Numerical simulation and optimization of centrifugal compressor return channels: methods and results

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Abstract. The authors actively use their engineering Universal Modelling Method for gas-dynamic design of centrifugal compressors for industrial partners. Currently, there are two approaches to improving the Method: improving the preliminary design and increasing the accuracy of gas-dynamic characteristics calculation. Computational Fluid Dynamics (CFD) methods give good results for the flow path stator part. Recently, massive CFD calculations of the vaneless diffusers (VLD) characteristics were generalized by a system of algebraic equations, which replaced the previous more complex mathematical model in the Method. Then, based on CFD optimization of a large series of return channels (RCh), corrections were made to the preliminary design of this element of the flow path. This paper presents the results of the joint CFD optimization of a vaneless diffuser and a return channel. Stator elements have many geometric parameters. For a stage with the flow rate coefficient of 0.0597 and the loading factor of 0.60, only the VLD relative radial length $D_4/D_2$, the number and the inlet angle of the return channel vanes were optimized. Engineering calculations and design experience predicted an optimal value of $D_4/D_2$ within the interval 1.9 - 2.0. CFD optimization demonstrated almost linear reduction of the total head loss towards the end of the investigated range $D_4/D_2 = 2.3$. After careful optimization of the U-bend, the optimization of $D_4/D_2$ was repeated. The influence of on the loss coefficient has decreased, but the value of $D_4/D_2 = 2.3$ is still far from optimum. It significantly exceeds technically acceptable radial size. The result obtained influenced the design plan of future calculation experiments with a large series of stator elements of the stages in a practically significant range of design parameters.

1. Introduction
The total capacity of industrial centrifugal compressors in developed countries reaches dozens of GW. It is important to create a compressor flow path with minimal energy consumption. The development
of CFD technologies formally solves both the direct and inverse problems of gas dynamics. The problem is that CFD calculation of gas dynamic characteristics of a compressor does not guarantee their compliance with actual characteristics. The authors of [1 - 4] presented data on good agreement of CFD-calculated characteristics with test results. Other publications [5 - 11] and own experience of the authors of this paper [12 - 14] show that the characteristics of centrifugal compressors according to CFD calculations do not correspond to test results.

The authors believe that CFD technologies are currently effective as an auxiliary tool, but not as a primary design tool. The final goal of the authors of this paper is to optimize and calculate the stator elements of centrifugal stages in the range of design parameters $\Phi_{des}$ (flow rate coefficient), $\psi_{T,des}$ (loading factor). This will make it possible to formulate design recommendations and obtain dataset from a computational experiment for creating a mathematical model. The first stage of the research presented below made it possible to work out the optimization technique and evaluate the possible results. The authors optimized the stator elements of a stage with design parameters $\Phi_{des} = 0.0597$ and head $\psi_{T,des} = 0.60$.

2. State of the art
In recent years among researchers from different countries, CFD calculations of return channels have become widespread. Of particular interest are the papers in which the calculated results are compared with experimental data. In [15 - 19], various aspects of CFD modeling of a high-flow-rate stage with $\Phi_{des} = 0.15$ and its return channels are analyzed. The authors have a modern test rig at their disposal, which made it possible to compare the stage characteristics and structure of the flow. Satisfactory compliance of the stage characteristics was obtained, and good results of comparison of the flow structure were attained as well. CFD calculations, in particular, correctly estimated the efficiency of stage variants with different radial lengths of a vaneless diffuser. The authors of [20] experimentally confirmed the CFD calculations of the RCh with additional vanes at the outlet and improved the RCh characteristic at $\Phi > \Phi_{des}$. The authors of [21] by CFD-optimization of RCh increased the efficiency of a two-stage compressor by 0.7% (experimental confirmation). RCh with a vaneless diffuser is optimized by many parameters.

