

Numerical modelling and experimental investigation of the influence of selected geometric elements on the working characteristics of radial compressor stages with prismatic blades

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Abstract. The article deals with a description of basic results of research of radial compressor stages with 2D e.g. prismatic rotor blades. The experimental facility and the measurement and evaluation process are described briefly in the first part. The second part of this contribution is the numerical simulation and evaluation of the influence of the technological holes in the 2D rotor blades to compressor stage characteristics. The influence of the labyrinth seals clearance on the working characteristics can be found in the last part. The comparison of measured and computed values is presented and discussed.

1 Introduction

Centrifugal compressors are widely used in the industry. The description of processes, operations and performance of the centrifugal compressors are described in many publications, for example in [1].

The verification of the new stages properties requires detailed measurement of performance characteristics to obtain reliable data for compressor design. Measured data are also important for validation of CFD numerical models. Validated numerical models can give also the complex information about the flow in the stage and help to design the channel profile.

The article authors participated on the large research and development project of new compressor stages design. The main goal of the project was to achieve higher stage efficiency and better operational properties. In addition to the basic working characteristics of the investigated stages a certain amount of supplementary information was acquired. This work briefly presents the influence of some geometric elements on the characteristics of stages with prismatic blading.

2 Research facilities

2.1 Experimental rig

The described development of the centrifugal compressor stages was a continuation of the systematic research and long term development program; see for example the paper [2]. Experimental measurements were realized in the development test facility Howden ČKD

Compressors Ltd. in Pilsen from 2014 to 2018, where the single-stage test compressor DARINA IV with the exchangeable flow part was placed. The compressor consisted of an inlet cone, an impeller, a diffuser, a return bend, a de-swirl channel and an exit part.

The compressor was driven by an electromotor with maximum input wattage 1.2 MW with frequency converter for a continuous speed control. The test rig is shown in Fig. 1.

The test rig was designed for impellers with diameter of 440 mm and for rotation speed from 4,000 to 18,000 rpm. A set of throttle-valves for the aerodynamic resistance adjustment was placed in the discharge piping. Mass-flow rates of the air through the suction and discharge piping were measured by orifices.

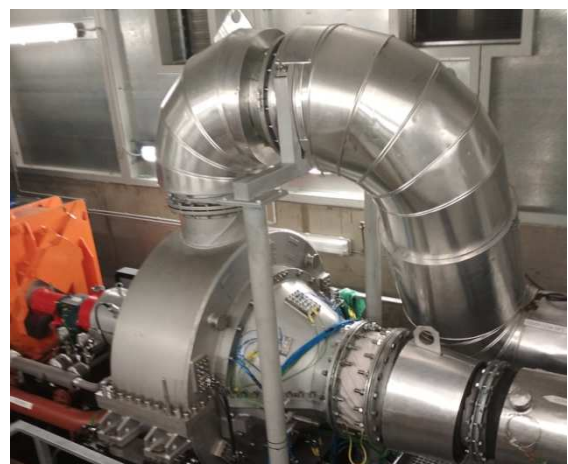


Fig. 1. Photo of the test rig for centrifugal compressor stages.

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2.2 Experimental equipment

Parameters of the measured stage were obtained by the complex measurement and data acquisition system.

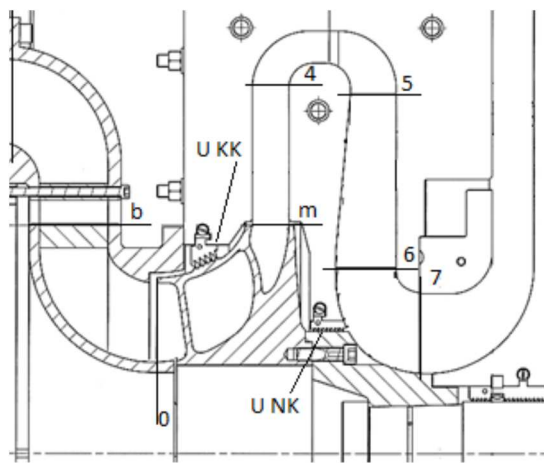


Fig. 2. The cross-section of the compressor with evaluated planes.

