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High-Pressure Centrifugal Compressor Performance Enhancement by Improving the Radial Diffuser

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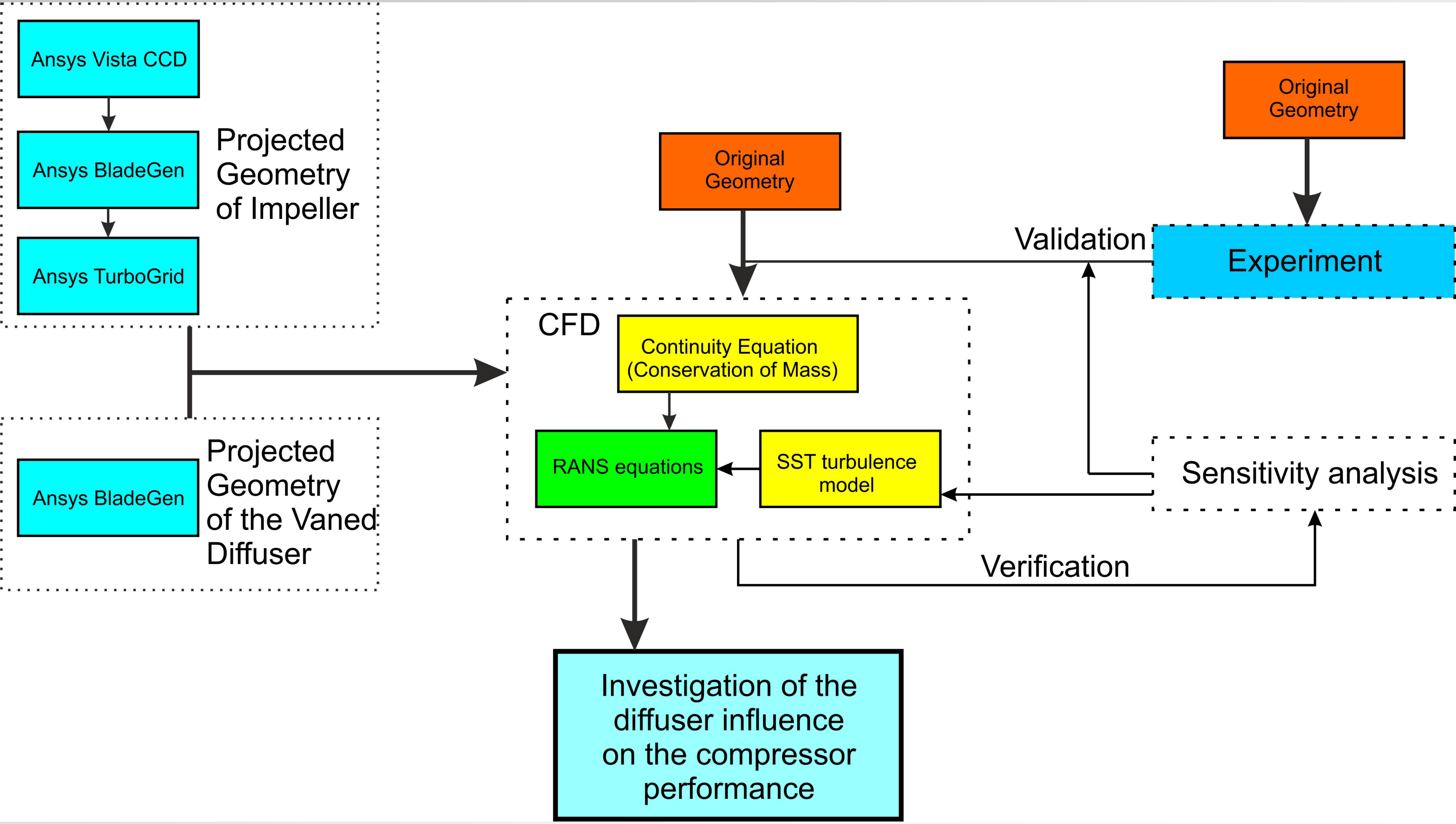
1 Introduction and Major Challenges

