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Analysis of the influence of theoretical head coefficient on the low-flow centrifugal compressor stage characteristics

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Abstract. The article presents the results of a numerical study of the influence of the theoretical head coefficients on the characteristics of a low-flow stage of a centrifugal compressor. The study was conducted in the Ansys CFX 19.2 software package. As a result of a numerical study, it was found that the drop in the efficiency of the compressor stage with an increase in the theoretical head coefficient is associated with a decrease in the efficiency of the stationary elements due to the increased absolute speed behind the impeller. At the same time, an increase in the theoretical head coefficient leads to a decrease in losses associated with the coefficient $1 + \beta_{lk} + \beta_{fr}$, and therefore, an increase in the polytropic efficiency of the impeller is observed at values of the theoretical head coefficient $\Psi t \leq 0.64$.

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

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1 Introduction

High-efficiency centrifugal compressors design is important task, due to the large capacities of the machines in operation [1-6]. 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. Centrifugal compressor stages with a low-flow coefficient are widely used in multistage high-pressure compressors, in refrigeration compressors, in oil and gas, chemical and other industries. Low volumetric capacity and significant losses due to gas leakage and friction losses lead to low efficiency of this type of stage. The cleanliness of the surface treatment has a large effect on the losses and the flow pattern due to the small size of the flow path channels. Special deformations of flow path parts accounting methods are required to maintain radial and axial clearances in such stages. The flow of a viscous gas in the flow path of a low-flow stage has several features that do not allow the theory of ideal liquid flow to be applied. A significant thickness of the boundary layer leads to flow path channels obstruction. The flow is caused by large friction losses on the limiting surfaces and a developed trace stream on the impeller blades. A quasi-three-dimensional model of the viscous gas flow in the inter-blade channels and recommendations for the design of such stages were obtained based on low-flow stages studies carried out at the Department of Compressor, Vacuum and Refrigeration Engineering [7]. Despite extensive research and testing of low-flow stages, their numerical studies are limited [8-13].

The work [14] shows the combined characteristics of low-flow stages with a close value of Φ_{opt} and the dimensionless rotation number K_n = 0.14...0.16, with different values of the conditional flow coefficient in the design mode. The analysis showed that a change in the theoretical head coefficient leads to a decrease in the stage reactivity degree. It is noted that with the value of the theoretical head coefficient ψ_t = 0.58...0.62, the efficiency of the stage practically does not change due to the opposite effect of the coefficient $(1 + \beta_{lk} + \beta_{fr})$ and the reactivity degree. In real tests, it is difficult to change the theoretical head coefficient by changing the outlet angle of the impeller blade on the same impeller. However, this can be realized using computational gas dynamics programs.

The study aims to assess the influence of the impeller theoretical head coefficient on the stage characteristics. Two tasks are being solved for this. The first task is to verify the numerical model of the compressor stage flow path with the results of model tests at the design mode of the stage operation. The second task is the numerical calculation of five versions of impellers with different coefficients of the theoretical head. The theoretical head coefficient change is achieved by changing the outlet angle of the blade β_{b2}.All other parameters of the stage remain unchanged. The Ansys CFX 19.2 software package is used to solve the set tasks. The results of the work were obtained using the computing resources of Peter the Great St. Petersburg Polytechnic University supercomputer center (www.scc.spbstu.ru).

2 Methods

The object of the study is an intermediate model stage of a centrifugal compressor with a design conditional flow rate Φ = 0.0083. The stage was designed and tested at the Compressor Engineering Department of SPbPU [7]. The conditional flow coefficient is determined by the formula:

$$\Phi = \frac{4\overline{m}}{\rho_0^* \pi D_2^2 u_2}, \tag{1}$$

where $\overline{m}$, kg/s - mass flow rate; ρ^*_0, kg/m^3 – inlet total to total density; D_2, m – outer impeller diameter; u_2, m/s - peripheral speed.

Real tests were carried out on a closed-loop bench. The obtained characteristics were used to verify the numerical model of the flow path. The mathematical model is built in the Ansys CFX 19.2 software package. The Design Modeler module was used to create geometric models of elements with the ability to create flow path contours. The TurboGrid and ICEM CFD modules were used to create a block-structured computational grid. The flow path consists of seven elements connected in the Ansys CFX module. The general diagram of the stage flow path is shown in Figure 1.

