frac{F}{S_2 + 0{,}5S_c(1-\Lambda)},$$

where
$S_c = \pi(r_3^2 - r_2^2)$ – the area of the face throttle surface;
$S_2 + 0{,}5S_c(1-\Lambda)$ − the effective area of the disc.
Considering the disc as a plate of constant thickness g rigidly fixed along the radius r_2 the deflection of the disc can be calculated by the formula [23]

$$w = K\frac{q \cdot r_3^4}{E_1 g^3},$$

where
$K = f\left(\frac{r_1}{r_3}\right)$ – a dimensionless parameter that is a function of the ratio $\frac{r_1}{r_3}$ and the values of this function are calculated for Poisson's ratio 0.3;
g − the thickness of the balancing disc;
E_1 is the modulus of elasticity of the disc material.
Taper of the face surfaces, which is determined by the deflection of the disc

$$\beta = \frac{w}{r_3 - r_1}. \quad (6)$$

When the disc is deformed, the values of the balancing force and the face gap decrease. According to the design requirements, the disc thickness g should be selected so that

$$\frac{\beta b}{2h_{f0}} \leq 0.1.$$

The last criterion is used for the geometry parameters of the face throttle, but the thickness of the disk g should be selected in such a way, that the deflection of the disk for pressure (distributed load) in face throttle results in the tape angel, which is not greater than $\frac{0.2h_{f0}}{b}$.

3 NUMERICAL SIMULATION

There are practically no peer-reviewed papers or chapters where joint fluid flow in cylindrical throttle-chamber-face throttle of balance disc of centrifugal pumps is numerically simulated. The main problem under simulating the fluid flow in such a system is the generation of a grid, allowing us to obtain reliable results. In balancing devices, there is a large difference in geometric dimensions: the length of the cylindrical throttle is about 0.09÷0.15 m, while the height is $1 \div 3 \cdot 10^{-4}$ m; the length of the face throttle is: 0.02-0.04 m, while the width is within $6 \div 12 \cdot 10^{-5}$ m. It directly affects the quality (dimensions) and quantity of volume elements.
Since FSI analysis is not feasible on 2D grids, this paper considers a 0.05° sector of the balancing device assuming axisymmetric flow. In order to obtain appropriate flow characteristics at the inlet to the cylindrical throttle and to avoid the influence of the inlet vortex on the boundary, the inlet in the simulation was located at a distance of 30 mm from the real inlet (Fig. 3). The independence of the flow rate and the value of the hydrostatic component of the force from the number of grid elements is used as a criterion to check the sensitivity of the results to the density of the grid.

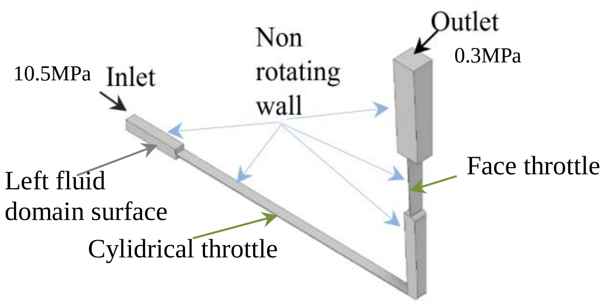


Fig. 3. Schematic of the numerical model of sector of balancing device with boundary conditions

Steady flow in the balancing device without misalignment is considered. The problem is solved in an axisymmetric formulation with pressure boundary conditions. The boundary conditions of 10.5 MPa total pressure is set at the inlet. The static pressure of 0.3 MPa is set at the outlet. The value of 2960 rpm is set for the rotating wall. For stationary wall, the no-slip boundary condition is set. Rotational periodicity is set on the left and right (is hidden on the Fig. 3) domain surfaces, so the flow in one segment is periodically identical to the others, but rotated by a fixed angle. Numerical computations are performed under double precision to ensure the stability and reliability of the results.

The pressure distribution along the length of the cylindrical and face throttle and, respectively, the value of the axial force depends significantly on the velocity at the inlet and outlet of these throttles. At the inlet to the cylindrical/face throttle, the flow undergoes a sudden narrowing, which can accompany the appearance of a vortex region in the section of narrowing. According to Bernoulli's equation, the growth of velocities at the inlet section is accompanied by a drop in pressure. After passing the inlet cross-section of the cylindrical/face throttle, the flow expands maximally, being limited by the constant cross-section of the throttle. At the outlet of the throttles sudden flow expansion takes place and development of a vortex regions is also possible. So, the two-equation eddy-viscosity *k-ϖ* SST (share stress transport) model is used for modeling the turbulence under the calculations. This model combines the advantages of the *k-ε* and *k-ω* models [24, 25] and includes an additional component limiting the overestimating of turbulence kinetic energy in areas of strong pressure gradients. This model provides the most accurate results for near-wall calculations.