The flow parameters are measured in large number of points on the evaluation planes. The labels of the planes in the stage are long-term used, see [2], and are depicted in Fig. 2; the main evaluation planes or so called measuring stations for determination of the stage parameters are the stations 0 and 7. There is a number of static and total pressure and total temperature probes (multiple sensors are placed in the stations 0 and 7). The torquemeter gives the value of torque and rotating speed of the compressor rotor.

Described research was carried out with stages RTK06 and RTK04 with various blade lengths which are designed for medium to lower flow rates. The impellers and stator parts of the stages RTK06W and RTK04N are depicted in Fig. 2.

2.3 Evaluation of measurement

The measurement campaigns are established to produce curves of operational characteristics – branches of the compressor map. The whole process of the measurement and evaluation is described in detail in the paper [3].

The aim is to measure one operational curve keeping the reference (blade) Mach number (1) constant. This number is based on the speed of sound in the gas at stage intake state and the velocity of the blades at the outer diameter of the impeller:

$$Ma_{2u} = \frac{2 \cdot \pi \cdot \Omega \cdot D_2}{a_0} = \frac{u_2}{a_0} \quad (1)$$

This means the rotational speed is being adjusted slightly based on the actual intake conditions.

Besides the whole stage evaluation, a part-by-part analysis can be carried out using the data acquired from the measuring stations visible in Fig. 2. Mean velocity vectors, performance characteristics of the diffuser or the losses of the de-swirl channel could be addressed.

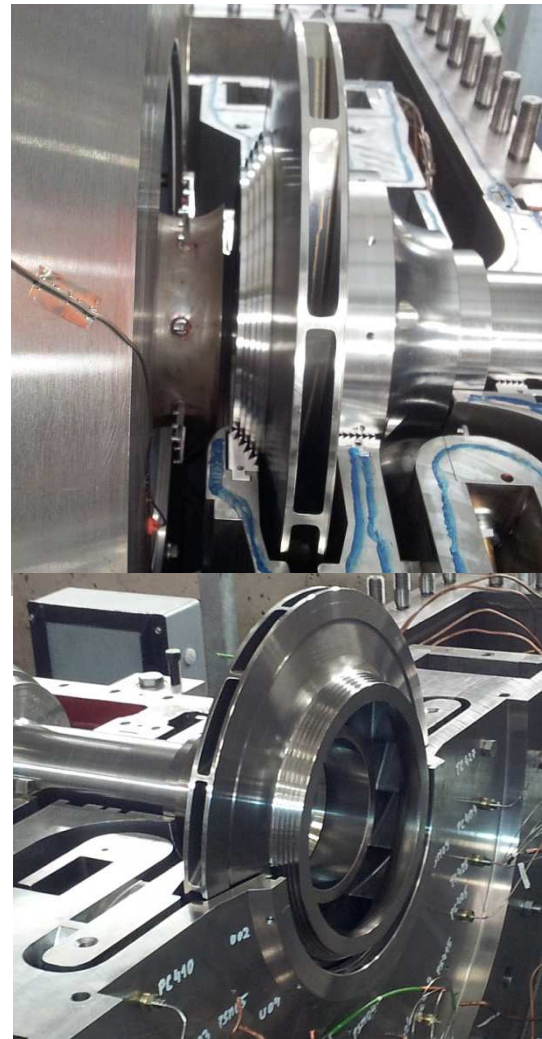


Fig. 3. Models of the stages RTK06W and RTK04N in the test compressor.

2.4 Parameters of stages

Performance of the centrifugal compressor stages are analysed through two of the main properties: the pressure ratio and the efficiency. These two parameters vary with the mass flow-rate and with the Mach number.