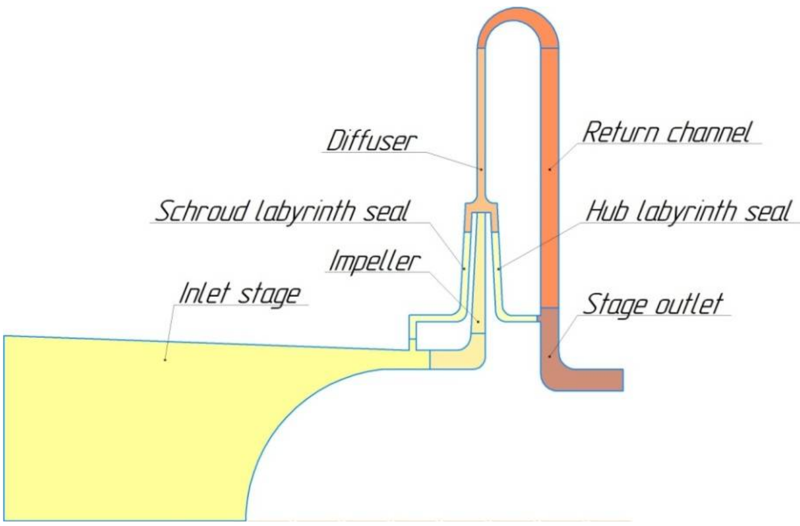

Fig. 1. Low-flow centrifugal compressor stage scheme.

Numerical simulation of viscous gas flow was carried out under the boundary conditions shown in Table 1. Six control points of mass flow, corresponding to the experimental data, were used to plot the characteristic.The Frozen Rotor model was used as the interunit interface, except for the connection of the impeller and the vaneless diffuser. The Stage model was set for this interface. The SST turbulence model was chosen, and the problem statement was stationary. N_2 Ideal Gas was taken as flowing gas in the stage. The boundary conditions were set according to the recommendations of the authors [15-22]

Table 1. The boundary conditions.

Mode number	1	2	3	4	5	6
P_0, atm	20	20	20	20	20	20
T_0, K	302	302	302	302	302	302
$\overline{m}$, kg/s	5,53	4,76	4,28	3,72	2,60	1,81
n, rpm	11000	11000	11000	11000	11000	11000

The main geometrical parameters of the impeller and the design coefficients of the low-flow stage are given in Table 2.

Table 2. Impeller parameters of the original model low-flow stage.

β_{b1}	β_{b2}	Z_{imp}	b_1,mm	b_2,mm	Ψ_T	Φ_{opt}
17	24,5	12	7,5	4,7	0,48	0,083

Four additional impeller variants were constructed to assess the effect of changing the blade outlet angle on the stage performance. The impeller blade outlet angle changing scheme by changing the middle line of the blade is shown in Figure 2.

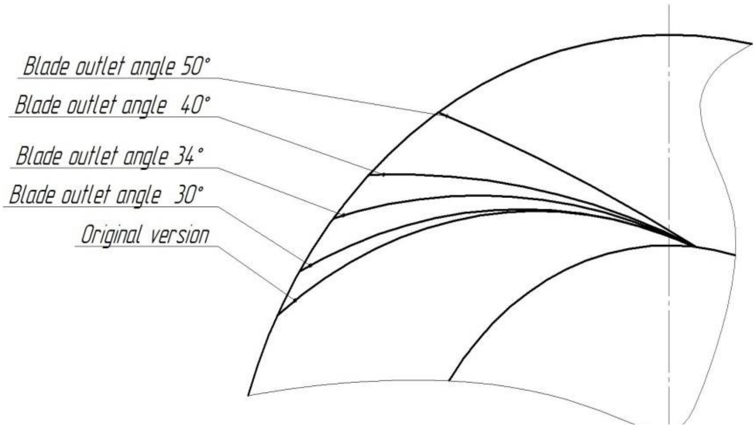

Fig. 2 A schematic diagram of the blade outlet angle variation along the midline

All calculations were carried out taking into account the surface roughness of the flow path [23-25]. The arithmetic mean roughness Ra = 5 was used for stationaryelements; roughness Ra = 0.63 was used for moving parts and seal walls. The following formula was taken to set the equivalent sand roughness in Ansys CFX [7]:

$$k_s = 2.19Ra^{0.877} \quad (2)$$

3 Results and Discussion

The technique [7] was used to process the results of a numerical study. The location of the check sections in which the main gas-dynamic parameters were taken is shown in Figure 3

