Axial forces in multistage back-to-back pumps

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Abstract. Since the pressure of the pumps for the oil and gas industry and feed water pumps for the thermal and nuclear power plants are increasing and at the same time the production costs need to be reduced, the demands for development of the modern multistage centrifugal pumps are increased. An enlargement of stage numbers results in an increase in the distance between the bearings and as a result the shaft deflection increases. Also, a large number of stages leads to occurrence of a significant axial force acting on the pump rotor. These issues are solved by applying a back-to-back design with a throttling bush between the groups of stages. An arrangement with such throttling bush results in an additional axial force due to the different directions of leakage in the channels (side wall gaps) between the main disks of the last impellers of both stage groups and stator elements. Therefore, the design of a throttling bush with special balancing holes was proposed. This paper examines the processes occurring in the throttling bush and compares the operating parameters for two design options—without special balancing holes, and without them.

1. Introduction
To ensure reliable operation of the pump with a large number of stages, a back-to-back arrangement is applied which involves a throttling bush between two groups of stages with a pressure drop equal to half the pump head. With the standard design of the throttling bush, the leakage in the side wall gap of the main impeller disk of the last stage of the first group of stages is directed outwards, and the leakage in the side wall gap of the main disk of the impeller of the last stage of the second group of stages is directed inwards (Figure 1). In this case, an additional axial force arises due to the difference in the directions of the leakages. When the pump is operating at the different flaw rates, this additional force increases which results in installation of a significantly bigger thrust bearing.
Moreover, it should be noted that the axial shift of the rotor relative to the stator can significantly affect the axial force value [1]. The studies presented by Guilich for a stage of a multistage pump with a specific speed of \( n_q = 22 \) showed that when the impeller is shifted towards the covering disc, a width of the gap on a side of the main disc increases, recirculation appears near the main disc and the fluid rotation decreases. When the impeller is shifted towards the main disk, a width of the gap on a side of the main disk decreases, the recirculation zone is shifted towards the covering disk, and the rotation of the fluid is thus enhanced (Figure 2).

In back-to-back pumps, this effect is doubled, since if the rotor is shifted relative to the stator towards the inlet in the first group of stages, a width of the gap at the main disk increases, as for the second group of stages — this width decreases, and vice versa.
Previous studies on the analysis of processes occurring in the impeller gaps and on determining the magnitude of the resulting axial force have shown the possibility of applying the numerical modeling to estimate similar problems in back-to-back pumps [3, 4, 5].

Experimental tests of a high pressure multistage centrifugal pump, carried out at the JSC “Nasosenergomash Sumy” with measuring the value of the axial force when the rotor is shifted, showed a similar result. The rotor shift toward the pressure side resulted in a significant increase and a change in a direction of the axial force (Figure 3). The theoretical calculation according to well-known methods of the residual axial force acting on the rotor of the pump showed that the axial force value varies slightly depending on the pump flow (within 5 kN, curve 1 in Figure 3). With a slight shift of the rotor relative to the stator towards the suction side, the axial force approximately doubles with a decrease in the flow rate (curve 2 in Figure 3). When the rotor is shifted relative to the stator towards the discharge side, the axial force near the BEP zone changed insignificantly, and when the flow rate was decreased to less than 75% of the BEP, the axial force increased sharply in the opposite direction (curve 3 in Figure 3). When pumps operate at a flow rate close to the shut down, such a sharp increase in axial force can result in a failure of the thrust bearings.

The main cause affecting the magnitude of the axial force for theoretically balanced impellers is a leakage in the sidewall gaps. It is possible to minimize the effect of leakage in the sidewall gaps by establishing more severe dimensional tolerances in the manufacture of pump parts. But at the same time, the effect caused by the different directions of leakage in the impeller gaps on both sides of the central bushing remains.

To uniform the direction of leakage, the balance holes were added to the throttling bush design [2]. Due to changing pressure in the outlet chambers, the direction of leakage in the side wall gap of the main impeller disk of the second group of stages changes, and the leakage becomes directed outwards, the same as in the first group (see Figure 4).

To analyze the working process in the central bushing with balance holes, as well as to determine the magnitude of the axial force acting on the bushing and to compare with the magnitude of the axial force acting on a standard bushing, a numerical study was carried out in the back-to-back pump.
2. Description of the numerical study
A numerical study was carried out by calculating a three-dimensional flow of a viscous incompressible medium in a flow path of the pump and in the central bushing using the ANSYS CFX 19.1 software [7]. The calculation domain included an impeller, a guide vane and an annular chamber of the first group of stages, as well as an impeller, a guide vane and an annular chamber of the second group of stages, and a central bushing. The calculation domain is shown in Figure 5.

The main parameters and dimensions of the pump hydraulics elements are shown in the Table 1.
Table 1. Main parameters and geometrical dimensions.

| Parameter                                      | Value   |
|------------------------------------------------|---------|
| Rated flow, $Q_{nom}$, m$^3$/s                | 0.0875  |
| Discharge pressure, $p_{dis}$, atm             | 200     |
| Rotational speed, rpm                          | 3000    |
| Specific speed, $n_q$                          | 27.6    |
| Impeller outer diameter, $D_2$, mm             | 315     |
| Central bushing seal diameter, $D_y$, mm       | 130     |
| Central bushing seal length, $L_y$, mm         | 230     |
| Seal clearance, $\delta_y$, mm                 | 0.3     |

To perform the calculation, the computational grids were built. For the area of impellers, guide vanes and annular chambers — these were the unstructured tetrahedral grids, for the area of the central bushing — the hexahedral block-structured grids. The total number of cells in the computational grid is 2.7 million, the total number of nodes is 0.7 million.