The evaluation of various compressor stage parameters was done according to the formulas (2) to (6). Values of the evaluated parameters were averaged by the mass flow rate in postprocessing. The parameters are: specific total work (2), specific isentropic work (3), pressure ratio (4), isentropic efficiency (5) and flow coefficient (6).

$$\Delta h_t = c_p \cdot (T_{t7} - T_{t0}) \quad (2)$$

$$\Delta h_{is} = r \cdot T_{t0} \cdot \frac{\kappa}{\kappa - 1} \cdot \left[\left(\frac{p_{t7}}{p_{t0}} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \quad (3)$$

$$\Pi = \frac{p_{t7}}{p_{t0}} \quad (4)$$

$$\eta = \frac{\Delta h_{is}}{\Delta h_t} \quad (5)$$

$$\varphi_0 = \frac{4 \cdot Q_0}{\pi \cdot D_2^2 \cdot u_2} \quad (6)$$

The results from stages CFD simulations were compared to the measured data converted to the standard inlet conditions. Obtained characteristics include the confidential technical data and only relative values are presented in the following sections. The values PI_N , ETA_N and m_N are normalized by the design point values in the measured characteristic.

3. CFD software and models

CFD simulations are nowadays an essential tool for the design of all types of devices in turbomachinery. They enable engineers to gain a lot of time between the concept and the test. The verification of simulations is carried out in all turbomachinery development groups. For the purpose of this work, many CFD simulations of various compressor stages were run in the NUMECA FINE/Turbo (NFT) and ANSYS CFX (ACFX) software environments.

The methodology of CFD simulations for the RTK stages and properties of the model has been evaluated and improved; see e.g. [3, 4, 5, 6]. The air, modelled as perfect gas, was used as the flow medium, $k-\omega$ SST turbulence model was used for the described simulations.

4 Simulations of the technological holes influence for a 2D compressor stage

4.1 Technological holes description

The problem of the technological holes for the stages with 3D rotor blades was described in the contribution [3]. The manufacturing technology of the stages with prismatic blading is different. Rotors are split into two parts and it is possible to machine the parts on a simple milling machine. Both parts are then welded or soldered together. The soldering process gives better quality thanks to low thermal stress introduction and better surface quality of the joint, but the welding is often cheaper and preferable.



Fig. 4. An example of a technological hole.

The welding of impellers requires the technological holes in the blade to reduce mechanical stress concentration at the end of the weld, see Fig. 4. The splitting plane is located near the middle of the blade height; the weld takes place in the aft part – from the hole to the trailing edge. The goal is to find the suitable location of the hole depending on the blade type and required stage efficiency. The stage RTK04W was selected for the case study.

4.2 Model of the holes

The model of the stage with holes was based on the standard stage model described in the paper [6]. The stage dimensions correspond to the stage simulated and experimentally tested.

The model hole position was varied to five positions along the blade length marked from 1 to 5 and to five positions along the blade height marked from A to E. The positions of the holes in the model are shown in Fig. 5. Such setup yields 25 simulation sets to produce performance curves at the constant rotational speed.

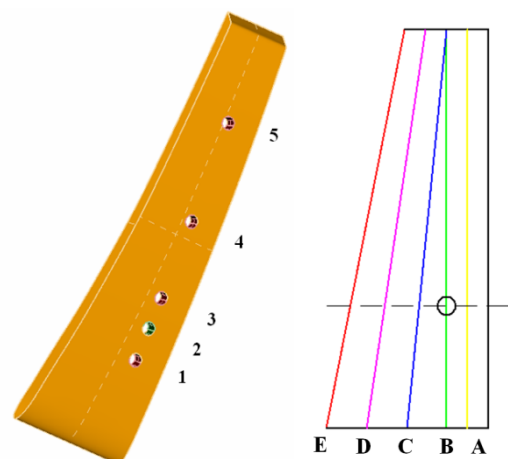


Fig. 5. Positions of holes in investigated variants.

The diameter of the circular holes was set to 4 mm except the case E, where the shape was semi-circular and the diameter was set to 5.6 mm - the cross-sectional area is then similar.

4.3 Results of the simulations

The results of investigated variants were evaluated in the same manner as for the stage without holes. Fig. 6 illustrates the pressure field on the blade surface and the contours of the relative Mach number in the slice through the hole in the blade.