Computational gas dynamics methods are able to improve stator elements of the flow path of centrifugal compressors. The authors of [22, 23] designed and investigated return channels of stages with design parameters $\Phi_{des} = 0.015–0.15$ and $\psi_{T,des} = 0.45–0.70$. The Direct Optimization program optimizes the main dimensions of RCh. The RCh with a constant vane height $b_2 = b_0$ and a smaller number of vanes in comparison with the preliminary design turned out to be the optimal ones. The loss coefficients of optimized RCh are less by up to 30%. Corresponding adjustments were made to the preliminary design methodology.

The main conclusion that follows from the publications of recent years is that the results of CFD calculations of return channels and vaneless diffusers are confirmed experimentally.

3. The objectives of the research
Universal Modeling Method [24, 25] makes the initial design of the VLD + RCh. Figure 1 shows the configuration.
This engineering tool is in design practice since mid-1990's. More than 400 centrifugal compressors with unit capacity of up to 25 MW designed using the Method effectively operate in different industries. The efficiency of best multistage compressors was more than 87% [20], which indicates good aerodynamic characteristics of an initial design of vaneless diffusers and return channels. To improve this flow path elements would be complicated but important.

The meridional form of VLD and RCh is circular arcs and straight lines. The choice of all dimensions is described in [25]. The new recommendation follows of the research presented in [21]. For 15 stages with design parameters \( \Phi_{des} = 0.015–0.15, \psi_{T,des} = 0.45 – 0.70 \), the optimal configuration is \( \bar{b}_h = \bar{b}_b \). It is accepted for return channels under investigation. In the meridional plane, the optimization parameters are:

- the relative radial length of the VLD and RCh: \( \bar{D}_s = \bar{D}_b \);
- the relative radii \( \bar{R}_s, \bar{R}_b \) that determine the friction losses in the U-bend and the local velocity gradients.

In the radial plane, the inlet angle of the vanes \( \alpha_{v,5} \) is subject to optimization to ensure a favorable flow inlet. The number of vanes \( z \) is optimized to minimize the sum of friction and separation losses.

Stator elements of the stage with design parameters \( \Phi_{des} = 0.0597, \psi_{T,des} = 0.60, \bar{D}_b = 0.35 \) were chosen as a specific object of optimization. Basic dimensions and parameters of stator elements according to preliminary design: \( \bar{b}_s = 0.039, \bar{D}_s = 1.73; \bar{b}_b = \bar{b}_b = 0.085, z = 22; \alpha_{v,5} = 14.65^\circ; \alpha_{2,\ldots,des} = 30.8^\circ, \zeta_{SE} = 0.1988 \).

4. **Influence of the radial length on the loss coefficient of the SE**

The Workbench of the ANSYS 19 software package was used. The parameterized model of the flow path was made in DesignModeler. In the TurboGrid mesh generator, design meshes for the main and splitter vanes were built. The values \( y + <20 \) were set in accordance with the requirements of the correct modeling of the boundary layer using the SST turbulence model (Shear-Stress-Transport). The total number of elements is 893,000. The loss coefficient \( \zeta_{SE} \) was the object for optimization in the
Direct Optimization program. An optimization method depends on the number of objectives, the set limits and the desired number of design points. For the research conducted in the work, the MOGA (Multi-Objective Genetic Algorithm) method was used.

The experience of RCh optimization in [20, 21] pointed out certain difficulties of multiparameter optimization. CFD calculations have a certain margin of error, so to find a truly optimal variant requires significant resources. Optimal solutions for return channels in [21] were obtained by optimizing two or three geometric parameters, and only then the others. Accordingly, the diameters $D_4 = D_5$, the number of vanes $z$, and the inlet angle of the vanes $\alpha_{s5}$ were optimized. The range of variation was chosen to be $D_4 = D_5 = 1.40–2.14$. Figure 2 illustrates the influence of optimization parameters on the SE loss coefficient:

![Figure 2. Influence of optimization parameters on the SE loss coefficient](image)

