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Numerical Study Of The Equivalent Roughness Effect On The Low-Flow Centrifugal Compressor Impeller Characteristics

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Abstract. The article presents the results of a numerical study of the effect of equivalent sand-grain roughness on the characteristics of a low-flow centrifugal compressor stage. Low-flow stages are widely used in high-pressure compressors as the last stages. These compressors are used for natural gas booster compressor stations, as well as in technological processes for the production of methanol, ammonia, high - pressure polyethylene, etc. The efficiency of the low-flow stages is lower than that of medium-and high-flow centrifugal compressors stages. Stages operate at high pressure and have low gas volume flow rates. For this reason they have narrow channels of the flow part, significant friction and leakage losses. The study was conducted in the Ansys CFX 19.2 software package. In the first part of the study, the low-flow stage numerical model is validated with the test results. Full-scale tests of the stage were carried out at the LPI-SPbPU earlier. In the second part of the study, impeller equivalent sand-grain roughness numerical study carried out. As a result, the characteristics of the

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stage were obtained at four different values of equivalent sand-grain roughness k_s . The impeller losses were estimated. It is established that when the k_s values increase, the polytropic head and efficiency values do not decrease equally over the entire characteristic. Characteristics are significantly reduced in areas of high expenditure and change slightly with minimal expenditure. The internal head coefficient varies slightly depending on k_s .

Key words: centrifugal compressor, CFD, sand-grain roughness, numerical simulation.

Introduction

High-efficiency centrifugal compressors design is important task, due to the large capacities of the machines in operation [1-3]. Modern computational methods allow us to pre-evaluate the characteristics of compressor stages. However, the results of numerical solutions do not always coincide with the results of tests of real objects. This depends on many factors: from the features of the mathematical model to the accuracy of the methods used and the capabilities of the supercomputer. There are many papers devoted to numerical studies of the flow parts of compressors [4-11]. Most of these studies are devoted to medium - and high-flow stages. A small number of studies for low-flow stages are presented. Because of specific applications of low – flow stages for the high-and ultra-high-pressure compressors. The stages are used in compressors for supplying natural gas to the reservoir, in technological processes for the production of ammonia, methanol and ethylene, as well as in the processing of associated petroleum gas. Numerical studies of low-flow stages [12-17] show that standard solutions lead to different results. The results show that the numerical characteristics are higher than the results of real experiments. There is an overestimation of the simulated characteristics of polytropic head and efficiency. As is known, the nature of gas flow in the flow part of low-flow stages has a number of features. In contrast to the medium-flow stages, the flow in low-flow stages has a number of features. Small channel widths and high pressures lead to the boundary layers closing in the meridional flow. Friction losses on the limiting surfaces in the impeller increase. Increase the friction loss in the gaps and leaks in the labyrinth seals. Studies of high and ultrahigh pressure stages in the LPI-SPbPU [18] have shown a significant influence of the impeller and fixed elements walls roughness on the stage energy characteristics. According to [19], there are three flow modes in rough channels:

1. Mode without roughness $v^* k_s^2 / \nu < 5$, where ν – flow velocity; v^* – dynamic speed; k_s – equivalent sand roughness. In this mode, the grain sizes are small and are located inside the laminar sublayer. Losses depend only on the Reynolds number Re . The streamlined surfaces are hydraulically smooth.

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2. The transition regime $5 < \nu^* k_s^2 / \nu < 70$. In this mode, the roughness elements partially protrude from the laminar sublayer, thereby creating additional resistance.

3. Mode with the effect of roughness $\nu^* k_s^2 / \nu > 70$, all roughness elements protrude from the laminar sublayer.

The channel can be hydraulically smooth or rough depending on the Reynolds number. Most compressor channels are not hydraulically smooth due to various micro-irregularities that occur during the manufacturing process. Therefore, it is advisable to numerically calculate flow regimes with rough walls.

The aim of the study is to evaluate the influence of the roughness of the bounding surfaces and impeller blades on the stage characteristics during numerical modeling in the AnsysCFX program complex.