The Fig. 7 shows the influence of differently positioned hole to the pressure. The results for positions along lines not shown (A, C, D) have been very similar to the positions along line B. The effect of the holes on the pressure and velocity field in the stage was assessed to have just a “local influence”. The most significant influence of the holes was encountered for positions along line B and E.

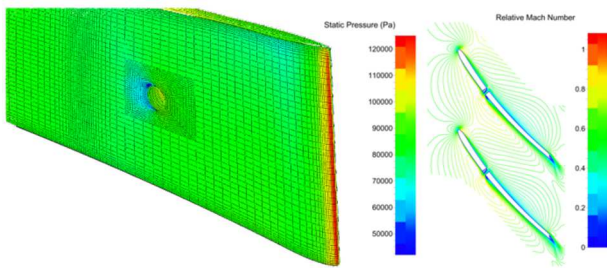


Fig. 6. Surface mesh and static pressure (left) and the relative Mach number (right).

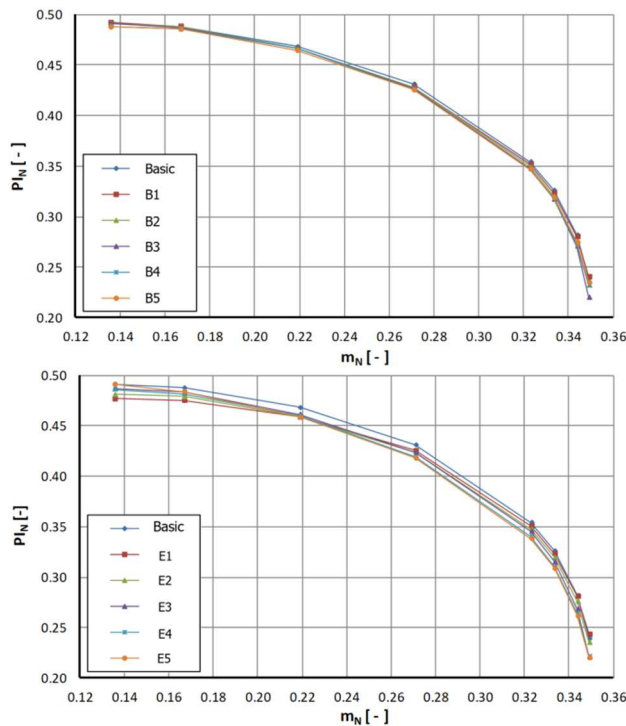


Fig. 7. The influence of the hole to the pressure ratio.

The influence on the efficiency is more important for the stage design - a selection of results for rotational speed of $12,666 \text{ rpm}$ ($Ma_{2u} = 0.8$) is shown in Fig. 8. It was concluded that the influence of the hole position is observable but not significant. The highest negative influence on the working characteristics is seen in the case of technological holes located on the shroud (line E) and the efficiency decreases as the hole is moved towards the shroud.

The holes are usually positioned on lines B or C and in positions 1 or 2, where significant drop of efficiency is not predicted by the presented results. The influence on the efficiency for the design conditions flow-rates (noted by black dots in Fig. 8) is from 0.2 to 0.8% compared to the design with no hole.

5 Influence of the reduced clearance of labyrinth seals to the stage working characteristics

5.1 Description of the problem

The seals inside the compressor create the resistance to flow between volumes at different pressures. Since one

of the walls bounding this volume of fluid rotates at a high peripheral speed, the use of contact seals is excluded. Contactless seals always allow a certain adverse flow that negatively affects the machine efficiency. The labyrinth seals are typically used. The Fig. 9 shows examples of seals used in radial compressors – a stepped labyrinth for impeller shroud disc and a straight-through labyrinth for sealing the shaft-diaphragm gap.