Beyond expectations, the optimum of $D_4$ was not found in the selected range. The loss coefficient decreases monotonously with increasing of $D_4$. For investigation of the optimized stator elements operation in off-design modes, the loss coefficient and the outlet flow angle characteristics was calculated. Figure 3 shows these characteristics for the initial variant of the SE and the variant with $D_4 = 1.9$. Hardly bigger $D_4$ could be in real compressors.
Figure 3. Loss coefficients of SE (above) and outlet flow angle (below).
Stage 2DI-0.0579-0.60-0.35, $\bar{D}_4 = 1.73$ and 1.9

The range of $\bar{D}_4$ was extended up to 2.3. The loss coefficient diminished again. Figure 4 shows the dependence of the loss coefficient (design flow rate) on the radial size of the stator elements.

The linear dependence $\zeta_{SE} = f(\bar{D}_4)$ indicates the remoteness of the optimum. At the same time, the value $\bar{D}_4 = 2.3$ is far beyond the constructive limitation acceptable for real compressors.

Table 1 shows a comparison of loss coefficients and efficiency for SE with $\bar{D}_4 = 1.73, 1.90, 2.20$.

| $\bar{D}_4$ | $\zeta_{SE}$ | $\zeta_{RCh}$ | $\eta_{SE}$ | $\Delta\eta_{SE}$ |
|------------|-------------|--------------|-------------|-----------------|
| 1.73       | 0.1988      | 0.1907       | 0.772       | 0.0808          |
| 1.90       | 0.1869      | 0.2376       | 0.7857      | 0.076           |
| 2.20       | 0.1758      | 0.2911       | 0.7987      | 0.0715          |
Figure 4. The minimized loss coefficient of SE, depending on \( \bar{D}_4 \).

The stage 2DI-0.0579-0.60-0.35 before the U-bend optimization

The stator element polytrophic efficiency is:

\[
\eta_{SE} = \frac{\lg(p_{\text{U}} / p_s)}{\kappa - 1} \frac{\lg(T_0 / T_s)}{\frac{T_0}{T_s} - 1}
\]  

(3)

The loss of efficiency of a stage in the SE is:

\[
\Delta \eta_{SE} = \frac{\zeta_{SE}}{2 \psi_T} \pi_{2-3}^2 = \frac{\zeta_{SE}}{2} \psi_T \frac{\psi_T}{\cos \alpha_{2-3}}
\]  

(4)

The radial length \( \bar{D}_4 \) influence obvious: the losses in the longer RCh are greater. However, VLD+RCh of greater radial length are more effective due to flow deceleration in longer VLD. Flow structure in the radial plane does not reveal principal differences between the original SE variant and the variant with a larger radial dimension \( \bar{D}_3 = \bar{D}_4 \) - Figure 5.

The flow pattern in Figure 6 points to a possible reason for the unexpected monotonous influence of \( \bar{D}_4 \) on the loss coefficient. A significant flow separation zone occurs in the U-bend. In this case, the pressure loss depends on the kinetic energy at the inlet to the U-bend. The smaller the radial length of the VLD, the greater the kinetic energy. Obviously, the size of the U-bend needs optimization.
Figure 5. SE of stage 2DI-0.0579-0.60-0.35. Streamlines at the vane-to-vane mean surface.

5. Optimization of the U-bend of SE with $D_4 = 1.9$

In [23] is recommended to correlate the optimal value of the radius of curvature of the channel with its width, i.e. $R_y/b_y$, $R_n/b_y$. The optimization was carried out for the inner radius of curvature $R_s$, auxiliary parameter $a$ connected with $R_y/b_4$, the number of vanes $z$, the inlet vane angle $\alpha_s$. The outer radius of curvature $R_n = f(a,R_s)$ was calculated by the formula $R_n = a(b_4 + b_5 + 2R_y)/2$. For $a = 1$, the outer contour is described by one radius $R_n$, as in the case of the U-bend in the section 5. In accordance with the chosen MOGA method, the optimization has converged when 118 SE variants were calculated. The parameters of the initial and the optimized SE variants are presented in Table 2.
Table 2. Parameters of the initial and best optimized SE variants