Before initiating the numerical simulations, a mesh independence study was conducted to evaluate the effect of mesh resolution on result accuracy. The results for the smallest value of h_{f0} = 65 μm are shown in Fig. 4.

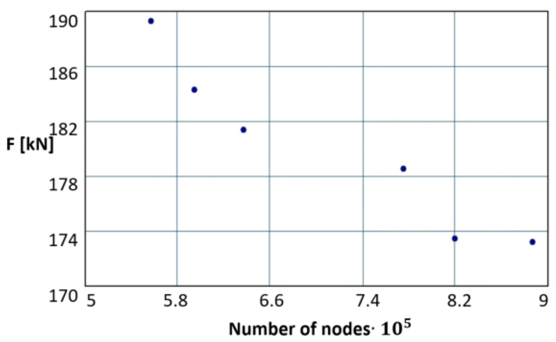


Fig. 4. Mesh independence study results

The value of dimensionless wall distance y+<1 is obtained over the length of the both throttles. To ensure such value of the y+ an additional 10 layers on stationary and rotating faces of the fluid domain is used for flow simulation. The increasing of the additional layers numbers above 10 doesn’t influence the value of the mass flow rate and axial force value. After the check of mesh sensitivity polyhedral mesh grid with 3043577 nodes, 3905775 faces and 818869 cells and additional 8 layers on stationary and rotating faces is used for simulation.

4 RESULTS AND DISCUSSION

Presented in Sec. 2 and 3 models make it possible to obtain the pressure distribution in the automatic balancing device and the axial force values for different values of the face throttle width h_{f0}. Results of the one-way FSI simulation allow to compare the face throttle taper angle with the analytical model and clarify the influence of face throttle deformations on the operation of the automatic balancing device.

4.1 Comparison with the analytical model

A comparison of the simulation results and the analytical model for the pressure value p_2 in the discharge camera is shown in Fig. 5. According to Fig. 5, the difference between analytical and numerical solution becomes larger as the width of the face throttle increases. The maximum difference for the throttle width of 12 micrometers reaches 18%. The deviation in the value of pressure p_2, which at the same time is the boundary condition for the fluid motion in the face throttle, directly affects the value of the axial force generated in the balancing device.

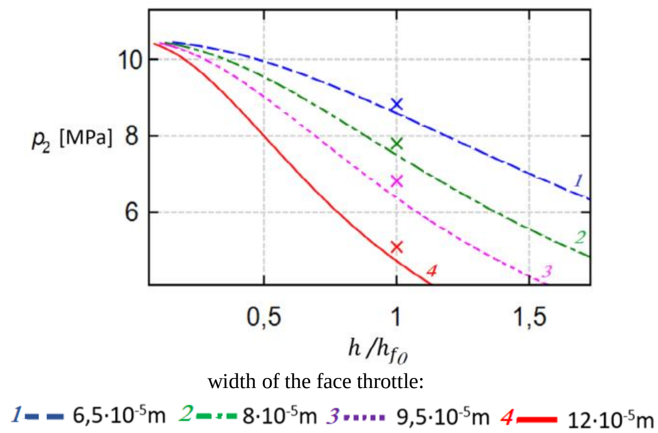
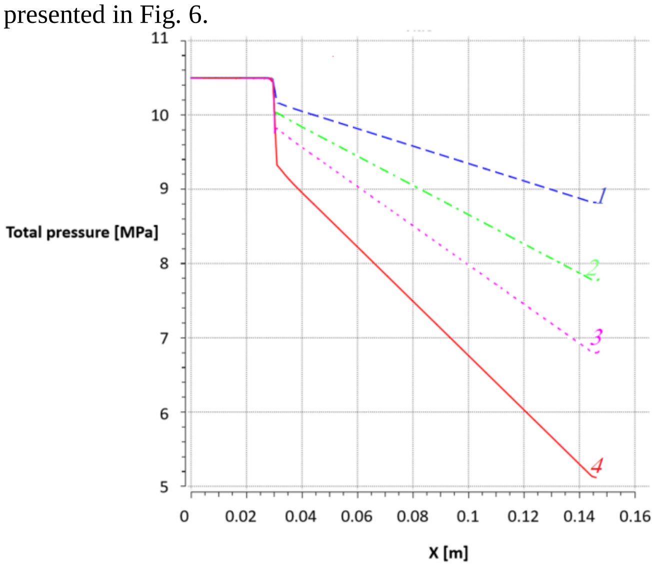


