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Numerical Simulation Of Gas-dynamic Characteristics Of The Semi-Open 3D Impellers Of The Two-Element Centrifugal Compressors Stages

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Abstract. In the course of the work, the results of estimating the error in the numerical calculation of two-element 3D impeller stages of centrifugal compressors were obtained. Gas dynamic characteristics of 6 computational models of two-element stages of a centrifugal compressor are simulated. Based on available experimental data, models of validation characteristics have been performed. A parametric model has been developed for carrying out multi-criteria and multi-parametric studies.

INTRODUCTION

When designing new or upgrading industrial and transport centrifugal compressors, the tasks of ensuring the main parameters are solved: volumetric productivity, pressure, a wide working area, as well as the requirements for the energy efficiency of the installation, while taking into account the constructive restrictions imposed by consumers on the size of the housing, the location of the pipes, the type of drive, etc. Such restrictions affect the choice of design and gas-dynamic parameters of the flow part, which cause deviation of the parameters from their optimal values. To solve this problem in the process of designing the flow part, a variant calculation-theoretical analysis is carried out.

Definitely, the process of improving the flow part to achieve a given efficiency takes considerable time and financial resources. Energy efficiency improvement is achieved through gas-dynamic improvement of the flow part, selection of optimal ratios of gas-dynamic parameters, qualitative profiling of the blades, the use of modern 3D impellers with a spatial shape of the blades, providing pressure and flow rate more at a relatively high efficiency. Previously, experimental refinement of the flowing part had the greatest importance in this process, but with the accumulation of a base of experimental data, the development of numerical methods, design techniques, it became possible to reduce the share of the experiment, and sometimes completely abandon it.

Numerical modeling, for the time being, plays a major role in the design of new projects, for which it is important to obtain the highest efficiency. A special role is played by the estimation of the error in numerical simulation, since semi-empirical turbulence models embedded in computational gas dynamics methods are based on relatively simple (classical) flows and allow one to obtain an average "picture" for the entire set of flows.

Thus, it is necessary to know under what conditions and parameters of the stage of a centrifugal compressor the smallest error is observed, and for what is the greatest for timely recording. Therefore, a special role is given to the validation [4,5] of the numerical gas dynamic characteristics of the step elements with experimental data.

Supercomputer technology, methods of computational gas dynamics and methods of mathematical programming allow solving multicriterion and multiparametric optimization problems for increasing the energy efficiency of the flowing part of centrifugal compressors. Examples of such works are [1,6,7]. To solve such problems, it is necessary to use parametrized models, whose geometric parameters are assigned variable names and adjust the relationships between them, which in its own way is quite a difficult task due to the presence of a number of interrelated factors, so the model often has to be simplified.

PURPOSE OF WORK

The purpose of the work: to simulate the characteristics of the two-element stages of centrifugal compressors with 3D impellers on the basis of computational gas dynamics methods and to make a comparison with the results of experimental studies to determine quantitative and qualitative error indices; to develop a solid-state three-dimensional parametric model of the flow part of an 3D impeller in the two-element stage of a centrifugal compressor. The end result of the work is the development of a verification database with corresponding performance indicators and deviations from the calculation of experimental data.

OBJECT OF RESEARCH

The object of the study are model two-element stages of centrifugal compressors with an axial wheel (Fig. 1). The design of a typical two-element stage centrifugal compressor consists of the following elements: inlet suction chamber, impeller, vaneless (VLD) or vane diffuser (VD). By its principle of operation, the stage are of the end type. Stage is made with an inter-disk clearance at the main disk and a labyrinth seal between the housing and the rotating rotor. A labyrinth seal is connected with the atmosphere and operates as a discharge of the piston – dummies. This type of stages was originally designed for general purpose compressors, turbochargers, refrigeration turbochargers, etc. The peculiarity of this impeller is that the blades of finite thickness are used, consisting of a radial part made of a radius and a rotating guide device. The change in blade height at the outlet was achieved due to the axial shift of the tire line of the stand. The input rotating guide device for unifying the tests remained unchanged. The impellers of the series RC-6, 5 refer to a semi-open type with a gap between the housing and the ends of the spatial blades. The objects and experimental data of the RK-6,5 series were obtained by the scientific group of Prof. A.M. Simonov at the department "Compressor, Vacuum and Refrigeration", there are characteristics of full-scale tests on the open circuit [2]. Table 1 summarizes the main parameters of research objects.