- Modern technologies for the production, design, and mathematical modeling of flow in hydraulic and pneumatic machines can significantly improve the quality of machines in various industries.
- The quality of compressor operation is affected by the shape of the blade, which is becoming easier and cheaper to make with the development of machines and production methods. However, compressor performance depends on many geometric parameters not only of the blade but also of the outlet diffuser.
- The best methods for studying characteristics are experimental, but they are quite expensive and the devices used to measure pressures and velocities can change flow patterns, which does not allow for an accurate assessment of the characteristics.
- Among the commercial and non-commercial flow simulation software, Ansys stands out, which allows using specialized programs (BladeGen, VistCPD, Ansys CFX) to carry out a full-fledged design and study of the characteristics of high-pressure compressor.
- General design methods based on physical considerations obtained by experimental studies and various one- and two-dimensional models do not allow for a full investigation of the flow in the diffuser and the flow separation from the walls and its impact on the characteristics.
- Therefore, it becomes relevant to study the three-dimensional flow in the compressor diffuser to achieve the perfection of the characteristics of high-pressure compressors.

Thus, **the aim of this study** is to design a vane diffuser for a high-pressure centrifugal compressor, to obtain gas flow patterns in vane diffusers with different blade inclination angles, and to determine the optimal angle.

Research Methodology



Mathematical Model

$$\frac{\partial u_i}{\partial t}+u_j\frac{\partial u_i}{\partial x_j}=F_i-\frac{1}{\rho}\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\frac{\mu_{ef}}{\rho}\frac{\partial u_i}{\partial x_j}\right];$$

$$\frac{\partial u_i}{\partial x_i}=0;$$

where: x_j are Cartesian coordinates; u_j are components of the mean velocity vector; $\mu_{ef}=\mu+\mu_t$ is the effective viscosity; μ_t is the turbulent viscosity; μ is the molecular viscosity.

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial}{\partial x_j}(\rho u_j k)=\frac{\partial}{\partial x_j}\left(\mu_{ef}\frac{\partial k}{\partial x_j}\right)+P_k-\beta^*\rho k\omega;$$

$$\frac{\partial(\rho\omega)}{\partial t}+\frac{\partial}{\partial x_j}(\rho u_j\omega)=\frac{\partial}{\partial x_j}\left(\mu_{ef}\frac{\partial \omega}{\partial x_j}\right)-\rho\beta\omega^2+Cd_\omega+\alpha\frac{\rho}{\mu_t}P_k,$$

where: k is the kinetic energy of turbulent pulsation; ω is the turbulence eddy frequency; P_k is production of turbulence kinetic energy; Cd_ω is a cross-diffusion term in the SST-model.

$$f_{r_1}=\max\left\{\min\left(f_{rotation},1.25\right),0.0\right\}\qquad f_{rotation}=\left(1+c_{r1}\right)\frac{2r^*}{1+r^*}\left[1-c_{r3}\tan^{-1}\left(c_{r2}\tilde{r}\right)\right]-c_{r1}$$