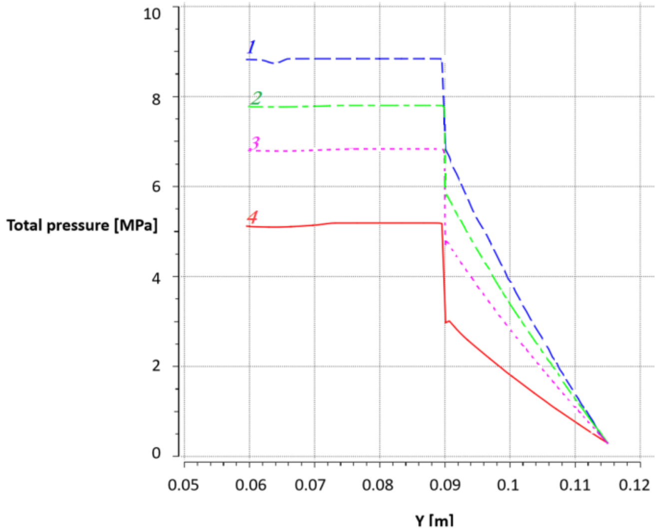
Fig. 5. Dependence of pressure in chamber p_2 on the height of the face throttle: continuous lines are calculated with Eq. 3,
“x” – the results of simulation

The numerical results of the joint solution of flow in cylindrical and face throttle in the balancing device confirm the accepted hypothesis about the constancy of pressure in the chamber.

The results of the pressure change along the length of the cylindrical and face throttle in the balancing device are presented in Fig. 6.



a)



b)
width of the face throttle:
1 6,5·10⁻⁵m 2 8·10⁻⁵m 3 9,5·10⁻⁵m 4 12·10⁻⁵m

Fig. 6. Pressure changes along the length of the cylindrical (a) and face (b) throttle in the balancing device

The pressure distribution along the length of the face throttle, obtained by numerical methods, allows the value of the axial force acting on its surface in the axial direction to be determined by numerical integration over the surfaces. Table 2 presents the axial force values obtained from the analytical solution and by numerical simulation.

TABLE 2. Values of the axial force

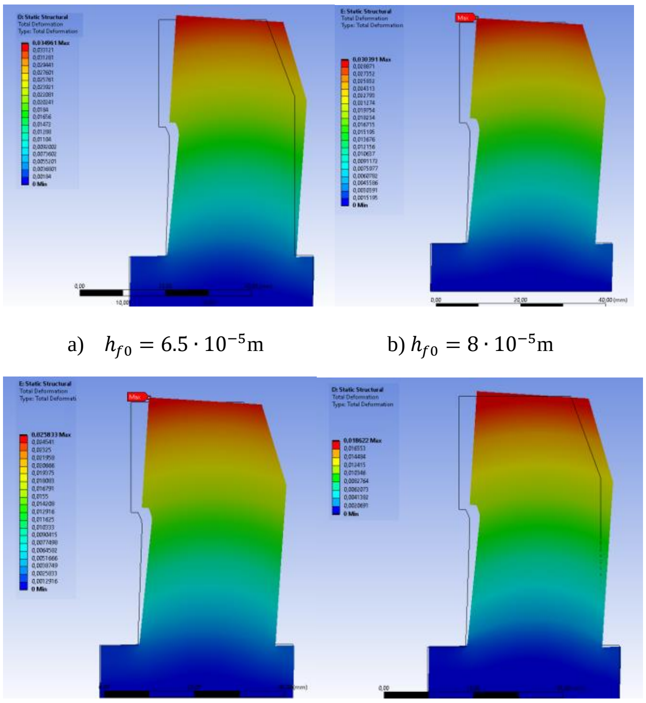
h_{f0} [m]	Analytical Approach (Eq.5), [kN]	Simulation (ANSYS) [kN]	Difference [%]
$6.5 \cdot 10^{-5}$	157.03	172.8	10.11
$8.0 \cdot 10^{-5}$	133.0	149.36	12.3
$9.5 \cdot 10^{-5}$	109.3	129.98	18.9
$12 \cdot 10^{-5}$	75.4	95.84	27,1

According to Table 2, the difference between the results of the simulation and analytical solution increases with increasing face throttle h_{f0}. The largest deviation between the analytical and numerical solution reaches 27,1% for the width of the face throttle $12 \cdot 10^{-5}$ m.

Knowing the pressure distribution in a face throttle it is possible to calculate the deformations of the balancing disc.