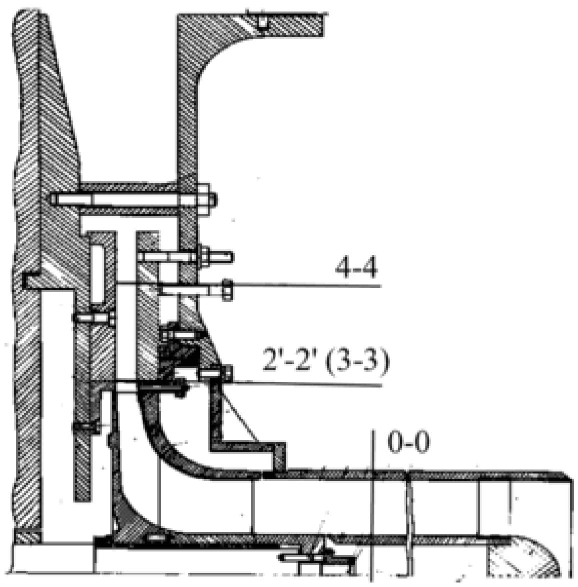

FIGURE 1. The scheme of the experimental model stage

Conditional flow rate coefficient is

$$\Phi = \frac{4 \cdot \overline{m}}{\rho_{inl}^{*} \pi D_2^2 U_2} \tag{1}$$

Polytropic efficiency at full range of parameters is

$$\eta_p^* = {h_p^*}/{h_i}, \; h_p^* = h_p + h_d, \; h_i = c_p \Delta T^*. \tag{2}$$

Polytropic head coefficient at full range of parameters is

$$\psi_p^* = h_p^* / u_2^2 \tag{3}$$

Table 1. Objects of research for development of verification base

№	**Index**	**D_2, m**	**z, pc**	**b_2/D_2**	**Φ_d**	**Ψt**	**β_{bl2}, grad**
1	RK-61			0.049			
2	RK-62	0.442	24	0.045	0.064	0.74	59.5˚
3	RK-63			0.04			
4	RK-51			0.041			
5	RK-52	0.442	24	0.038	0.064	0.9	90˚
6	RK-53			0.034			

NUMERICAL CALCULATION

Figure 2 shows the computational model of the two-stage stage RK-61. Ansys CFX 18.0 was used for modeling. The solution method with the turbulence model RANS SST is chosen. The model of the medium is a perfect gas, obeying the Mendeleev-Clapeyron equation. All tests were carried out at Mach number Mu = 0.78. At the inlet, a total pressure equal to atmospheric and total temperature was set. The output is a mass flow. The calculated grid was about 6 million elements with y + <2. The calculation continued until the convergence of the decision on unbalances and mean square discrepancies, about 200 iterations.

Figure 3 shows the constructed and calculated models of the working wheels of the RK-6,5 series, with a total of 6 impellers. The impellers are characterized by different theoretical head ratio Ψt and b_2/D_2 ratios. Such geometrically similar impellers are very convenient for analyzing the influence of specific gas dynamic parameters on the error of calculations.

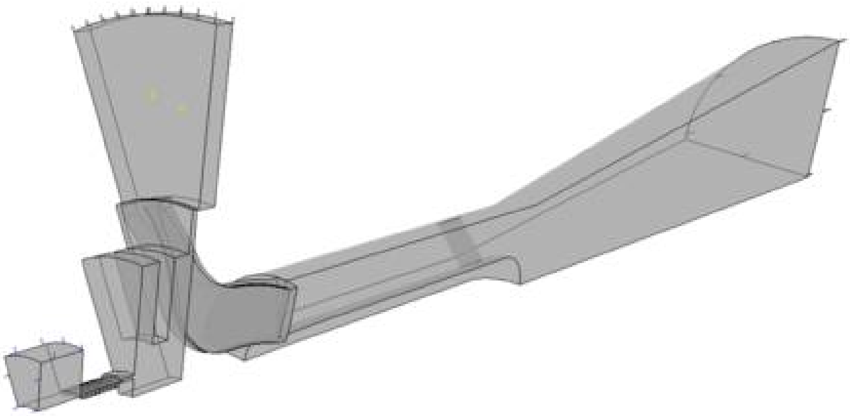

FIGURE 2. Calculation model of stage RK-61

Below in Figures 4-7, gas dynamic characteristics are presented in comparison with the data of experimental studies.