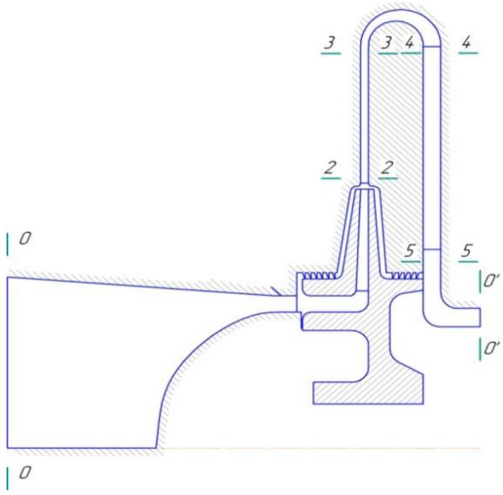

Figure 3 A schematic diagram of the check sections

Polytropic head coefficient taking into account the difference in gas kinetic energies:

$$\psi^*_{pol} = \frac{h^*_{pol}}{u_2^2}, \tag{3}$$

Internal head coefficient:

$$\psi_i = \frac{h_i}{u_2^2}, \tag{4}$$

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

Total to total Polytropic efficiency:

$$\eta^*_{pol} = \frac{\psi^*_{pol}}{\psi_i}, \tag{5}$$

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

$$\psi_t = \frac{c_{u2}}{u_2}, \tag{6}$$

где c_{u2} – circumference velocity.

Figure 4 shows the dependence of the total head coefficient on the conditional flow rate coefficient in comparison with the test results. Figure 5 shows the relationship between the theoretical head coefficient on the conditional flow rate coefficient. Figure 6 shows the dependence of the polytropic head coefficient and polytropic efficiency on the conditional flow rate coefficient in section 2-2.