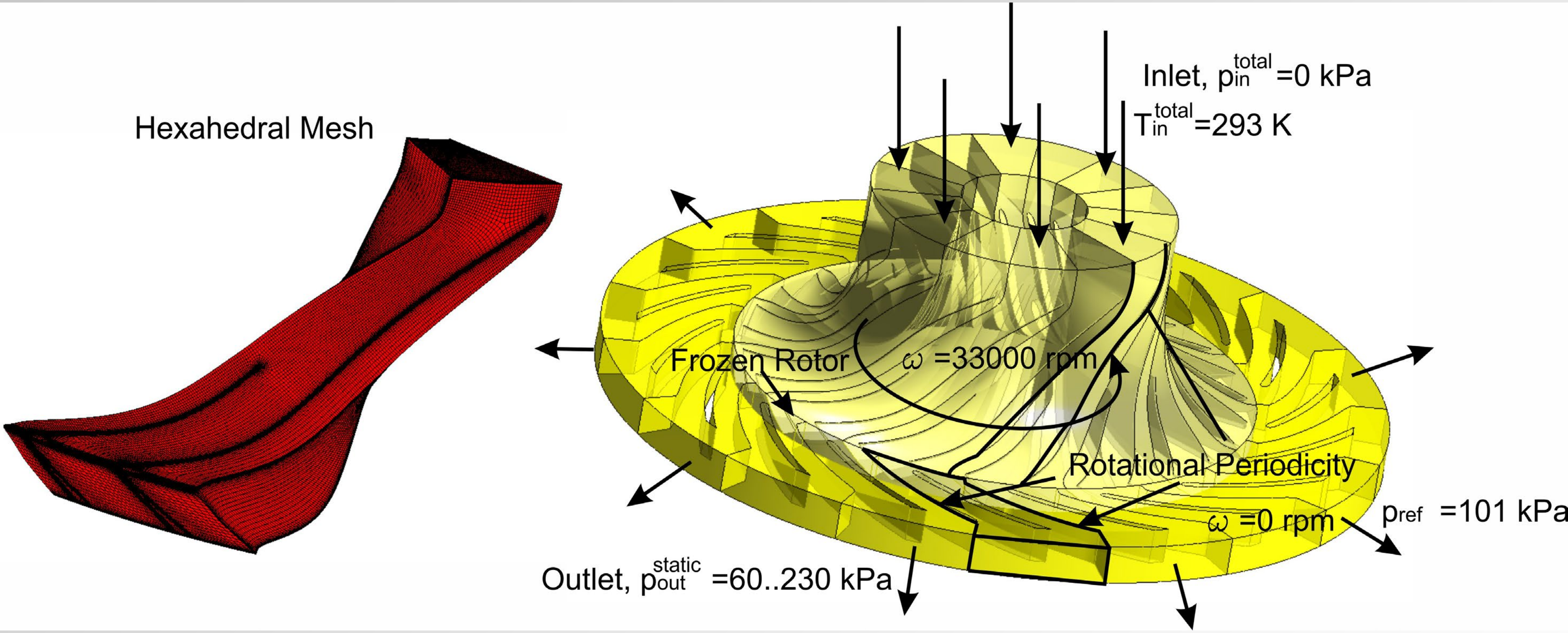
$$r^*=\frac{S}{\Omega}\qquad \tilde{r}=2\Omega_{ik}S_{ik}\left[\frac{DS_{ij}}{Dt}+\left(\varepsilon_{imn}S_{jn}+\varepsilon_{jmn}S_{in}\right)\Omega_m^{rot}\right]\frac{1}{\Omega D^3}$$

$$S^2=2S_{ij}S_{ij};\qquad \Omega^2=2\Omega_{ij}\Omega_{ij};$$

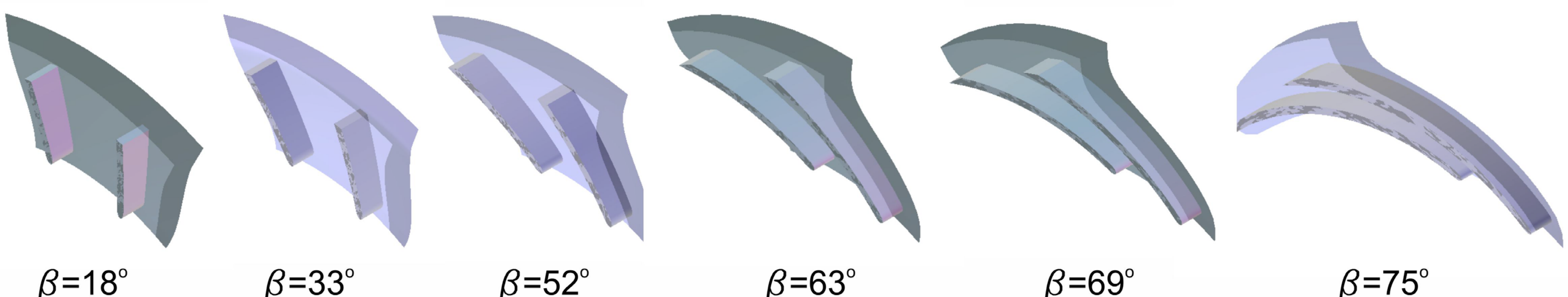
$$D^2=\max\left(S^2,0.09\omega^2\right).$$

$$S_{ij}=\frac{1}{2}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\qquad \Omega_{ij}=\frac{1}{2}\left(\left(\frac{\partial u_i}{\partial x_j}-\frac{\partial u_j}{\partial x_i}\right)+2\varepsilon_{mji}\Omega_m^{rot}\right)$$