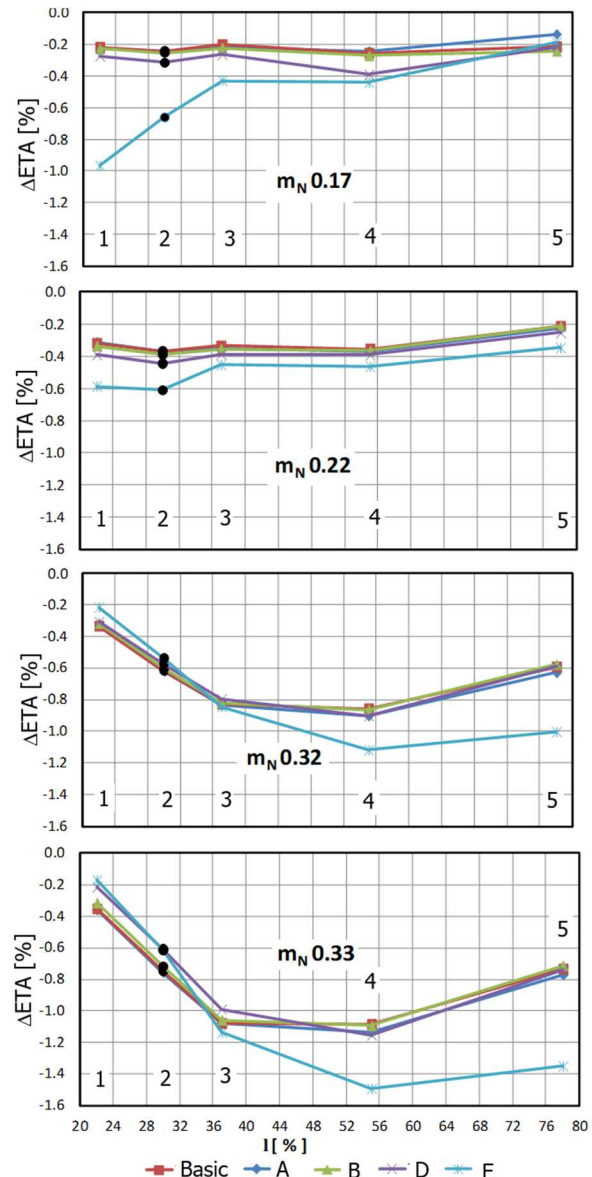


Fig. 8. The influence of the hole on the efficiency of the stage.

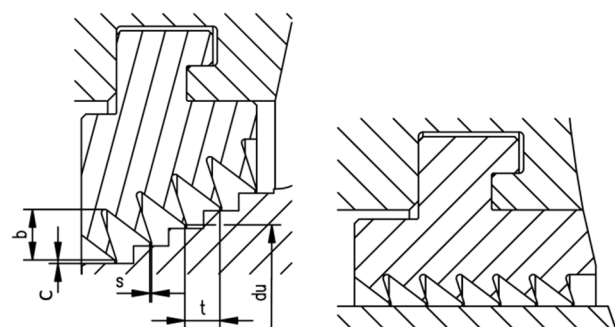


Fig. 9. The labyrinth seals for a radial compressor stage.

The basic parameter of the labyrinth is the width of clearance c , see left part of Fig. 9, which influences the leakage flow rate in the seal and consequently raises the losses in the stage.

A research study was conducted to determine the influence of reduced clearance to the working characteristics of the standard stage RTK06W (without holes in blades). It is published in [6]. Both experimental measurements and numerical simulations were used for the analysis. In this chapter, the analysis is focused on both seals in the stage and their equally large clearance adjusted together. The location of the seals is denoted by the flags 'U KK' and 'U NK' in Fig. 2.

The study was conducted in the environment of an industrial research process, which is primarily concentrated on producing compressor maps for various stages and a wide range of reference Mach numbers. Due to the stiffness and tight schedule of the process, the analysis of the seal leakage were secondary and certain trade-offs were necessary. That is why the results of measurements were only interpolated to the design speed of the stage RTK06W. On the other hand, CFD is not so expensive to prepare, but computing many operating regimes may become prohibitive when including complexities, like seals, in the cases.

5.2 Experimental measurement

The experiments were focused on three typical reference Mach numbers of the stage - $Ma_{2u} = 0.7, 0.9$ and 1.1 (10,550, 13,568 and 16,377 rpm). Two repeats of measurements with classical aluminium alloy seals (AL) and one or two runs with plastic seals (PL) with significantly smaller clearance c were administered. The resulting curves of both aluminium seals measurements were practically identical.