Balancing discs are made of 420 steel with volume hardening HRC 35...42 in some modifications of centrifugal pumps. Nowadays the balancing discs are coated with a carbide alloy cladding that provides a high sliding coefficient and high wear resistance. But not all manufacturers provide data on the material of the disc. In this work, the simulation of disc deformations was calculated for 321 steel, for which Young's modulus is $2.1 \cdot 10^{11}$ Pa, density 7850 kg/m^3, Tensile Yield Strength $2.5 \cdot 10^8$ Pa.

Results of the structural analysis are presented in Fig. 7. In this paper, the deformation of only rotating disc, which determines the geometry of the face throttle, is considered.



a) $h_{f0} = 6.5 \cdot 10^{-5}$m b) $h_{f0} = 8 \cdot 10^{-5}$m

c) $h_{f0} = 9.5 \cdot 10^{-5}$m d) $h_{f0} = 12 \cdot 10^{-5}$m

Fig. 7. Deformations of the balancing disc for different values of the height of the face throttle

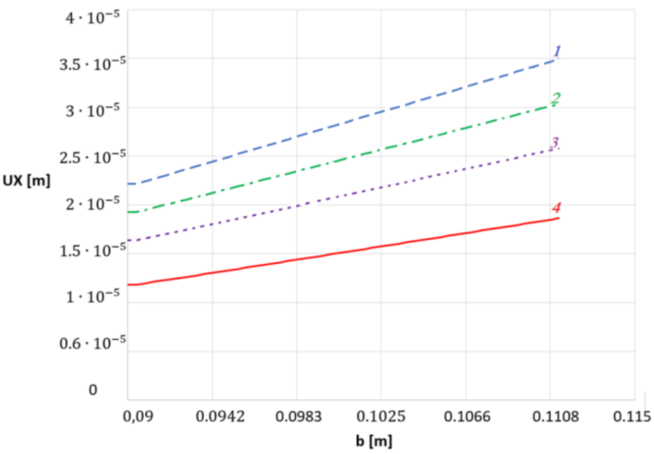
The angle of taper caused by the deformations of the balancing disc can be calculated by formula:

$$\beta = \frac{UX_3 - UX_2}{r_3 - r_2}, \tag{7}$$

where

UX_2, UX_3 – axial deformations of the face throttle surface at radii r_2 and r_3.

Dependence of the deformations on the length of the face throttle is presented in a Fig. 8.



width of the face throttle:
1 6,5·10⁻⁵m 2 8·10⁻⁵m 3 9,5·10⁻⁵m 4 12·10⁻⁵m

Fig. 8. Dependence of the deformations UX on the length b of the face throttle

Values of the angle of taper β for different values of the width of the face throttle are presented in a table 3.

TABLE 3. Values of the angle of taper β

h_{f0} [m]	Analytical Approach (Eq.6)	Simulation (ANSYS) (Eq.7)
$6.5 \cdot 10^{-5}$	$5.1 \cdot 10^{-4}$	$6.35 \cdot 10^{-4}$
$8.0 \cdot 10^{-5}$	$4.32 \cdot 10^{-4}$	$5.525 \cdot 10^{-4}$
$9.5 \cdot 10^{-5}$	$3.55 \cdot 10^{-4}$	$4.7 \cdot 10^{-4}$
$12 \cdot 10^{-5}$	$2.45 \cdot 10^{-4}$	$3.386 \cdot 10^{-4}$

The numerical results give an overestimated value compared to the analytical one. The difference varies from 25% to 35% and increases for higher values of the width of face throttle. One of the reasons for this discrepancy is the difference between the pressure values in the balancing disc (Fig. 3), obtained as a result of analytical and numerical calculations. For the same value of the face throttle, the pressure value in the chamber p_2, calculated analytically, will be smaller, which also causes smaller deformations. Another reason for the difference is the simplified analytical model of the disc adopted in the deflection calculation (Eq. 6). According to this model, a disc of constant thickness is considered; in fact, the thickness of the disc varies along the radius.