Computational model and boundary conditions



Diffuser models with the studied angles

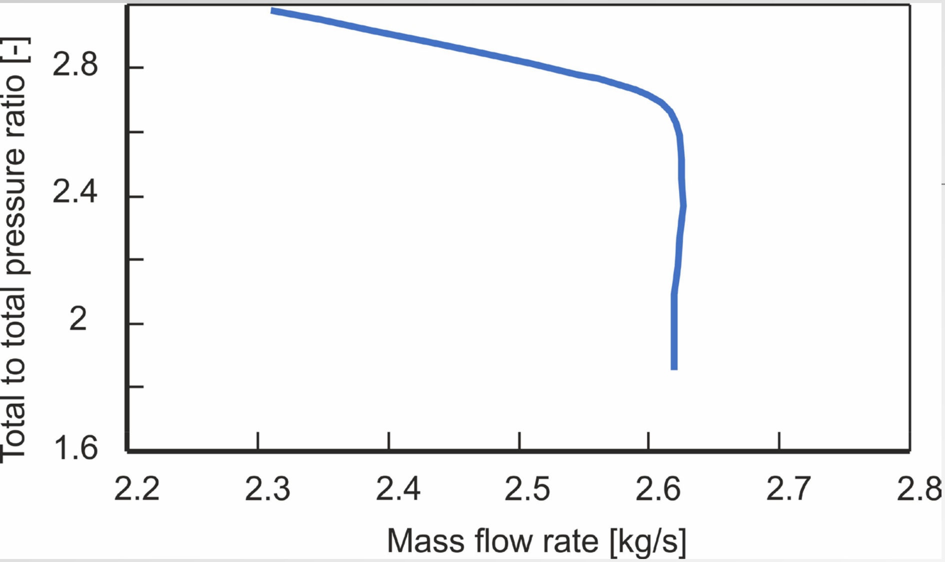


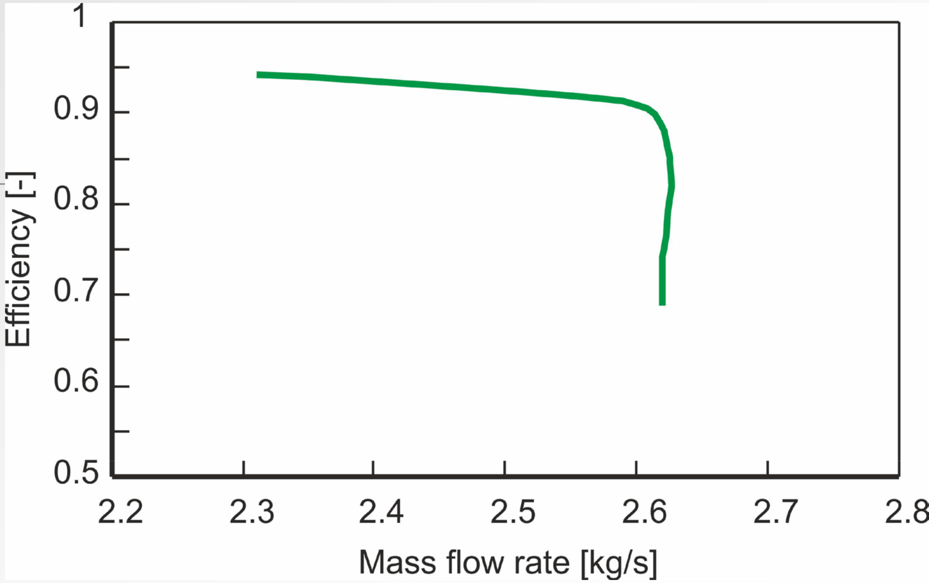
Results

To study the characteristics, the following formulas were used to determine the main performance indicators: the total to total pressure ratio (π^*) and polytropic efficiency (adiabatic compression) (η^*):

