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Numerical simulation of radial compressor stage

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Abstract. Article describes numerical simulations of air flow in radial compressor stage in NUMECA CFD software. In simulations geometry variants with and without seals are used. During tasks evaluating was observed seals influence on flow field and performance parameters of compressor stage. Also is described CFD results comparison with results from design software based on experimental measurements and monitoring of influence of seals construction on compressor stage efficiency.

1 Introduction

This article is about CFD simulation of radial turbocompressor stage in NUMECA FINE/Turbo software including flowing in labyrinth seals area. The goal was find out influence of centrifugal impeller seals presence in geometry on stage efficiency and pressure ratio. Chosen stage was designed in ČKD KOMPRESORY a. s. company, which is supplier of compressors for technical gases compressing and transport in gas industry, metallurgy, chemical industry and power engineering industry area.

Solved stage is designed for placement into multistage compressor with impellers located between bearings. It is stage with three-dimensional shaped impeller blades (3D impeller), which are generally used in higher mass flow coefficients area.

The simulated stage is designed for middle mass flow coefficients, which creates boundary between using of 2D and 3D impellers. With descending mass flow value through the stage, seals loss to total stage efficiency ratio is increasing. Proportional value of leakage loss is usually from several tenths of percent at stages with high mass flow to ten and more percent at narrow stages with 2D impellers.

Stage is composed from axial guidance device with regulatory blades, centrifugal impeller, vaneless diffuser and return channel. Axial guidance device with turning blades is used for regulation in the radial turbocompressor intake.

2 Geometrical model

Basic geometrical model of solved radial compressor stage consists of axial guidance device channel, centrifugal impeller channel, vaneless diffuser and return channel. In the axial guidance device is nine turning blades, which are able to regulate medium mass flow through the compressor. In this case working medium is air and regulatory blades are in zero position.

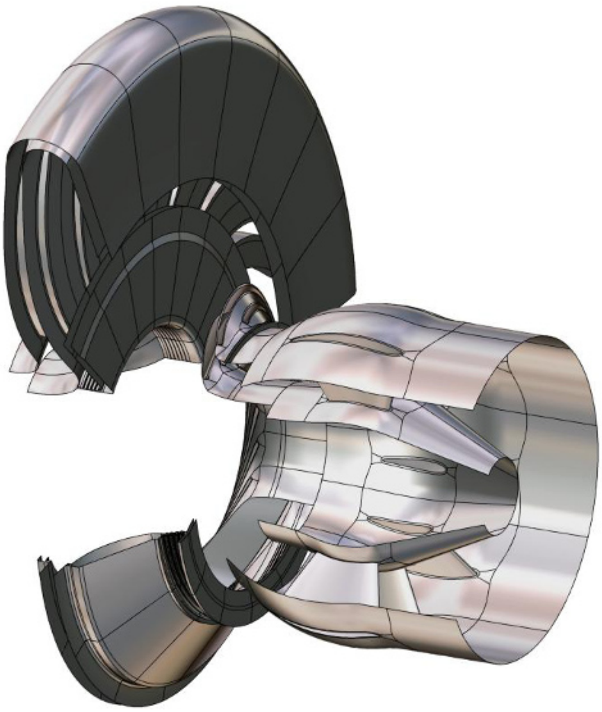

Fig. 1. Geometrical model of radial compressor

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Behind the guidance device the impeller inlet is located. Impeller has fifteen 3D shaped blades. At the impeller outlet is the inlet of diffuser channel. In this case diffuser is vaneless, but in some design solutions vanes are placed in diffuser because of better flow direction of pressurized medium to return channel. In the return channel is twenty-two vanes.

Fig. 2. Centrifugal impeller

For purpose of finding out of seals influence on compressor stage power parametres geometrical model obtains impeller shroud and hub seals. In the case of shroud, seal is stepped with five edges. In the case of hub, seal is straight also with five edges.