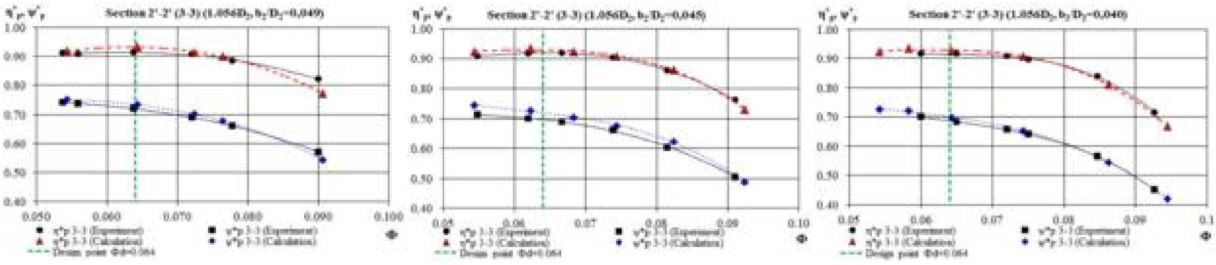

FIGURE 4. The plots of the η^*_p, ψ^*_p dependencies on the conditional flow coefficient Φ for the models RK61, PK62, PK63 in the section 2-2

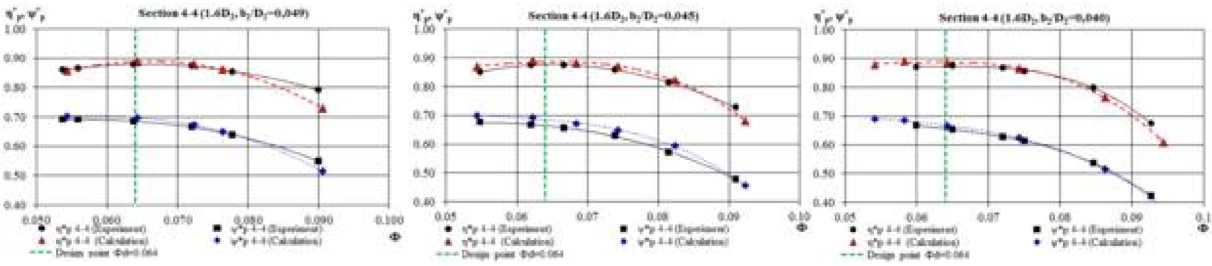

FIGURE 5. The plots of the η^*_p, ψ^*_p dependencies on the conditional flow coefficient Φ for the models RK61, PK62, PK63 in the section 4-4

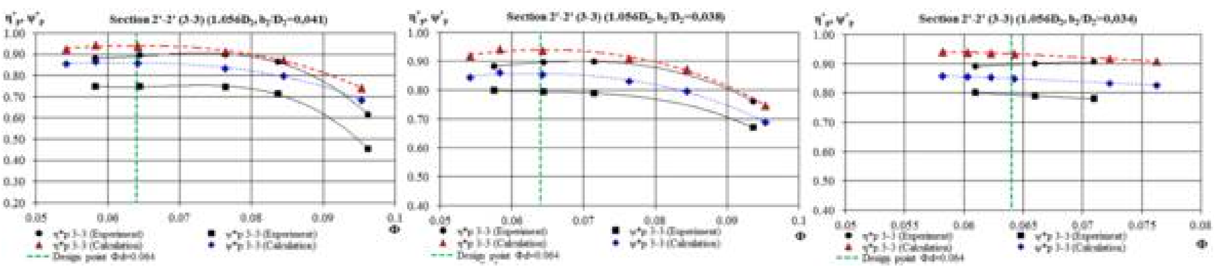

FIGURE 6. The plots of the η^*_p, ψ^*_p dependencies on the conditional flow coefficient Φ for the models RK51, PK52, PK53 in the section 2-2

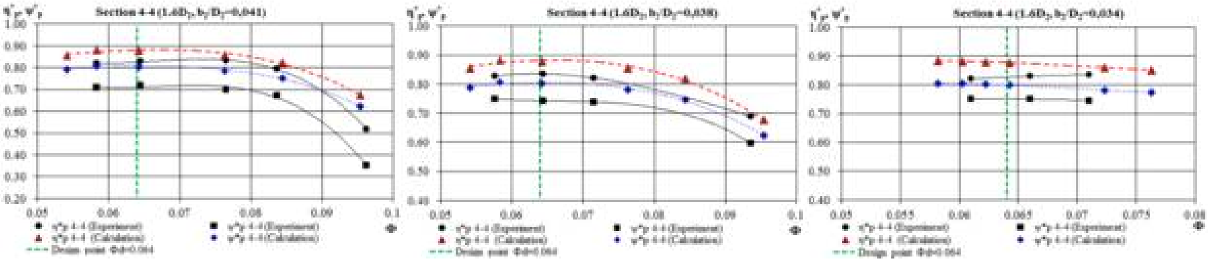

FIGURE 7. The plots of the η^*_p, ψ^*_p dependencies on the conditional flow coefficient Φ for the models RK51, PK52, PK53 in the section 4-4