The object of the study is a model low-flow stage of a centrifugal compressor with a design flow conditional coefficient $\Phi=0.0075$ and a theoretical head coefficient $\psi_t=0.48$. The Stage was designed And tested by the Department of compressor, vacuum and refrigeration equipment of LPI-SPbPU [18]. The Low-flow centrifugal compressor stage scheme is shown in fig. 1. Impeller walls arithmetic average roughness is $Ra=0.63$ or $k_s=1.46$, for the first calculation take equal 2.

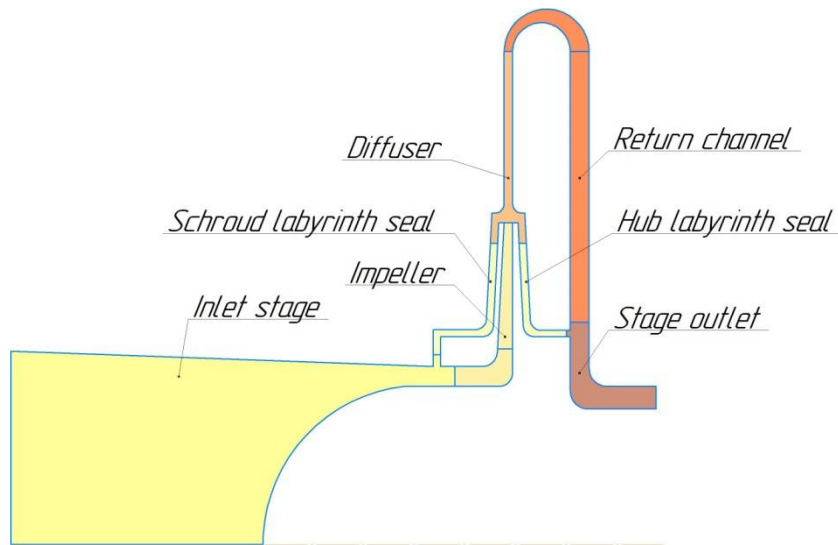


Fig. 1. Low-flow centrifugal compressor stage scheme.

2 METHODS

Stage CFD model was carried out in the Ansys software package. The numerical model of the flow part is based on the recommendations [20-27]. Geometric models was build in the DesignModeler. TurboGrid and Icem CFD modules were used to create a block-structured computational grid. The CFD model consists of seven do-

mains. A perfect nitrogen gas was selected as the working medium. Total pressure is 405300 Pa, and the total temperature is 302 K set at the stage inlet. The SST turbulence model used. The equivalent sand-grain roughness k_s of the fixed element walls are 9 μm , and labyrinth seals and gaps walls are 3 μm . There four values of the equivalent sand-grain roughness k_s of the impeller walls are used: 2, 8, 32, 128 μm . Figure 2 shows the investigated walls of the impeller.

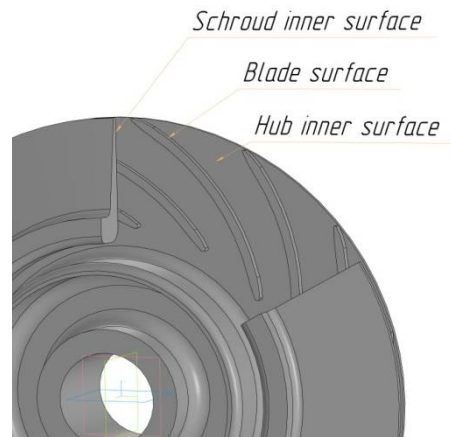


Fig. 2. Investigated surfaces of the impeller.

The analysis of the impeller characteristics was considered in four control sections.

The first is located at the stage inlet, symbol 0-0. The second – at the impeller outlet on the outer diameter D_2 , symbol 2-2. The third – in the vaneless diffuser at a distance of $1,05D_2$, symbol 2'-2'. The fourth section is located directly at the stage outlet, symbol 0'-0'.