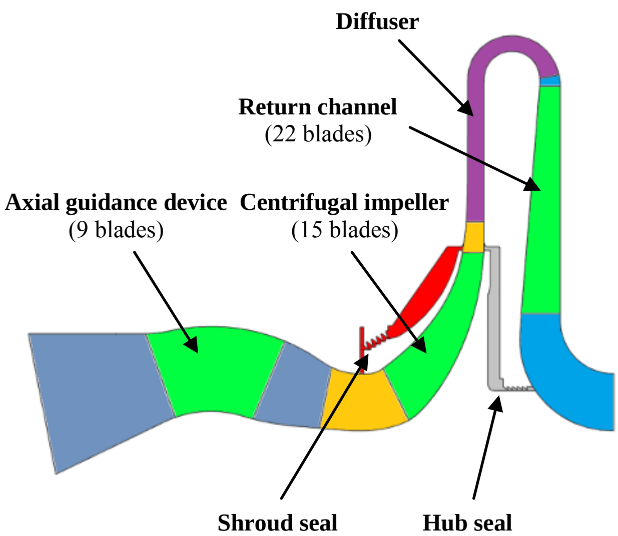

Fig. 3. Compressor scheme

3 Grid and boundary conditions

Computational grid was created in NUMECA TurboGrid 8.9-3 software. Grid is composed only of hexaedral cells and is block structured because of computational time reduction. For obtaining undistorted results is necessary to keep the grid fine enough and also is necessary to keep certain grid cell properties, which are orthogonality (> 20), expansion factor (< 3) or aspect ratio (< 3000).

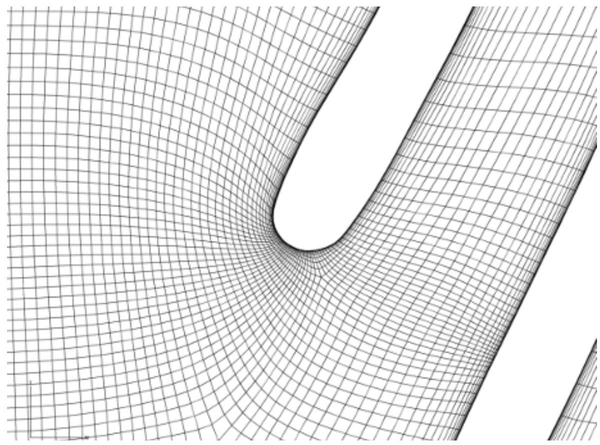

Fig. 4. Detail of impeller blades grid

Base computational grid variant without seals consists of 2 800 000 hexaedral cells and variant with seals consists of 4 900 000 cells. Grid structure in channel area is in both variants identical because of the best possible results comparison. Thanks to very small height of cells at the walls (0.005 mm) was possible to reach maximal y+ around 10 in all solved computations.

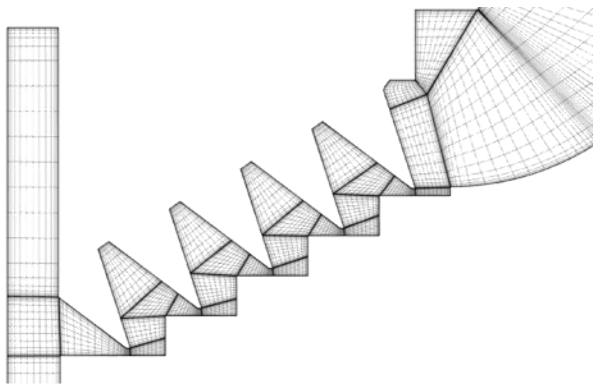

Fig. 5. Detail of stepped seal grid

Stage was modelled with pressure inlet boundary condition on axial guidance device inlet and mass flow outlet boundary condition on outlet from return channel. Boundary between specific functional model areas are defined as mixing plane interface. Thanks to that we were able to reduce number of blades to one in each compressor stage area. Pressurized medium is in our case air. Solved compressor stage works with quite small mass flow and pressure ratio, so it is possible consider medium as ideal compressible gas. SST k-ω turbulence model was used and cases were computed stationary.