The standard clearance of the alloy seals is from 0.4 to 0.5 mm. Both cases are analysed, while 0.4 mm clearance is taken as a norm. The plastic seals allow the clearance to be lowered down to 0.15 mm.

Fig. 10 shows the whole measured pressure characteristic of the stage and the limits of practical operating range. These limits are imposed upon Fig. 11, which shows the efficiency characteristic at $Ma_{2u} = 0.9$.

The Table 1 shows the averaged efficiency rise in the operating range for the stage with plastic seals based on polynomial regressions from Fig. 11. The average increase of efficiency $\Delta\eta$ was 1.28 %. For $Ma_{2u} = 0.7$ and 1.1 the efficiency increased on average by 0.81 and 1.21 % respectively. The influence was larger at lower compressor mass-flow rates, as also seen in Table 1.

Table 1. Increase of the efficiency for $Ma_{2u} = 0.9$.

m_N [-]	$\Delta\eta$ [%]
0.17	1.20
0.16	1.22
0.14	1.26
0.13	1.33
0.11	1.42

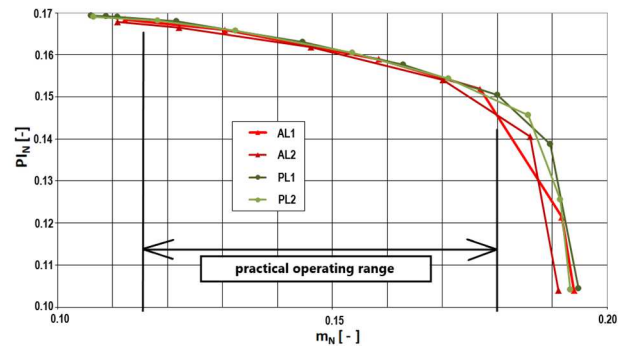


Fig. 10. Measured pressure characteristics of the stage for 13 568 rpm.

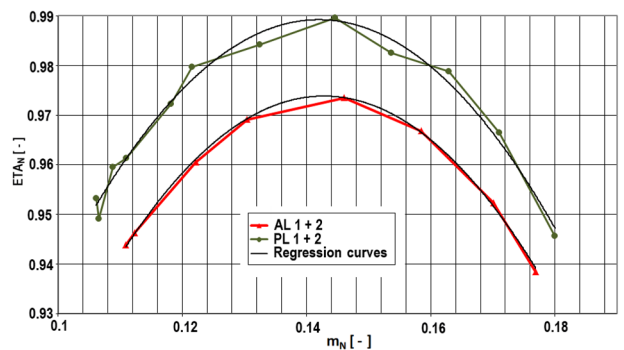


Fig. 11. Efficiency characteristics in the practical operating range of the stage at $Ma_{2u} = 0.9$. Polynomial regression curves added.

5.3 Computational model of the stage

The model of the stage with holes was based again on the standard stage model described in the paper [6]. The conditions of the CFD study were slightly different to the measurement because only cases for the design speed of 11,921 rpm ($Ma_{2u} = 0.8$) were run.

A set of simulations was prepared considering various seal clearance. The value of clearance was set to $0.1, 0.25, 0.4$ and 0.5 mm, always equal for all the labyrinth teeth.

In Fig. 12 and 13 the working characteristics can be observed. An obvious but limited impact to the pressure and also efficiency characteristics is deduced.

The violet dashed line in Fig. 12 stands for the standard seal design (0.4 mm). Here, the complete range of the pressure characteristic curve is drawn, including also the right-most point where the stage is choked. This point is excluded from the efficiency curve in Fig. 13.