The results of the numerical study were processed using the formulas given below. The relationship between equivalent sand roughness and arithmetic average roughness was calculated as [18]:

$$k_s = 2.19Ra^{0.877} \quad (1)$$

Flow Conditional coefficient:

$$\Phi = \frac{4\bar{m}}{\rho_0^* \pi D_2^2 u_2}, \quad (2)$$

where \bar{m} , kg/s - mass flow rate; ρ_0^* , kg/m³ – inlet total to total density; D_2 , m – outer impeller diameter; u_2 , m/s - peripheral speed.

Coefficient of polytropic head by total to total parameters:

$$\psi_{pol}^* = \frac{h_{pol}^*}{u_2^2}, \quad (3)$$

where h_{pol}^* , J/kg - polytropic head by total to total parameters.
Internal head coefficient:

$$\psi_i = \frac{h_i}{u_2^2}, \quad (4)$$

где $h_i = \Delta i^*$, J/kg - internal head.

Total to total Polytropic efficiency:

$$\eta_{pol}^* = \frac{\psi_{pol}^*}{\psi_i}, \quad (5)$$

To calculate the theoretical head coefficient, the dependence was used:

$$\psi_t = \frac{c_{u2}}{u_2}, \quad (6)$$

где c_{u2} – circumference velocity.

To calculate the loss coefficient in the impeller ζ used formula [18]:

$$\zeta_{imp} = \frac{2\psi_t \left(1 - \frac{\psi_{pol}^*}{\psi_t}\right)}{r_1^{-2} + \left(\frac{\Phi_{lk}}{2r_1 b_1}\right)^2}, \quad (7)$$

where Φ_{lk} - conditional flow coefficient, taking into account the size of leaks \bar{m}_{leak} .

3 Results

The results of the work were obtained using computational resources of Peter the Great Saint-Petersburg Polytechnic University Supercomputing Center (www.spbstu.ru). Figure 3 shows the dependences of the efficiency on the conditional flow coefficient for different settings of the equivalent sand-grain roughness of the impeller in the section 2'-2'.

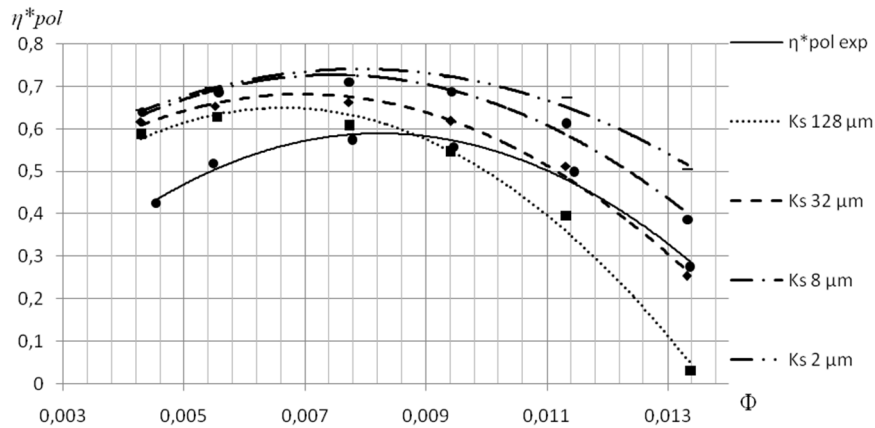


Fig. 3. Numerical and experimental characteristics of the polytropic efficiency in the section 2'-2'; k_s of the impeller inner surfaces changes from 2 to 128 μm .

Figure 4 shows the numerical and experimental polytropic head characteristics in the section 2'-2'.