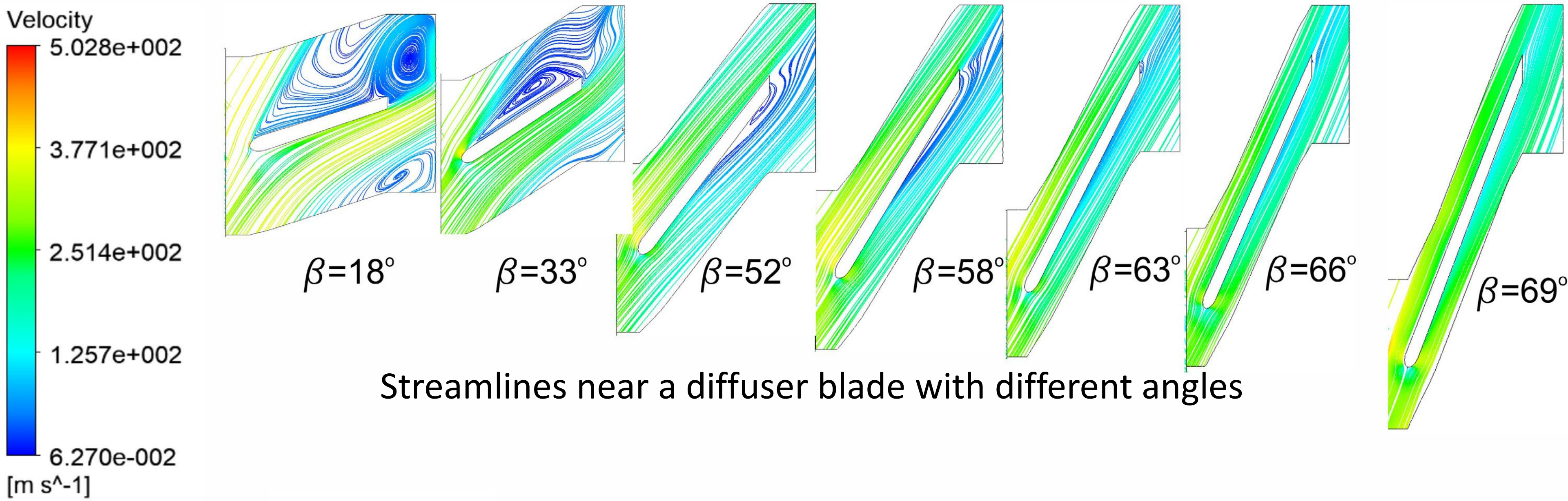
$$\pi^* = \frac{p^*_{out}}{p^*_{in}}, \quad \eta^* = \frac{\ln\left(\pi^*\right)}{\ln\left(T^*_{out} / T^*_{in}\right)} \frac{k-1}{k},$$

where p^*_{out}, p^*_{in} are total outlet and inlet pressures; T^*_{out}, T^*_{in} are total outlet and inlet temperatures; $k = 1,4$ is the dry air adiabatic index.