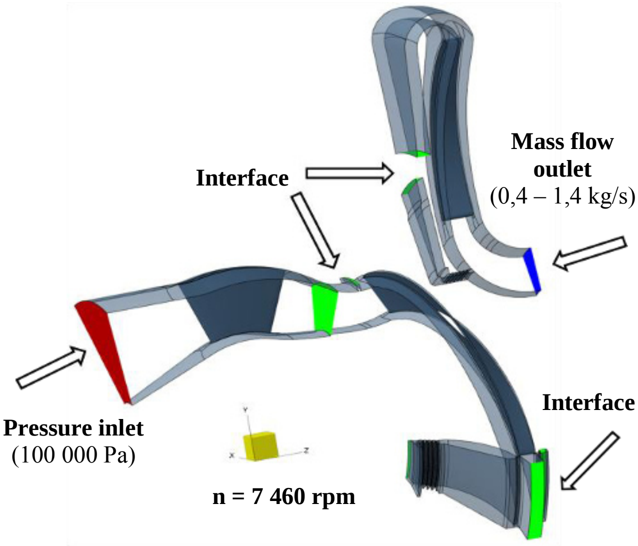

Fig. 6. Boundary conditions

4 CFD simulations

To determine radial compressor stage power curves case was solved in six working points per variant without and with the seals. We got large part of power curves, which is possible to find basic compressor stage properties from. During results evaluating were the most important parameters stage efficiency, pressure ratio and total axial force acting on centrifugal impeller. Next we evaluated flow field character in the each compressor stage domain and mass flows through impeller shroud and hub seals were compared with ČKD design software.

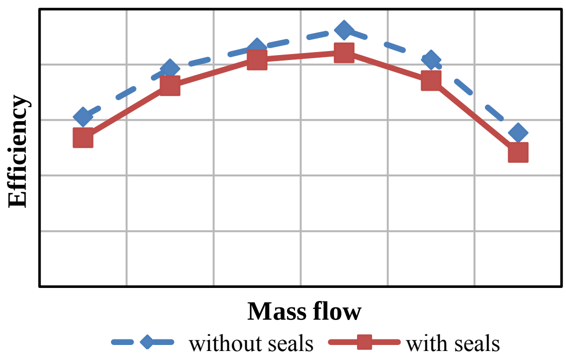

Fig. 7. Compressor stage efficiency

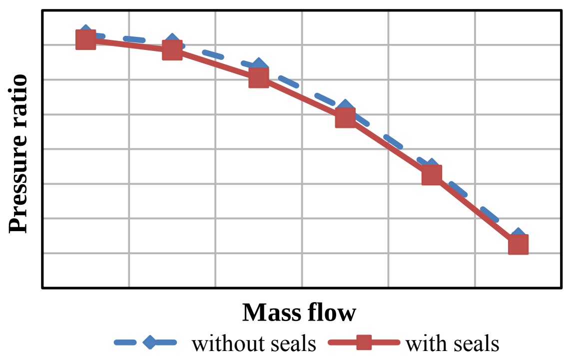

Fig. 8. Compressor stage pressure ratio

As was expected, seals added into compressor stage model had influence on results. In the case without seals was reached higher pressure ratio and compressor stage efficiency too. Difference between values of efficienty is at all working points around 2%.

After evaluation of CFD computations were mass flows through both seals compared with ČKD design software (KSTK), which is adjusted according to earlier ČKD computations and measurements. We achieved positive results, where computed mass flow through straight seal of hub was nearly the same as mass flow obtained from design software. In the shroud seal is situation worse, but it could be caused by using uncertain constants and this work gives data for updating them. Flow character in stepped seals is more complicated than in straight seals.

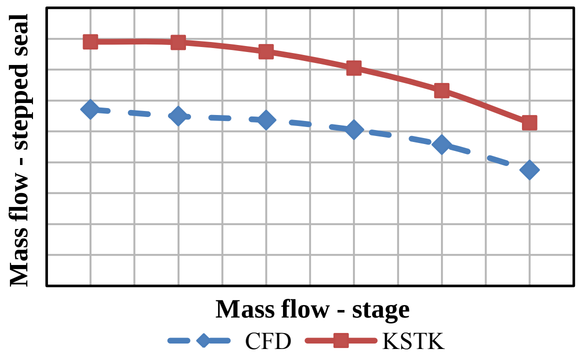

Fig. 9. Mass flow through stepped seal

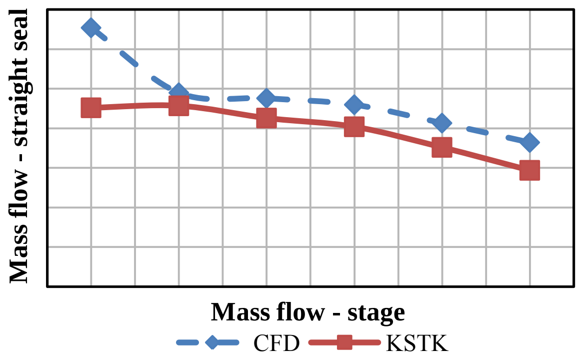

Fig. 10. Mass flow through straight seal

5 Axial force

Very important part, which has an influence on design and construction solution of compressor and other blade machines, is total axial force on centrifugal impeller. Axial force magnitude and direction is necessary to know because of correct design of compressor shaft bearings. Evaluation of axial force is possible only at geometrical variant with impeller seals.