4.2 Effect of deformation of the face throttle surface on the operation of the balancing disk

The obtained pressure distributions (Fig. 6) along the length of the cylindrical and face throttles clearly show that the axial force is significantly influenced by local hydraulic resistances at the inlet of both throttles. Pressure drop at the inlet and pressure drop along the length of the cylindrical throttle depend on the resistance of the face throttle. From Fig. 6 *a* the smaller the width of the face throttle, the greater the pressure value in the chamber p_2 and the smaller the pressure drop in the cylindrical throttle. For the values of the face throttle studied, the pressure losses at the inlet to the cylindrical throttle are as follows: 0.3 MPa for $h_{f0} = 65\ \mu$m; 0.47 MPa for $h_{f0} = 80\ \mu$m; 0.73 MPa for $h_{f0} = 95\ \mu$m and 1.05 MPa for $h_{f0} = 120\ \mu$m. The maximum difference in inlet pressure drop is 3.5 times for the studied values of the face throttle width. This, in turn, leads to the different area under the pressure diagram: as bigger the value of the pressure drop at the inlet the smaller value of the force is observed. It should be noted that the pressure drop in the cylindrical throttle determines the value of the radial force, which affects the total stiffness of the rotor and its dynamic characteristics. And if the analytical model does not take into account the pressure drop at the inlet of the cylindrical throttle, this can lead to overestimated radial force values and incorrect prediction of the rotor's natural frequency.
The pressure distribution in the face throttle (Fig. 6 *b*) shows that the value of the axial force directly depends on the local hydraulic losses at the inlet of the face throttle.
According to results presented in Fig. 7 for all studied values of the average width of the face throttle, the highest deformation values are observed on the outer diameter of the balancing disc and are comparable to its average value. Deformation of the face throttle surfaces leads to the increasing of the average width and formation of the diffuser shape of the clearance. It results in smaller values of the axial force generated. For the values of the face throttle studied, the values of axial force after deformations are as follows: 156.67 kN for $h_{f0} = 65\ \mu$m; 142.2 kN for $h_{f0} = 80\ \mu$m; 125.21 kN for $h_{f0} = 95\ \mu$m and 86.4 kN for $h_{f0} = 120\ \mu$m. For the smallest and largest values of the average width of the face throttle, the change in axial force after deformation is up to 10% of the force in the undeformed state; for other width values, this deviation does not exceed 5%.
The balancing disc, in addition to the unloading function in the pump, plays the role of an end seal and determines up to 20 % of the total efficiency of the pump. Theoretically, reducing the width of the face throttle can significantly increase the overall pump performance, but in practice and according to the results obtained, large relative deformations can cause an increase in losses through the balancing device and lead to surface contact, thereby reducing pump reliability.
The presented results of the one-way FSI can be used while selecting the design parameters of the balance disc, such as the nominal face gap, expected leakage, and axial force value. However, experimental verification is necessary to confirm absolute values taking into account effects such as surface roughness or misalignments, and ensure the reliability of the design. While one-way CFD captures the main trends and parametric sensitivity, experiments provide essential validation for final design decisions.

5 CONCLUSIONS

The balancing disc, which operates not only as a support bearing but also as an end seal, is one of the most loaded and unreliable elements of a multistage centrifugal pump. Its reliability and operability are related to the noncontact mode of operation of the face surfaces. During operation, high force loads lead to deformation of the balancing disc components and, consequently, to disturbance of the flatness of the surfaces forming the face throttle. As a consequence, the load-bearing capacity is significantly reduced and there is a risk of surface contact, which can lead to failure of the pump as a whole. The developed model represents in a comprehensive way the characteristics of the system that balance the axial force of a multi-stage pump. It takes into account both pressure changes throughout the device and deformations along the length of the face throttle.
This work focuses specifically on studying the relationship between hydrodynamic characteristics and disc deformations, which allows for more accurate determination of the device's operating characteristics and a more valid choice of geometry at the design stage.
Analytical axial force calculations and FSI simulations in ANSYS were carried out for a seven-stage 180-1050 centrifugal pump and allow the following conclusions:
1) numerical results of the joint solution of flow in cylindrical and face throttle in the balancing device confirm the hypothesis about the constancy of pressure in the chamber;

2) the difference in the results of the analytical approach and numerical simulations does not exceed 10% for the chamber pressure p_2;
3) the difference in the results of numerical and analytical solution for axial force reaches 27,1% for the largest studied width of the face throttle 12 μm;
4) for a multistage pump with a traditional design of the balancing device, the change in pressure along the length of the face throttle causes a change in its shape from parallel to diffuser. As a result, the axial force in the face throttle decreases but the flow-rate increases. Under some deformations there is a risk of contact and seizure;
5) numerical results of the deformation of the surface of the face throttle make it possible to calculate the values of the angle of taper β and to determine the effect of the angle on the value of the axial force;
6) as the width of the face throttle increases and, consequently, its hydraulic resistance decreases, the pressure in the chamber of the device also decreases, which leads to smaller deformations (which are characterized by the taper angle, Table 3).
7) axial force is directly dependent on local hydraulic losses at the inlet of the throttles, and analytical models that do not take them into account should not be used in engineering practice for the design and calculation of automatic balancing discs.

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