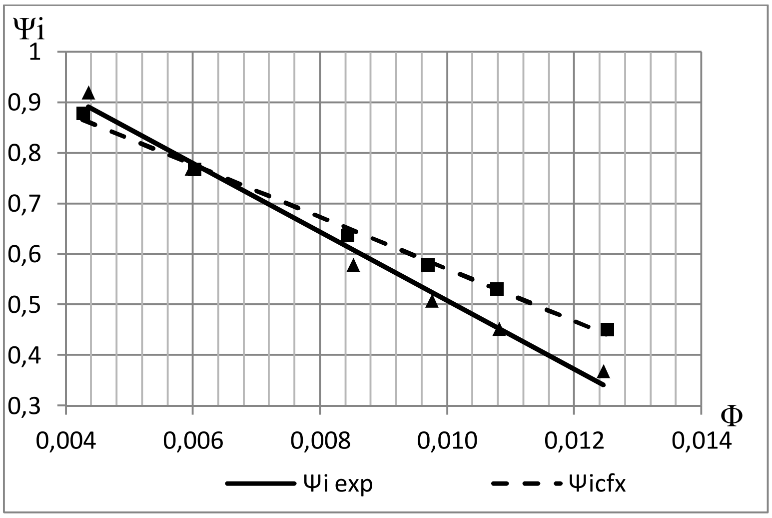

Fig. 4 The total head coefficient vs. conditional flow rate coefficient

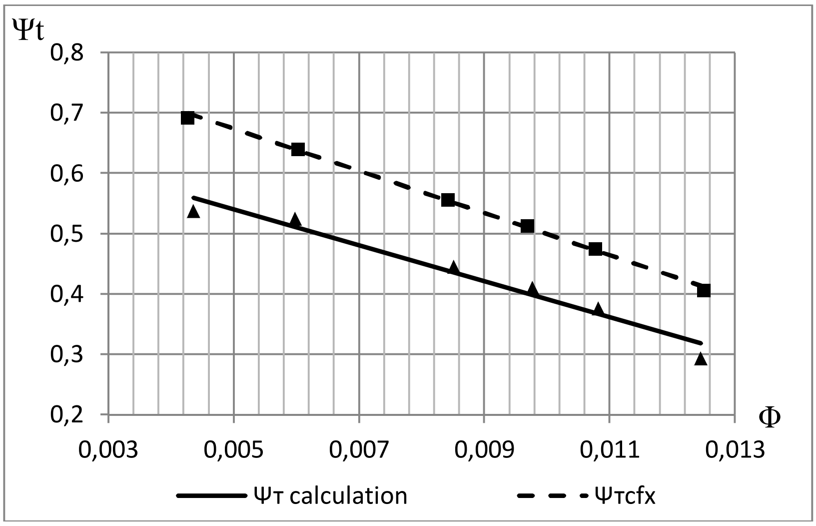

Fig. 5 The theoretical head coefficient vs. conditional flow rate coefficient

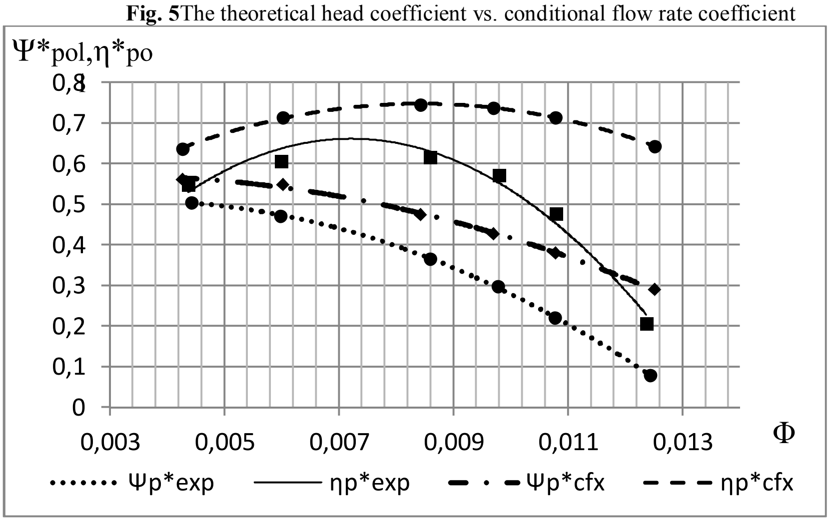

Fig. 6 Thepolytropic head coefficient and polytropic efficiency vs. conditional flow rate coefficient in section 2-2

The results of the work [26] were used for the grid independence analysis of the compressor flow path numerical model. According to the results of numerical modeling, it is obvious that the numerical characteristics are overestimated in almost all operating modes of the stage. Significant overestimation is observed in the area of high flow. Perhaps this kind of numerical characteristics has to do with the method to set the roughness of the impeller. The equivalent sand roughness of the investigated impeller version was set in such a way that the walls were considered hydraulically smooth.

For further research, three additional versions of impellers were built, differing only in the outlet angle of the blade. Table 3 shows the values of the outlet angles depending on the impeller version.

Table 3. Values of the blade outlet angle for the investigated impeller variants

Impeller version	1	2	3	4	5
βl_2, °	24,5	30	34	40	50

The stationary elements of the flow path are not changed. The inlet angle of return channel vane is designed for the first version of the impeller. Comparison of the characteristics of full parameters polytropic heads of the investigated version is shown in Figure 7.

