INTRODUCTION

The article shows the results of a CFD analysis of several types of balancing devices for semi-open radial impellers. The objective of the study is to determine the balancing device which provides an acceptable level of axial force and ensures minimal hydraulic losses for the preliminary designed impeller.

IMPELLER GEOMETRY

The main dimensions of the impeller and blade shape were initially designed for the following parameters:

Table 1. Impeller design parameters

<table>
<thead>
<tr>
<th>Flow Q, m³/h</th>
<th>Head H, m</th>
<th>Speed n, rpm</th>
<th>Specific speed nq</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>32</td>
<td>2900</td>
<td>36</td>
</tr>
</tbody>
</table>

The calculations and design methods for the impeller main dimensions, blade angles and shape are not discussed as part of this article. The values are given for information in Table 2 and Fig. 1.

Table 2. The impeller main dimensions and blade angles

<table>
<thead>
<tr>
<th>Outlet diameter, D₂, mm</th>
<th>Outlet width, b₂, mm</th>
<th>Inlet diameter, D₁, mm</th>
<th>Outlet angle, β₂, deg</th>
<th>Blade wrap, φ, deg</th>
<th>Number of blades, m</th>
<th>Max. allowable front clearance, Cᵣ, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>178</td>
<td>22.5</td>
<td>100</td>
<td>22.5</td>
<td>146</td>
<td>5</td>
<td>0.7</td>
</tr>
</tbody>
</table>

Fig. 1. Impeller main dimensions and blade angles

CALCULATION METHOD AND NUMERICAL MODEL

The calculations are performed assuming a steady-state using the continuity equation and the three-dimensional, time-averaged Navier-Stokes equations for incompressible flow. Eddy viscosity is determined by the k-ω based Shear Stress Transport Model. Boundary conditions are averaged. The walls are modeled without the wall function (y⁺<1).
The model consists of one rotating domain (one blade sector) without volute or any stationary devices upstream and downstream of the impeller. (see Fig. 2). Inlet boundary conditions are: 1 atm. total pressure; normal velocity direction; 5% turbulence intensity. The outlet boundary condition is: 27.8 kg/s mass flow. All walls are without slippage, and also the two casing walls have counter rotating conditions. The meshes of periodic regions match together and have 1:1 interface. The liquid is water at 20 deg C.

![Fig. 2. Boundary conditions and 3D-model of periodic domain (impeller without balancing device, with smooth rear side of the hub)](image)

Unstructured mesh (see Fig. 3) consists of ~12.5 million tetrahedral elements in the flow core and prism elements for boundary layer discretization. The number of nodes is ~5.5 million. Maximum mesh element size is 2.4 mm (for flow core), minimum mesh element size is 0.12 mm (in the front and back clearance area). The thickness of the first prism layer ensures $y^+<1$ for all the walls including clearance areas.

![Fig. 3. Unstructured tetra mesh](image)

**CALCULATION RESULTS**

As the first step of the study, flow dynamics for an impeller without any thrust balancing is calculated using the computational model described above. The calculation results are velocity and pressure distributions for impeller flow path and sidewall gap. Visualization of the results (static pressure distribution, streamlines and meridional velocity distribution in the impeller flow path and sidewall gap) is shown in Fig. 4 and Fig. 5. The obtained flow structure in the sidewall gap is typical for a flow between static
and rotating smooth discs as reported in [1], [2] and other research papers. The sidewall gap is closed at the bottom, so there is no leakage there.

**Fig. 4. Visualization of static pressure distribution**

**Fig. 5. The impeller streamlines along blade and sidewall gap meridional velocity distribution**

**AXIAL FORCE CALCULATION METHOD**

For the current study the impeller axial force ($\overrightarrow{F_{AX}}$) is calculated automatically (using results of CFD) and can be approximately represented as the vector sum of pressure forces on front and rear impeller surfaces:

$$\overrightarrow{F_{AX}} = \overrightarrow{F_F} + \overrightarrow{F_R} + \overrightarrow{F_{ATM}}$$  \hspace{1cm} (1)

The values of these pressure forces ($|\overrightarrow{F_F}|$, $|\overrightarrow{F_R}|$) are the sums of axial forces acting on the surfaces of each impeller mesh cell and can be determined using the following equations:

$$|\overrightarrow{F_F}| = \sum_{i=1}^{n} P_i S_i$$  \hspace{1cm} (2)

$$|\overrightarrow{F_R}| = \sum_{j=1}^{m} P_j S_j$$  \hspace{1cm} (3)
$P$ – static pressure on mesh cell; $S$ – area of mesh cell projection on orthogonal (to the rotation axis) plane; $n, m$ – number of cells on impeller front and rear surfaces.