The time steady calculation was carried out for several flow rates. The simulation of the flow in a flow path was carried out by numerically solving the system of Navier-Stokes differential equations and the continuity equation. The standard $k$-$\varepsilon$ turbulence model was used to simulate the turbulence. The conditions for entering the calculation domain were specified at the inlet to the impeller of the first group of stages. At the inlet, the value of the mass flow was set ($G = 87.2375$ kg/s). At the outlet of the first group of stages in the transfer channels, the static pressure was set ($p_1 = 106.91$ atm), which was determined as follows:

$$p_1 = p_{in1} + \rho \cdot g \cdot (i_{1gr} - 1) \cdot H_{st}$$  \hspace{1cm} (1)

where $p_{in1}$ — pump inlet pressure, Pa;
$i_{1gr}$ — number of stages in the first group, pcs;
$H_{st}$ — head of one stage, m;

At the inlet to the second group of stages, the value of the mass flow was set, which is equal to a value of the mass flow at the inlet to the calculation domain, $G_{2gr} = 87.2375$ kg/s.
At the outlet of the calculation domain, the pressure value ($p_2 = 202.52$ atm) was set, which was determined as follows:

$$p_2 = p_1 + \rho \cdot g \cdot i_{2gr} \cdot H_{st} + \rho \cdot g \cdot (i_{2gr} - 1) \cdot H_{st}$$  \hspace{1cm} (2)

where $i_{2gr}$ — number of stages in the first group, pcs
The “Frozen Rotor” type was used as an interface in the interaction of stationary and rotating elements. At solid walls, the boundary conditions are set: no-slip condition; velocity components are set zero. The walls condition was set as rough. The roughness value corresponded to the real conditions in the manufacture of a pump.

The study was carried out for two design options of the central bushing: the standard design and the design with balancing holes (Figures 6 and 7).

3. Discussion of the study results
As a result of the calculations performed, the fields of pressure and velocity distribution in the impeller gaps and the central bushing hole were obtained.

To determine the integral values, the control sections and elements were created in the Post-Processor (Figure 8).
To build the gaps pressure distribution curves, the fluid-filled area of the sealing gap was nominally divided into cylindrical surfaces with radii that are approximately at the same distance from each other. The pressure values were obtained on these surfaces. To determine the axial force, we integrated the product of pressure and the throat area $df$.

Figure 9 shows the gap pressure distribution curves from the side of the main disks of the impellers of the first and second groups of stages for the design option of bushing with balancing holes. The pressure magnitudes correspond to the actual pressure distribution in a fifteen-stage pump. It should be noted that the distribution looks more or less uniform, and slightly depends on the operating mode of a pump.

As for the standard bushing design (without holes), the distribution of pressure in the gap is completely different (Figure 10). The pressure drop is different in the first and second group of stages. At the inlet to the hole from the gap of the impeller of the first group, a significant pressure drop occurs followed by a gradual growth toward the periphery. In the second group of stages, the pressure
is slightly changed in the outward direction. In general, we can observe an unequal pressure distribution in the gap of the first and second groups of stages.

![Graph](image)

**Figure 10.** The gap pressure distribution curves for the first (a) and second (b) groups of stages for the standard bushing design (without balancing holes).

Figures 11 and 12 show the flow direction lines in the impeller sidewall gaps and in the central bushing hole. As it can be seen, in the case of a central bushing with balancing holes, the flow in the sidewall gaps of impeller of the first and second groups is directed outwards. The flow split occurs in a hole of the central bushing.

![Diagram](image)

**Figure 11.** Contour curve of the flow for the design option of bushing with balancing holes.

As for the standard bushing design without balancing holes, the flow in the gap of the impeller of the first group is directed outwards, and the flow in the gap of the impeller of the second group is directed inwards. In the hole itself, the flow is directed strictly from the impeller of the second group to the impeller of the first group. Thus, it is confirmed that the presence of balancing holes makes it possible to unify the direction of flow in the gaps of the impellers of both groups of stages.
4. Conclusions

As a result of the study, the following conclusions can be drawn.

In the multistage pumps, including the BB5 pumps according to API 610 [6], despite the apparent theoretical balancing of the axial force, a significant effect is caused by the conditions of fluid flow in the impeller sidewall gaps which depends on the fluid flow direction and the size of the impeller sidewall gaps.

The conducted numerical research of various design options of the central balancing bushing indicates a more even pressure distribution in the impeller sidewall gaps in the design with balancing holes due to unification of the processes occurring in the sidewall gaps of the last impellers of the back-to-back pumps.

When designing and manufacturing the BB4 and BB5 back-to-back pumps, a special attention should be paid to the axial position of the rotor relative to the stator, since even a slight shift can cause an abrupt increase in the resulting axial force, which in its turn can damage the thrust bearing.

Experimental and numerical studies have shown the need for additional research of the various options for uneven impeller sidewall gaps and its designs.

References

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