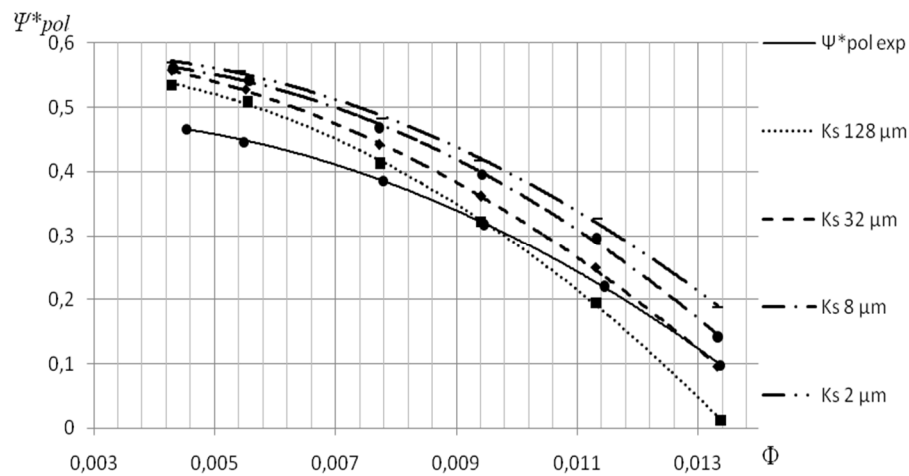


Fig. 4. Numerical and experimental characteristics of the internal head coefficient of the stage; k_s of the impeller inner surfaces changes from 2 to 128 μm .

Figure 5 shows the dependence of the internal head coefficient on the conditional flow coefficient at different values of the equivalent sand-grain roughness.

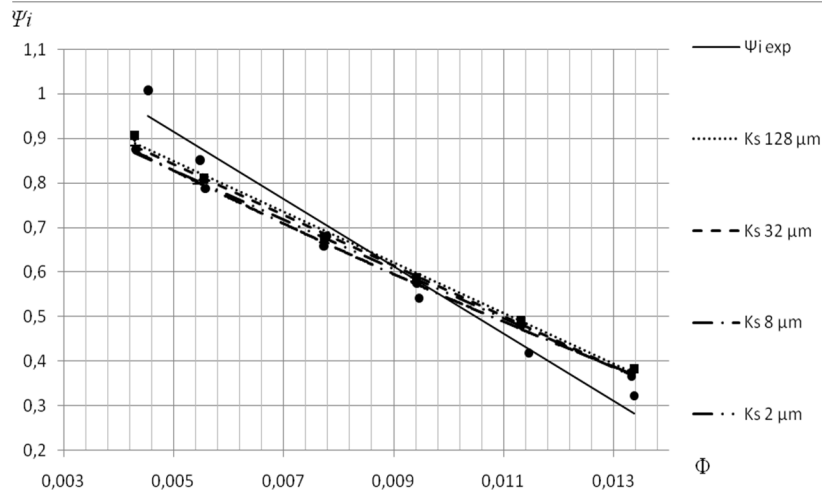


Fig. 5. Numerical and experimental characteristics of the internal head coefficient of the stage; k_s of the impeller inner surfaces changes from 2 to 128 μm .

The internal head coefficient practically coincides in modes close to the design mode. At higher values of the flow coefficient, there is an overestimation of the temperature at the outlet of the impeller. This leads to an overestimation of the internal head coefficient.

The results of numerical modelling variant with $k_s=2 \mu\text{m}$ (that corresponding to the test roughness) showed almost equivalent increase in values over the entire characteristic of the efficiency and head coefficients.

With an increase in the equivalent roughness of the impeller, the polytropic efficiency and pressure coefficients decrease in the modes of increased flow. Further increase in k_s leads to the fact that the right part of the characteristic approaches the experimental data and then becomes much smaller. In low-flow modes, the effect of roughness is less. This effect decreases as the flow rate decreases.

The results obtained, in terms of pressure losses, correspond to the physical representation of a decrease in the height of the laminar boundary layer with an increase in the velocity of the turbulent flow. At the same time, the internal and theoretical head characteristic remains almost unchanged. Figure 6 shows the dependences of the theoretical head coefficients calculated using formula 6 and [18]. In [18], the theoretical head coefficient in section 2-2 was calculated using the potential flow method.