$\overrightarrow{F_{ATM}}$ is a force acting on the shaft from the atmosphere, see Eq. (4):

$$|\overrightarrow{F_{ATM}}| = P_{ATM} \pi D_S^2 / 4$$  \hspace{1cm} (4)

$P_{ATM}$ – atmospheric pressure; $D_S$ – seal diameter (see Fig. 1).

The calculated axial force for the impeller without thrust balancing $\overrightarrow{F_{ax}} = 2597 \, N$. Generally, this value is approximate and can be used only for comparison of several types of balancing devices which are calculated using the same method. To get a more accurate value, an unsteady calculation of the whole pump (including full impeller and volute) is required, so far as the volute has a serious effect on the impeller pressure distribution, including leading edges of the impeller blades and sidewall gaps. The research of this unsteady process is reported in [3].

The second parameter to be compared is the impeller hydraulic efficiency ($Eff$). For the current impeller: $Eff = 0.840$. This value is approximate as well, because it is based on the torque of the impeller and hydraulic losses which are calculated using RANS with eddy viscosity and the semi-empirical turbulence model.

**RADIUS AVERAGING METHOD FOR STATIC PRESSURE DISTRIBUTION**

For better visualization of calculation results, the resulting static pressure distribution needs to be processed. All the cells on the rear and front surfaces are grouped by radius (the step is 2 mm); averaged static pressure is calculated for every range of radius (so that rings of average values are estimated). Averaged static pressure distribution diagrams on front (black) and rear (red) surfaces of the hub without thrust balancing are shown in Fig. 6. The shapes of the resulting diagrams (linear and parabolic) are typical for a disc with blades and for a rotating smooth disc.

![Fig. 6. Radius averaged static pressure distribution diagram](image)

**GEOMETRY OF BALANCING DEVICES**

There is a variety of impeller balancing devices, but only four are examined in the next stage of this study. Calculation method, meshes and boundary conditions are the same. The geometries of these balancing devices are shown below (Table 3).
<table>
<thead>
<tr>
<th>#</th>
<th>Description</th>
<th>Sketch</th>
<th>3D-model</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Impeller without any thrust balancing, with smooth rear surface. The width</td>
<td><img src="https://via.placeholder.com/150" alt="Sketch" /> <img src="https://via.placeholder.com/150" alt="3D-model" /></td>
<td></td>
</tr>
<tr>
<td></td>
<td>of the sidewall gap is 2.3 mm.</td>
<td><img src="https://via.placeholder.com/150" alt="Sketch" /> <img src="https://via.placeholder.com/150" alt="3D-model" /></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Open impeller with cut-outs in the hub. Clearance between the hubside of the</td>
<td><img src="https://via.placeholder.com/150" alt="Sketch" /> <img src="https://via.placeholder.com/150" alt="3D-model" /></td>
<td></td>
</tr>
<tr>
<td></td>
<td>blades and the casing is 1.3 mm.</td>
<td><img src="https://via.placeholder.com/150" alt="Sketch" /> <img src="https://via.placeholder.com/150" alt="3D-model" /></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Impeller with short rear radial blades and a collar above the blades for</td>
<td><img src="https://via.placeholder.com/150" alt="Sketch" /> <img src="https://via.placeholder.com/150" alt="3D-model" /></td>
<td></td>
</tr>
<tr>
<td></td>
<td>protecting the structure of the flow at impeller outlet. The clearance</td>
<td><img src="https://via.placeholder.com/150" alt="Sketch" /> <img src="https://via.placeholder.com/150" alt="3D-model" /></td>
<td></td>
</tr>
<tr>
<td></td>
<td>between the rear blades and the casing is 1.3 mm. The sidewall gap width</td>
<td><img src="https://via.placeholder.com/150" alt="Sketch" /> <img src="https://via.placeholder.com/150" alt="3D-model" /></td>
<td></td>
</tr>
<tr>
<td></td>
<td>is 2.3 mm. The radial clearance between the collar and the casing is 1</td>
<td><img src="https://via.placeholder.com/150" alt="Sketch" /> <img src="https://via.placeholder.com/150" alt="3D-model" /></td>
<td></td>
</tr>
<tr>
<td></td>
<td>mm. The number of rear blades is 10.</td>
<td><img src="https://via.placeholder.com/150" alt="Sketch" /> <img src="https://via.placeholder.com/150" alt="3D-model" /></td>
<td></td>
</tr>
</tbody>
</table>
4 Impeller with rear radial blades and a collar above the blades for protecting the structure of the flow at impeller outlet. The clearance between rear blades and the casing is 1.3 mm. The sidewall gap width is 2.3 mm. The radial clearance between the collar and the casing is 1 mm. The number of rear blades is 10.