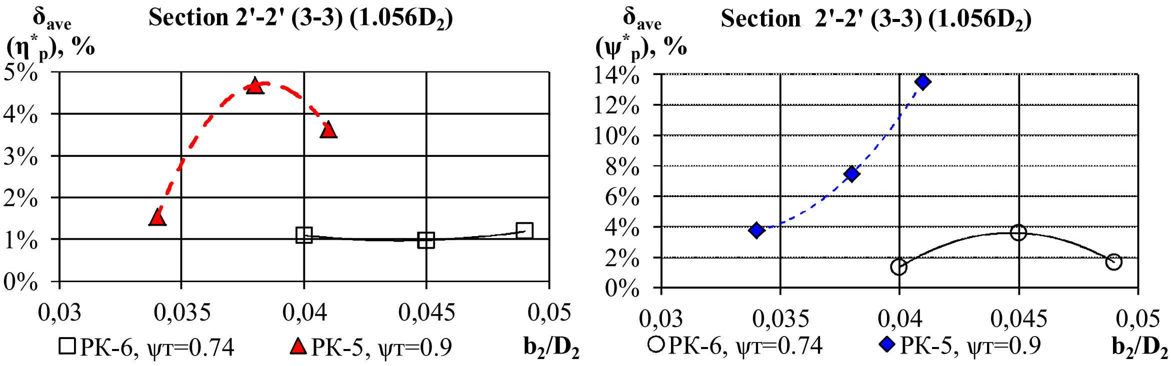

FIGURE 8. Graphs of the average relative error with respect to η^*_p (left) and ψ^*_p (right) from b2 / D2 for series PK5, PK6 in section 3-3

Error analysis is performed for the impeller without taking into account the maximum flow point, where the greatest discrepancy with the experimental data is observed (in particular for RK-5).

For RK-6 models, the relative error for the polytropic efficiency and the polytropic head coefficient for the full parameters in the design mode does not exceed 1.5% in the section at the exit from the impeller 2'-2 '(3-3) and at the outlet from the VLD (4 -4), which is comparable to the error of the full-scale experiment. The average relative error over the entire characteristic does not exceed 2% for polytropic efficiency and 4% for the polytropic head coefficient in full parameters.

With an increase in the theoretical head in the RK-5, the error in the computation in the design mode η * n has not increased significantly more than 5% in the cross section (3-3) and (4-4). The average relative error throughout the characteristic does not exceed 6% for polytropic efficiency and 14% for the polytropic head coefficient in full parameters.

Figure 8 shows that when the ratio b_2/D_2 decreases significantly, the difference in the numerical and experimental gas dynamic characteristics of η * n for the PK-6 series is not observed. However, an increase in the error increases with an increase in the theoretical head coefficient Ψt and, on average, for PK-5 is more than three times higher than for PK-6, and the maximum error is observed for large values of b_2/D_2. At $b_2/D_2 = 0.034$ for RK-5, the error becomes comparable with RK-6.

PARAMETERIZATION OF THE FLOWING PART

To parametrize and build a solid geometric model, Ansys DesignModeler was used - embedded CAD, integrated into the workbench work system.

As a result, a model is constructed (Fig. 9.10), which allows building models of flowing parts of the 3D impeller within a wide range. The average line of the rotating guide device is constructed by a quadratic or cubic law, providing a given angle of flow entry on the average radius. The model allows to build both semi-open and closed impellers.