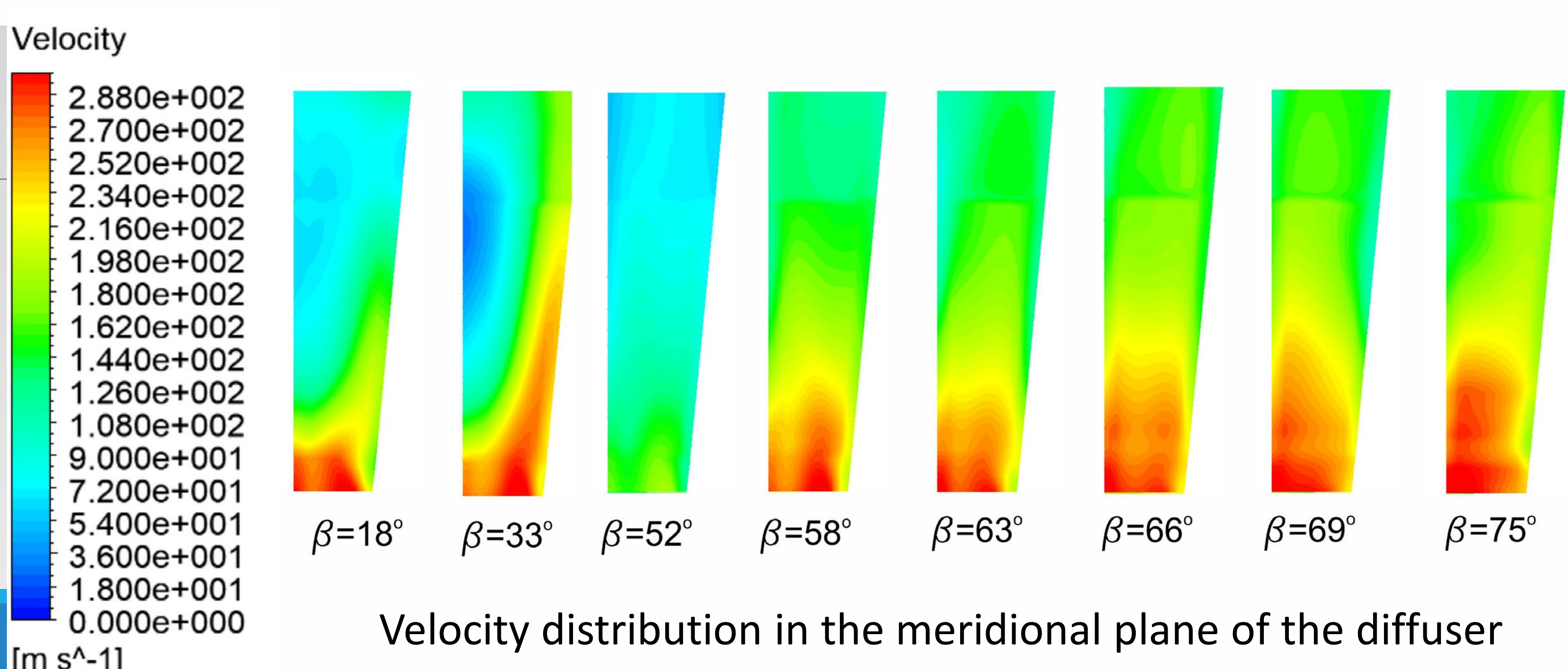




Results



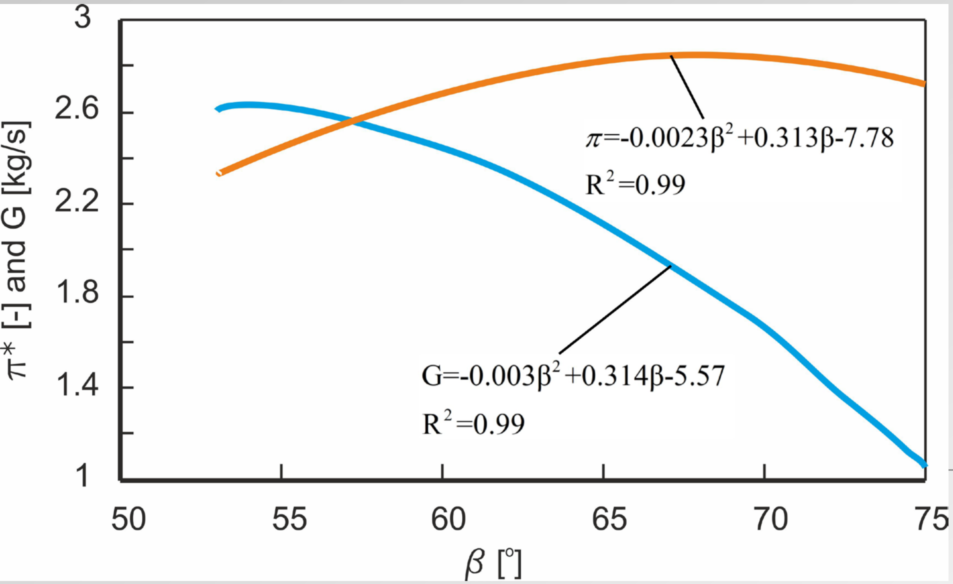
Streamlines near a diffuser blade with different angles