| №  | $\alpha_5$, deg | $z$ | $R_i/b_1$ | $R_i/b_2$ | $\zeta_{SE}$ | $\alpha_0'$, deg | $\Delta c_{av}$ |
|----|-----------------|----|----------|----------|-------------|-----------------|----------------|
| 118| 14.00           | 18 | 1.90     | 2.82     | 0.1759      | 88.50           | 0.297          |
| Initial | 13.25 | 18 | 1.23     | 2.49     | 0.1869      | 88.92           | 0.230          |

Decrease of the loss coefficient is associated with better organization of the flow in the meridional plane, Figure 7. The maximum radial size of the SE with an optimized U-bend is slightly larger. The SE loss coefficient decreased by 6.4%. The loss of efficiency in the SE decreased by 0.47%; such a saving deserves attention.

Figure 7. On the left is the U-bend shape. Preliminary design is shown in gray, the design after optimization in red. In the center are streamlines of the original version, on the right are streamlines after U-bend optimization

6. Influence of the radial length on the loss coefficient of the SE after U-bend optimization

With optimized values of $R_i/b_1$ and $R_i/b_2$, the number and inlet angle of the SE vanes with $\bar{D}_4 = 1.73$ and 2.3 were optimized. The separation zones in the meridional plane have practically disappeared. The U-bend with $R_i/b_1 = 1.90$ and $R_i/b_2 = 2.82$ can be considered optimal for all investigated $\bar{D}_4$. Table 3 shows the parameters of the SE with the optimal U-bend at $\bar{D}_4 = 1.73$, 1.9 and 2.3 and with the U-bend from after the preliminary design (ORG):

Table 3. Parameters of the SE with the optimal U-bend and with the U-bend after the preliminary design

| №  | $\bar{D}_4$ | $z$ | $\alpha_5$, deg | $\zeta_{SE}$ | $\Delta \eta_{SE}$ | $\zeta_{SE}$ (ORG) | $\Delta \eta_{SE}$ (ORG) |
|----|-----------|----|-----------------|-------------|-----------------|-----------------|-----------------|
| 1.73 | 22       | 14.65     | 0.1841          | 0.0749      | 0.1988          | 0.0808          |
| 1.9 | 18       | 13.25     | 0.1769 (+0.3%)  | 0.0719      | 0.1869          | 0.076 (+0.48%)  |
In rows 4 and 6, the increment in the stage efficiency due to the increase in the radial size of the SE is given in brackets. With an optimal U-bend, the efficiency gain is smaller. However, the growth rate is still significant. With a lot of design practice [24, 25] the authors were unable to use vaneless diffusers with $\bar{D}_4 > 1.86$. Anyway the result in Table 3 deserves attention.

Conclusion

The authors propose in the near future to continue the search for the optimal dimensions of the stator elements of stages with vaneless diffusers in the practically significant range of design parameters $\Phi_{des} = 0.015–0.15$, $\psi_{r,des} = 0.45–0.70$. From the study of the stage with parameters $\Phi_{des} = 0.0597$, $\psi_{r,des} = 0.60$ it follows that the optimal radial size $\bar{D}_4$ for high-flow stages can be much larger than the constructively acceptable values. In this case, the geometric parameter $\bar{D}_4$ cannot be optimized. It is necessary to optimize stator elements with several fixed values of $\bar{D}_4$ in a constructively acceptable and practically significant range. Before doing this work, it is necessary to pay attention to the role of the return channel geometry parameters that were not studied in the present work. For example, high-flow stages may require the 3D shape of vanes. It is also necessary to study profiles and midline shape of vanes.

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