5 Impeller with rear profiled blades. The height of the rear blades is 3 mm. The clearance between the rear blades and the casing is 1.3 mm. The number of rear blades is 5.

It is important to note, that the collars (#3, #4) above the radial rear blades are used only for prevention of mixing of the flows from the front blades and from rear radial blades. They do not work as a groove seal / wear ring (the radial clearance is 1 mm). Without the collar, large vortex structures will develop which would have an adverse effect on velocity distribution at the outlet of impeller.

The balancing effect of the rear blades heavily depends on the axial clearance between the rear blades and casing cover. The same value of axial clearance (1.3 mm) is used for all the types of rear blades (see Table 3). The effect of the clearance size on the axial force requires separate research and is not considered as part of this investigation.

**BALANCING DEVICES CALCULATION RESULTS AND COMPARISON**

Processed calculation results (streamlines, visualization of sidewall gap velocity and pressure distribution, diagrams of circle averaged pressure distribution) for every balancing device from Table 3, are listed below (Table 4, Fig. 7). Diagrams of circle averaged pressure distribution on the front surfaces of the impellers are not shown in Fig. 7, since, generally, they are similar for all the calculated impellers (see Fig. 6).

A rotational coordinate system is used for all figures with velocity distributions and streamlines. Pictured vectors are relative velocity projections on an orthogonal plane (to the axis of rotation) located in the middle of the sidewall gap or in the middle of rear
blades depth along the axis of rotation (except #1). It should be noted that the front blades disarrange the flow periodic structure in the sidewall gap.

Table 4. Visualization of the calculation results.

1. The results for an impeller without thrust balancing are shown in Fig. 4, Fig. 5, Fig. 6

2. Open impeller with cut-outs in the hub.
3. Impeller with short rear radial blades

4. Impeller with rear radial blades
5. Impeller with rear profiled blades
1 - Impeller with a smooth rear surface of the hub; 2 - Impeller with cut-outs in the hub; 3 - Impeller with short rear radial blades; 4 - Impeller with rear radial blades; 5 - Impeller with rear profiled blades.

Fig. 7. Diagram of circle averaged pressure distribution on the rear surface of the impeller disc

Pictures in the table 4 (#3, #4) and diagrams (Fig. 7) show, that the collars provide very effective protection for flow structure above the rear blades. There are two separate flows above and below the collar, and these flows have different rotational speed and different pressure distribution laws. Hydrodynamic theory and mathematical descriptions of velocity and pressure distribution in sidewall gaps are not described as part of this article. See [1], [2] and other research papers.

Table 5. Calculated axial force and efficiency

<table>
<thead>
<tr>
<th>#</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cal. axial force, N</td>
<td>1637</td>
<td>2104</td>
<td>747</td>
<td>1060</td>
</tr>
<tr>
<td>Efficiency reduction, %</td>
<td>2.8</td>
<td>0.4</td>
<td>1.4</td>
<td>1</td>
</tr>
</tbody>
</table>

Obviously, impeller #1 (without balancing) has the highest hydraulic efficiency (~84%) and the highest axial force (2597 N). All the other calculation results are compared with impeller #1 (see table 5).

Making cut-outs in the hub is one of the most popular balancing methods for semi-open impellers. They reduce the surface area of the hub affected by the pressure. The cut-outs provide ~40% (see Table 5) axial force reduction in comparison with impeller #1. On the other hand, about ~3% of hydraulic efficiency reduction can be seen as result of hydraulic losses. Because of the sudden expansion in impeller flow path, there is a wide vortex region (see #2, Table 4) in the outlet of impeller. This vortex region causes high hydraulic losses and efficiency reduction.

Short rear radial blades (impeller #3) provide only ~20% the axial force reduction, which is usually not enough for high-power pumps. Extending the rear radial blades closer to the outside diameter of the impeller significantly decreases the axial force (see Table 5). This balancing device provides an axial force reduction of ~75%. In addition, long rear blades cause ~1.5% hydraulic losses. Profiled long rear blades (impeller #5) reduce the axial force by ~60% and have only ~1% efficiency losses.
CONCLUSIONS

Both options (#4, #5) reduce the axial force for the current pump to acceptable levels. Additional study and hydraulic design works on the rear blades can be performed.

All these calculation results require experimental confirmation, and they are only valid for impellers with similar specific speed and size. Further research and experimentation are required for different specific speeds, types of impellers and balancing devices.

REFERENCES