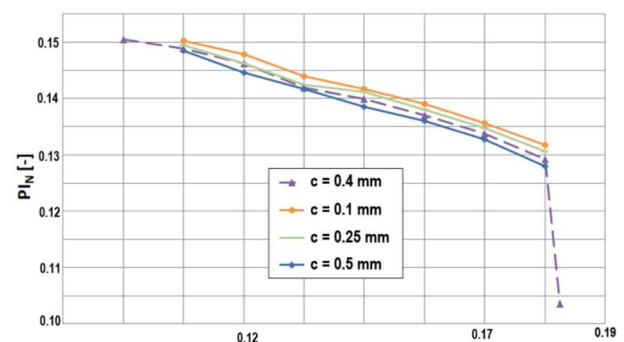


Fig. 12. Computed pressure characteristics for various width of seal clearance.

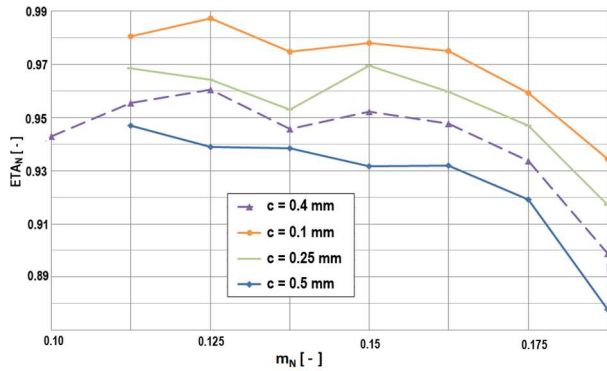


Fig. 13. Influence of the clearance width to the efficiency of the stage.

5.4 Comparison of results

Table 2 shows the averaged departure of efficiency caused by varying clearance width. The efficiency with a clearance of 0.4 mm is taken as a nominal value.

Table 2. Deviations of the efficiency.

clearance [mm]	ΔETA [%]
0.4	0
0.1	2.83
0.25	1.25
0.5	-1.54

The values of the efficiency from CFD simulations show very strong dependence on the clearance width. A linear interpolation of presented CFD results for a clearance of 0.15 mm yields an increase of the efficiency by 2.3% .

When revisiting this guess using experimental data, one needs to interpolate between the data for $Ma_{2u} = 0.7$ and 0.9 to obtain the guess for the desired $Ma_{2u} = 0.8$. Interpolating the data presented in paragraph 5.2, the increase of the efficiency for the case of 0.4 mm clearance width and $Ma_{2u} = 0.8$, the efficiency departure reaches slightly above 1% .

There could be many reasons for such a large difference (2.3% vs. 1%). One of them can be the contact of PL seals with the rotor, which could and did occur due to the large compressor shaft bearing clearance (around 0.2 mm). This contact could have caused a growth of the seal clearance to values of approximately $0.2 - 0.3\text{ mm}$. When the interpolation of CFD results is repeated for such clearance width, a value of 1.25% efficiency increase could be expected. Unfortunately, it was not possible to measure the exact value of the seal clearance after the experiment due to technical problems.

6 Influence of an increased clearance of labyrinth seals to the stage parameters and the flow-rate through the seal

6.1 Description of the problem

One of the last research projects based on the experimental compressor DARINA IV was carried out with the RTK04N stage. The goal was to assess the influence of an increased clearance of the stepped labyrinth seal on the outer side of the impeller shroud disc. The project was primarily focused on the leakage flow itself and the measured characteristics were just a supplement to it. Detailed study supported by CFD can be found in [7].

Unlike the study described in Chapter 5, only the shroud disc seal was modified. The clearance width was gradually increased from the basic value $c = 0.4\text{ mm}$ to the values 0.6 mm and 0.8 mm . Again, an equal clearance was set for all the teeth of the seal.

6.2 Experimental results

In Fig. 14 you can see the measured pressure characteristics for the mentioned values of clearance at $Ma_{2u} = 0.9$ ($13,568\text{ rpm}$). There is clear evidence that the influence of the clearance width to the pressure characteristic is non-linear. A quadratic dependence is expected as the leakage flow cross-section grows with the clearance width to the power of 2.