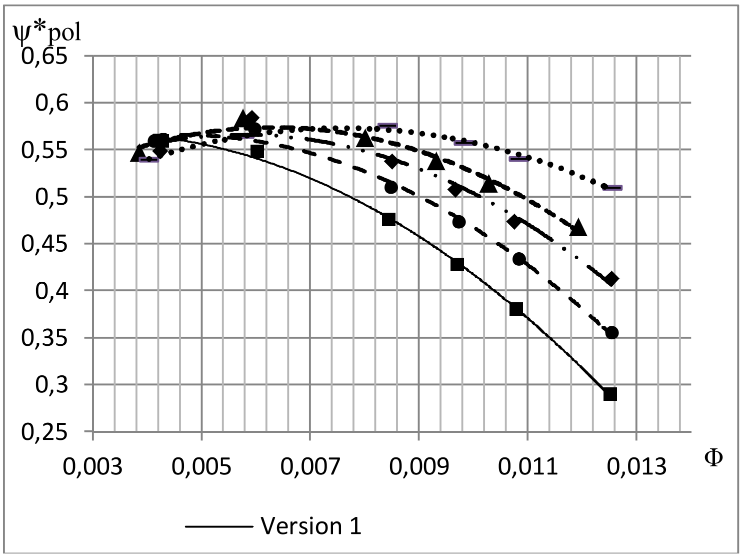

Fig. 7 Characteristics of the full parameters polytropic head in section 2-2 for different versions of the impeller.

Figure 8 shows a comparison of the total efficiency in section 2-2 for various impellers.

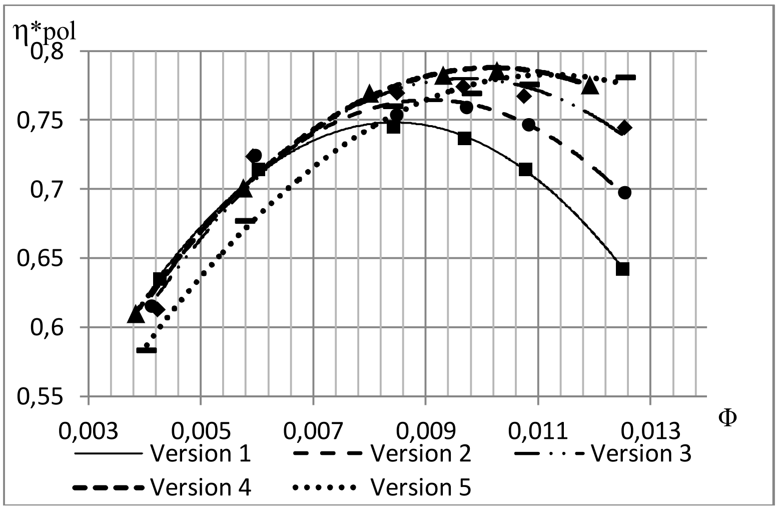

Fig. 8 Comparison of the full parameters polytropic efficiency in section 2-2 for different impeller versions.

The obtained characteristics show the shift of the stage operation design mode in the region of higher values of the conditional flow coefficient. As the outlet angle of the blade increases at a constant mass flow rate and a constant peripheral speed, the absolute velocity at the impeller outlet becomes higher. And this leads to an increase in losses in stationary elements. A decrease in the relative velocity at the outlet leads to an increase in the diffuseness of the inter-blade channels.

Figure 9 shows a comparison of polytropic efficiency in the 0-0 section for different impeller options.

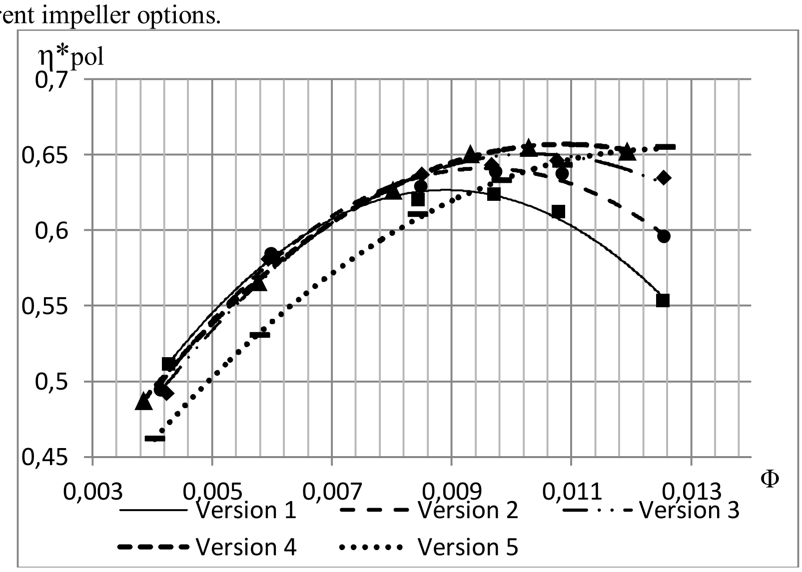

Fig. 9 Comparison of polytropic efficiency by full parameters in the 0-0 section for different impeller versions.