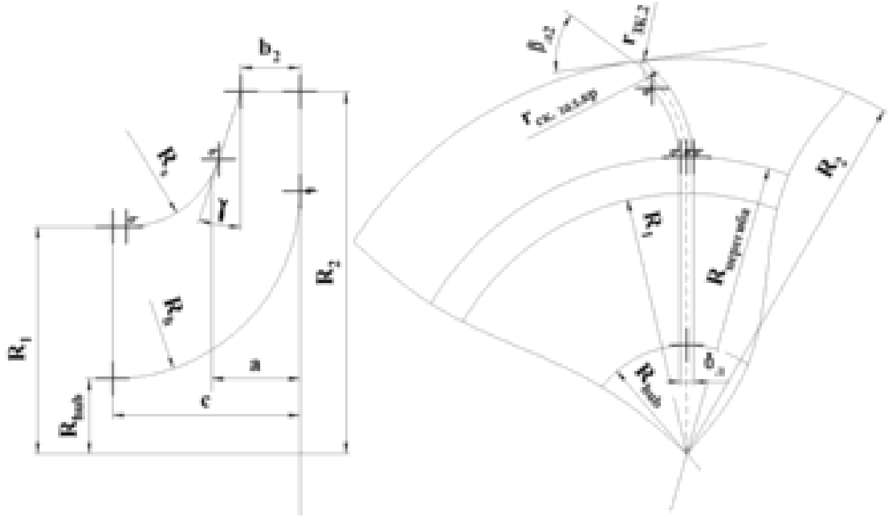

FIGURE 9. Scheme of the meridional and radial section of the impeller.

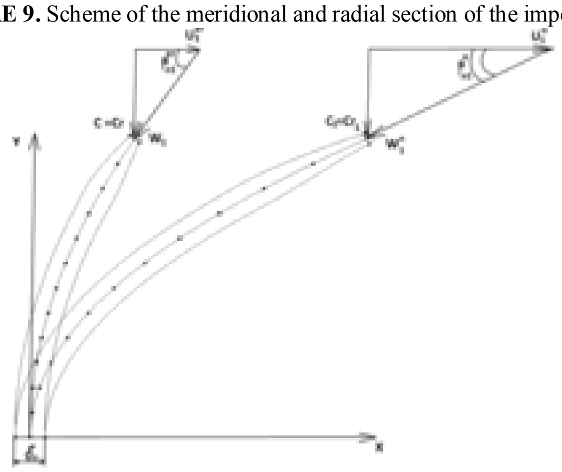

FIGURE 10. Sections of rotating guide device on the hub and shroud with the indication of the velocity triangles.

A total of 16 parameters are used to determine the geometric model of the flowing part of the impeller. By specifying these parameters in tabular form, the model is constructed.

R2 is the Diameter of the impeller
R0=R1 is the The radius of the blade start
R_{hub} is the Radius of the hub
Rh is theRadius of the rounding of the main disk
Rs is the Radius of the rounding of the shroud
γ is the Shroud disc inclination angle
b_2 is the The height of the blades at the exit
β_{bl1} is the The angle of the blade inlet
β_{bl2} is the The angle of the blade exit
δ_{bl} is the Blade thickness
a is the Beginning of the inflection point
c is the Axial extension of the blade
Z_2 is the Number of blades
r_{le} is the Leading edge rounding radius
r_{te} is the Trailing edge rounding radius
r_{rbl} is the Radius of curvature of blade

Advantages of the model are:

1. Integration with the calculator grid generator, and Ansys CFX solver.
2. The possibility of automatic multiparametric and multicriteria optimization
3. The possibility of connecting the diffuser, interdisk clearance, labyrinth seals and inlet nozzle.

The disadvantages of the model are:

1. Lack of modeling of blade blades.
2. Construction of the radial part of the blade only along the radius.
3. Use of blades of finite thickness

CONCLUSIONS

Results of comparison of gas dynamic characteristics with experimental data of 6 computational models of two-link stages are obtained. For all stages, the minimum error in the optimum mode zone is characteristic. The effect of the theoretical impeller head on the error is noted. There is a discrepancy in the calculation error between the same type of impellers, which decreases with decreasing b_2/D_2.

Such a discrepancy in the calculation errors for geometrically similar impellers (the same rotating guide device, D_2, D_h, D_1, the shape of the hub line) of different pressure is explained by the imperfection of the semi-empirical turbulence models used. In order to obtain a good agreement between the numerical and experimental characteristics, it is necessary to calibrate the empirical coefficients for a particular model. Before calibration, it is necessary to study the direct effect on the error in the computation of geometric and associated gas dynamic parameters. In the next paper, analogous studies are continuing for the series RK-4 and RK-2 having a theoretical flow coefficient coinciding with the RK-6 and an input directing device, but having different D_1, D_2. In general, a qualitative and quantitative agreement with the experimental data was obtained for the two-level step RK-6. Data can be used to expand the existing verification database. Such a database can be used to design new stages of centrifugal compressors based on numerical simulation of the characteristics of centrifugal compressors.

A model is developed for parametric projection of geometric parameters of the impeller for integration into the multicriterion and multiparametric optimization algorithm developed earlier [1]. Such a model will allow to carry out automated CFD-calculations of different types of two-element stages with 3D impellers to achieve the requirements of the technical task.

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