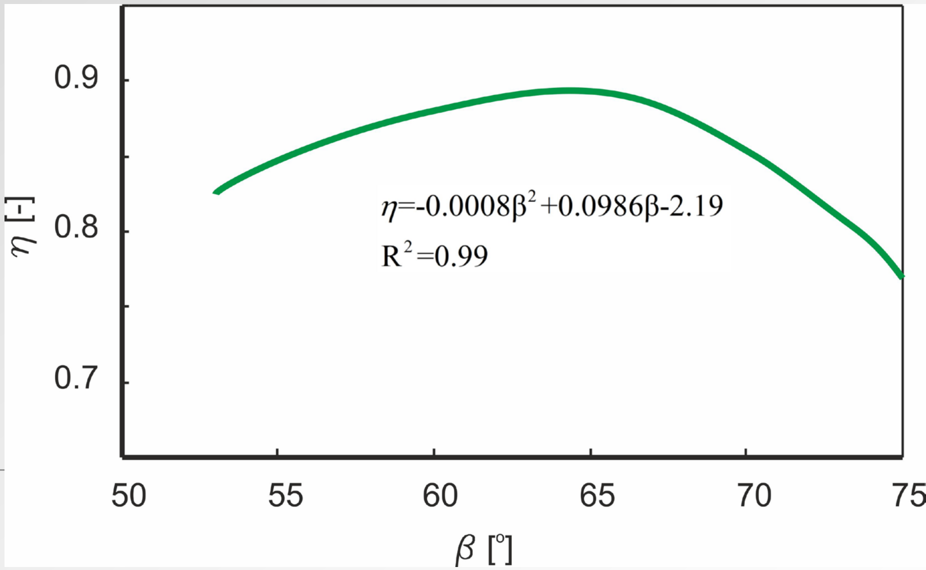
Velocity distribution in the meridional plane of the diffuser

Results

Dependence of performance indicators on the angle of the diffuser blade:



a) pressure ratio and mass flow rate;



b) efficiency

Conclusions

In this study, a vane diffuser for a high-pressure centrifugal compressor is designed and gas flow patterns in diffusers with different blade angles are obtained to determine the optimal inclination angle. The problem is solved based on numerical flow modeling by solving the RANS equations in conjunction with the continuity equation and the SST turbulence model equations. The adequacy of the proposed approach is confirmed by comparing the flow patterns in the model compressor with the results of experimental studies.

The new CAD-designed impeller had an efficiency increase of 5 percent and a pressure ratio decrease of 9 percent. The changed design features of the new impeller required the design of a new radial diffuser.

The analysis of the effect of the vane diffuser blade angle showed that the angle $33° \le \beta \le 52°$ changes the surface of flow separation from the blade, but, subsequently, with an increase in the angle, the separation becomes smaller. At the angles $\beta \ge 58°$, there is almost no separation for a sufficiently large range of angles.

The regression equation for the total pressure ratio, mass flow rate, and compressor efficiency depending on the inclination angle of a radial vane diffuser was obtained. To ensure maximum performance, it is necessary to provide different angles: for the maximum pressure ratio, the angle should be $\beta = 68°$. To ensure maximum mass flow rate in the compressor, the optimal angle is $\beta = 52°$, but at this angle, there is a significant separation of the liquid from the diffuser blade and it cannot be used for rational operation of the compressor. For maximum efficiency, the optimal angle is $\beta = 62°$. As the blade angle increases from a certain value, the mass flow rate at the compressor outlet decreases. At the same time, the inclination angle choice to ensure the optimal pressure ratio will lead to a 2% decrease in efficiency, while the angle to ensure the maximum flow rate will lead to a 6% decrease.

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Thank you for your attention!

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