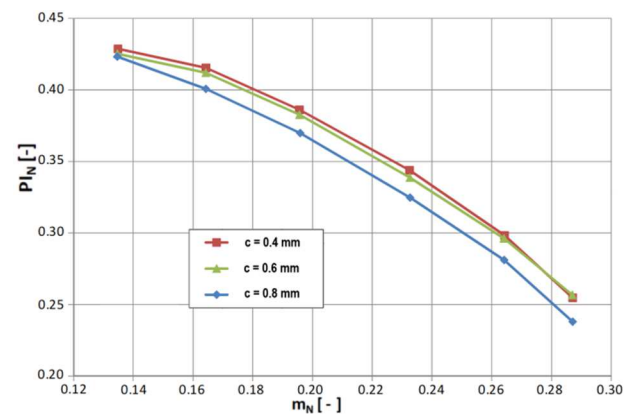


Fig. 14. Measured pressure characteristics of the stage with various clearance width of the impeller shroud disc seal at $Ma_{2u} = 0.9$.

The decrease of efficiency for the same set of cases is shown in Table 3, where m_N stands for an arbitrarily normalized compressor flow rate. The differences are higher for higher flow-rates. This phenomenon was already observed and described in chapter 5.2 for the stage RTK06W.

The study also took care to measure the leakage influences at various rotational speeds. Besides already mentioned $Ma_{2u} = 0.9$, values of 0.7 and 1.1 were set to be measured too. The decrements of the efficiency, averaged over all of the compressor flow rates for given speed, are presented in Table 4. The efficiency with standard clearance of $c = 0.4 \text{ mm}$ is taken as a norm.

Table 3. Decrease of the efficiency at $Ma_{2u} = 0.9$ for higher values of clearance width.

m_N [-]	$\Delta\eta$ [%]	
	$c = 0.6$ [mm]	$c = 0.8$ [mm]
0.6	-0.4	-0.31
0.67	-0.56	-0.92
0.75	-0.63	-1.21
0.88	-	-1.4

The values for $Ma_{2u} = 0.7$ and 0.9 are very close to each other for a given clearance width. The highest speed, however, presents significantly larger decrease of efficiency, which seems not to vary a lot with an increase of the clearance width.

Table 4. Efficiency decrements averaged over compressor flow rates at different rotational speeds.

Ma_{2u} [1]	average $\Delta\eta$ [%]	
	$c = 0.6$ [mm]	$c = 0.8$ [mm]
0.7	-0.6	-1.05
0.9	-0.53	-0.96
1.1	-1.26	-1.20

7 Conclusions

The paper describes three effects of the selected geometric elements on the working characteristics of the radial compressor stages. The effects were solved in various research projects related to the development of the new family of radial compressor stages.

At first, a CFD study shows the influence of the technological holes in blades of welded impellers. The predicted influence to the efficiency of the stage is about 0.5 % of the overall efficiency. The results also show that the locations of the holes typically used by the structural designers correspond to the locations with low influence to the stage working characteristics.

The second complex study shows the influence of a concurrent reduction of the clearance width in both of the labyrinth seals present in the stage. The practical issues obstruct direct comparison between experimental measurement and numerical simulation. Despite that, the methods used for comparison do not pose a significant uncertainty to the results. The trends can still be deduced clearly. The uncertainties connected with both experimental and computational methods themselves would still be high above negligible.

The third study is just a complement of the previous. It supplies experimental results for the higher values of the clearance in the impeller shroud disc labyrinth. The original study is more complex than presented here. A detailed analysis of the leakage flow in the stepped labyrinth is to be published in a subsequent contribution.

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Nomenclature

a	Speed of sound
c	Clearance of the leakage
c_p	Isobaric specific heat capacity
D	Impeller outer diameter
η	Isentropic efficiency
FIO	Flow coefficient for intake volumetric flowrate
l	Hole position measured from blade tip
m	Mass flow rate
M_{2u}	Blade Mach number
PI	Total pressure ratio
p	Total pressure
r	Specific gas constant
T	Total temperature
u_2	Blade speed at impeller exit
Δh	Specific enthalpy difference
κ	Ratio of specific heats
Ω	Rotational speed of the impeller

Common indices:

0	Measuring station 0 (stage intake)
2	Impeller exit
7	Measuring station 7 (stage discharge)
is	Isentropic
N	Normalized
t	Total

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