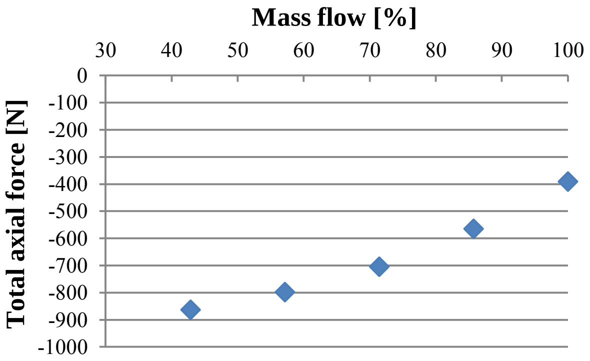

Fig. 11. Total axial force acting on impeller

Our simulation shows that total axial force acts in the direction of the compressor intake (against inlet flow direction) and according to expectation the force is growing with decreasing mass flow.

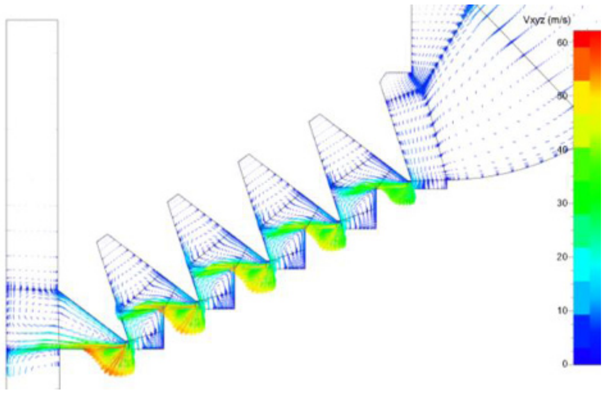

Fig. 12. Velocity field in stepped seal

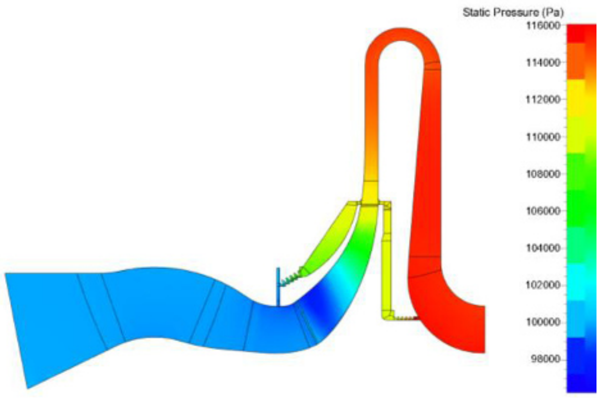

Fig. 13. Static pressure field

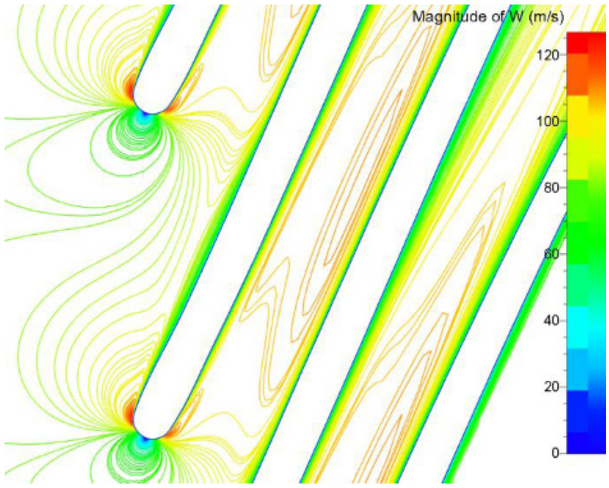

Fig. 14. Velocity field between impeller blades

6 Conclusion

The main purpose of this computation was to find out influence of radial stage impeller seals on stage power parameters. As emerged from results analysis seals are decreasing stage efficiency in all working point around 2% against geometrical variant without seals. Similar decrease is showing at stage pressure ratio. This knowledge is usefull during next design or optimalization of this compressor stage. Also good accordance with ČKD design software is showing at the mass flow through seals area.

For the next computations stage with higher working parameters will be used. If coresponding seals influence and accordance with ČKD design software will be proved, new space for geometry optimalization will open in order to increase compressor stage efficiency.

Acknowledgments

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