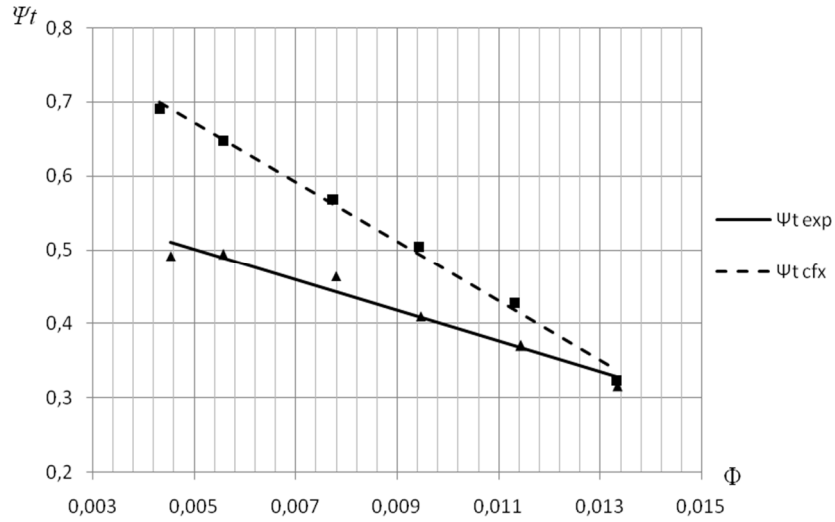


Fig. 6. Dependences of the theoretical head coefficients on the conditional flow coefficient. k_s of the inner surfaces of the impeller 2 μm .

Figure 7 shows the dependence of the impeller loss coefficient on the conditional flow coefficient for different values of the equivalent sand roughness of the impeller inner surface.

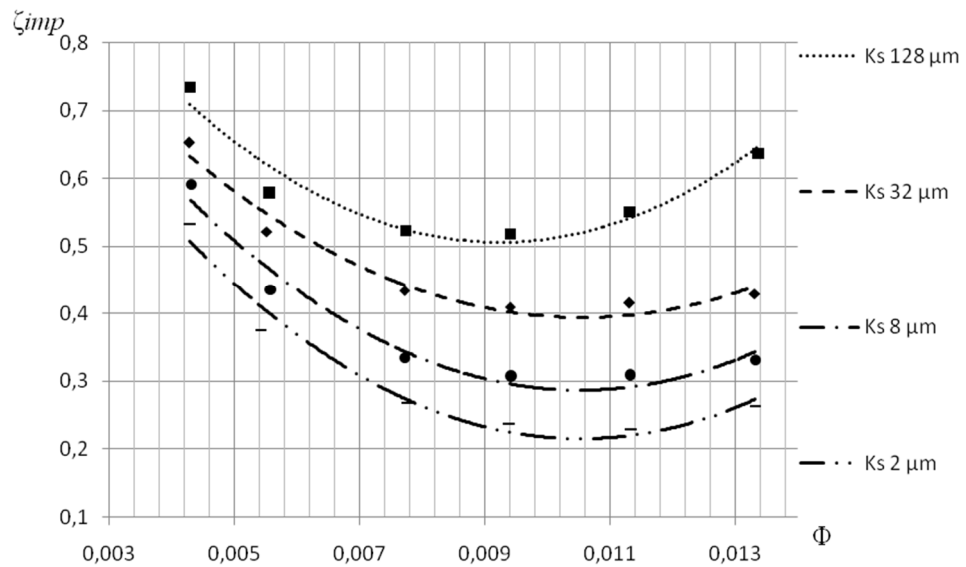


Fig. 7. Dependence of the impeller loss coefficient on the conditional flow coefficient for different values of equivalent sand roughness.

4 Discussion

Numerical calculations were performed with different equivalent sand-grain roughness k_s of the impeller walls. The k_s values of all other elements walls remained unchanged. The calculated characteristic with the design roughness ($k_s=2 \mu\text{m}$) is located equidistant above the experimental curve. As the roughness increases, the steepness of the characteristic increases at increased flow rates. This is due to the different thickness of the laminar sublayer when the gas moves through the flow part, depending on the flow rate. At higher flow rates, the speeds are higher. The thickness of the laminar sublayer decreases and the roughness leads to an increase in losses. Lowering the gas velocity below leads to thickening of the laminar sublayer. The flow regime is close to hydraulically smooth, so even with the equivalent sand roughness of $k_s=128$ microns, the reduction in parameters is not so high. This type of change in characteristics corresponds to the observed experimental studies on this topic.

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