Figure 10 shows the characteristics of the theoretical head coefficient for various versions of the impeller.

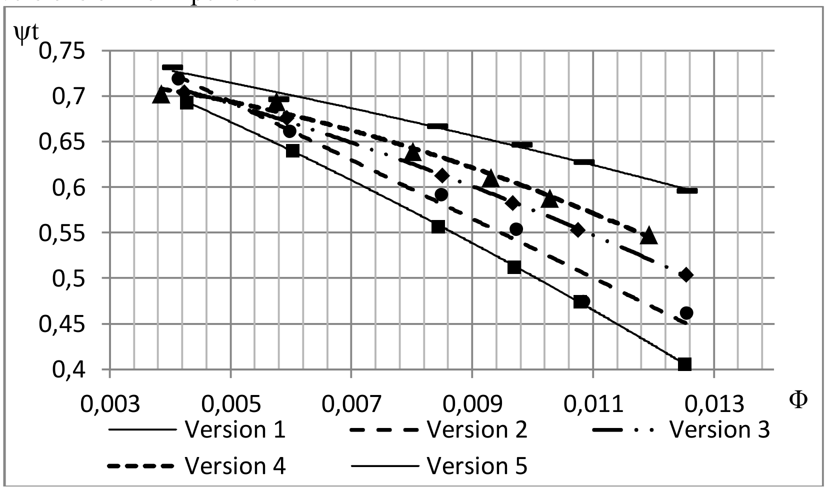

Fig. 10 Characteristics of the theoretical head coefficient for different versions of the impeller.

Figure 11 shows the dependence of the polytropic efficiency for the full stage parameters on the theoretical head coefficient. Evaluation of the efficiency of the entire stage is very conditional due to the off-design return channel operation mode. In the first impeller version, insignificant shock flow is observed in the return channel. And in the fourth version, the full shock flow happens in the return channel. Because of this, the polytropic efficiencies in the 0-0 section differ by no more than 10%. Obviously, with the shockless design of the return channel for each version of the impeller, the differences in efficiency will be larger, but this was not done in the framework of this study.

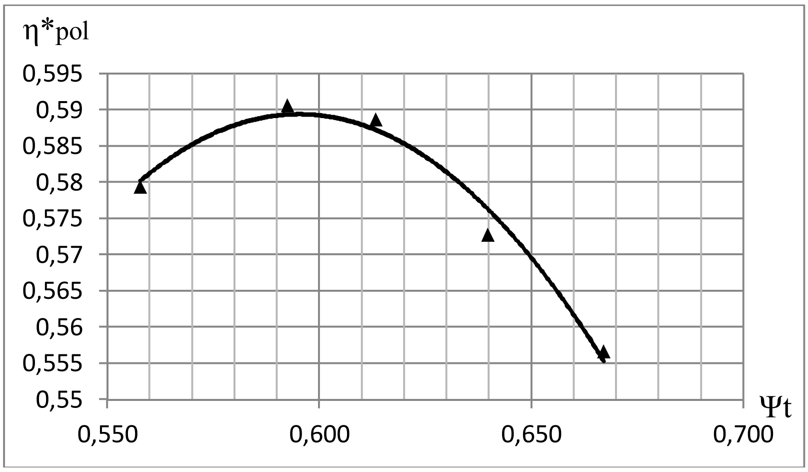

Fig. 11 The polytropic efficiency for the full stage parameters vs. the theoretical head coefficient.

4 Conclusions

The sharp drop in the efficiency of the compressor stage with an increase in the theoretical head coefficient is explained by a decrease in the efficiency of the stationary elements due to the increased absolute speed behind the impeller. At the same time, an increase in the theoretical head coefficient leads to a decrease in losses associated with the coefficient $1 + \beta_{fr} + \beta_{lk}$. And therefore, it leads to the impeller polytropic efficiency increase at values of $\Psi_t \leq 0.64$.

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