Turbomachine Design for Supercritical Carbon Dioxide Within the sCO2-HeRo.eu Project

The paper aims to give an overview over the keystones of design of the turbomachine for a supercritical CO₂ (sCO₂) Brayton cycle. The described turbomachine is developed as part of a demonstration cycle on a laboratory scale with a low through flow. Therefore, the turbomachine is small and operates at high rotational speed. To give an overview on the development, the paper is divided into two parts regarding the aerodynamic and mechanical design. The aerodynamic design includes a detailed description on the steps from choosing an appropriate rotational speed to the design of the compressor impeller. For setting the rotational speed, the expected high windage losses are evaluated considering the reachable efficiencies of the compressor. The final impeller design includes a description of the blading development together with the final geometry parameters and calculated performance. The mechanical analysis shows the important considerations for building a turbomachine with integrated design of the three major components: turbine, alternator, and compressor (TAC). It includes different manufacturing techniques of the impellers, the bearing strategy, the sealing components, and the cooling of the generator utilizing the compressor leakage. Concluding the final design of the TAC is shown and future work on the machine is introduced. [DOI: 10.1115/1.4040861]

Introduction

Against the background of the changed energy market, the supercritical carbon dioxide cycle has attracted considerable attention due to its compactness and proposed thermodynamic efficiency. Therefore, several companies and institutes are working in the field of development of sCO₂ turbomachinery. In general, the main focus is to develop a more economically efficient turbomachine for the production of electricity in a power plant or for shipboard propulsion. Contrary to the production of electricity, the main objective of the turbomachine (TAC), consisting of turbine, alternator and compressor, described within this paper is to propel a closed, unrecuperated Brayton cycle for transportation of residual decay heat from a nuclear reactor core to an ultimate heat sink (Benra et al. [1]). However, the challenges in design are similar, especially because the compactness of the turbomachine becomes an issue in a size of a small demonstrator.

The turbomachine is developed as part of the supercritical CO₂ heat removal system (sCO2-HeRo) project, which aims to develop and proof the concept of a new self-launching, self-propelling, and self-sustaining safety system for nuclear power plants (Benra et al. [1]). After a station blackout, the system transports the residual decay heat of the nuclear reactor core to an ultimate heat sink. The system can also be retrofitted to existing power plants. Unlike other safety systems, it does not need external energy supply, but it even produces electrical energy. To have a compact turbomachine and compact heat exchangers and therefore be able to retrofit the system to existing power plants, the closed unrecuperated Brayton cycle is realized using sCO₂ as working fluid. In the sCO₂-HeRo project, a demonstrator unit is developed and experimentally tested at the Gesellschaft für Simulatorschulung (GfS) in order to prove the concept and assess technology readiness level 3. Furthermore, the cycle shall be used to gain experience on the design, performance, and operation of sCO₂ loops and the consisting components. Table 1 contains the thermodynamic design parameters for the TAC, which relate to the use of the sCO₂-HeRo system as a demonstrator for a safety system to remove residual heat in nuclear power plants. The application in the safety cycle raises requirements regarding the stable operation, also at off-design conditions, with a minimum number of auxiliary units.

Design Overview

The sCO₂-HeRo TAC (Fig. 1) has an integrated design with a single-stage compressor, a single-stage turbine, and a generator in one casing. This represents the most compact machine and theoretically allows a hermetic system without any external leakage.

Table 1 Thermodynamic parameters for the SCO2-HERO TAC [1]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>T_C0</td>
<td>33 °C</td>
</tr>
<tr>
<td>P_C0</td>
<td>78.3 bar</td>
</tr>
<tr>
<td>( \pi )</td>
<td>1.5</td>
</tr>
<tr>
<td>T_F0</td>
<td>200 °C</td>
</tr>
<tr>
<td>( m )</td>
<td>0.65 kg/s</td>
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</tbody>
</table>

1Corresponding author.

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The contour lines in the Cordier diagram in Fig. 2 indicate the expected efficiency for turbocompressors with conventional blading design at given design parameters, while the dashed line indicates the curve of optimum efficiency according to Cordier. The sketched impellers in Fig. 2 illustrate the preferable impeller type. The dots represent the compressor parameters of several small-scale TAC systems either tested in the past years or still under investigation. All compressors are centrifugal compressors:

- Sandia National Lab (SNL) [3]
- Bechtel Marine Propulsion Corporation (BMPC) [4–7]
- Korea Institute of Energy Research (KIER) [8–10]
- Tokyo Institute of Technology (TIT) [11]

The compressors from SNL, BMPC, and TIT apply an unshrouded compressor design, while the impellers of KIER and KAERI have a shrouded design in order to overcome thrust issues, which showed up during testing of the SNL compressor (Cho et al. [8]). The compressors developed in these projects tend to be on the right of the region of best efficiency (Fig. 2). This is caused by a low flow coefficient due to the high density of sCO2 combined with the small power output (small mass flow rate) of the demonstration cycles. The TACs in Fig. 2 operate at similar compressor inlet pressures and temperatures, except for the one at BMPC, which operates further away from the critical point. The major difference of the sCO2-HeRo compressor (star) is the significantly lower mass flow in the sCO2-HeRo cycle resulting in an even smaller flow coefficient and a smaller machine. Thus, challenges related to the small size of the components become more severe, limiting the achievable efficiency.

Aerodynamic Design

The aerodynamic design must account for the properties of sCO2 close to the critical point. The high density of sCO2 means that windage losses become significant and the overall dimensions of this machine are rather small.

The optimum rotational speed is a compromise between stage efficiency and windage losses. The windage losses plus compressor power must not exceed the expected turbine power output. Because windage losses are approximately proportional to the parameters in Eq. (1), the rotational speed has to be limited to a value of around 50,000 rpm. In contrast, the optimum rotational speed for maximum stage efficiency for this small machine is according to Fig. 2 around 200,000 rpm.

Table 2 Compressor design parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tr>
<td>( n )</td>
<td>50,000 rpm</td>
</tr>
<tr>
<td>( \varphi_{\text{opt}} )</td>
<td>0.009</td>
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<tr>
<td>( \Psi_{\text{opt}} )</td>
<td>1.8</td>
</tr>
<tr>
<td>( \sigma_{\text{opt}} )</td>
<td>0.06</td>
</tr>
<tr>
<td>( \delta_{\text{opt}} )</td>
<td>12.4</td>
</tr>
<tr>
<td>( D )</td>
<td>40 mm</td>
</tr>
</tbody>
</table>

- Korea Atomic Energy Research Institute (KAERI) (Information regarding the diameter could not be retrieved. Therefore it is not included in Fig. 2) [12]
- University of Duisburg-Essen, HeRo-project (HeRo) [1,13] and Table 2.

The compressors from SNL, BMPC, and TIT apply an unshrouded compressor design, while the impellers of KIER and KAERI have a shrouded design in order to overcome thrust issues, which showed up during testing of the SNL compressor (Cho et al. [8]). The compressors developed in these projects tend to be on the right of the region of best efficiency (Fig. 2). This is caused by a low flow coefficient due to the high density of sCO2 combined with the small power output (small mass flow rate) of the demonstration cycles. The TACs in Fig. 2 operate at similar compressor inlet pressures and temperatures, except for the one at BMPC, which operates further away from the critical point. The major difference of the sCO2-HeRo compressor (star) is the significantly lower mass flow in the sCO2-HeRo cycle resulting in an even smaller flow coefficient and a smaller machine. Thus, challenges related to the small size of the components become more severe, limiting the achievable efficiency.
Besides limiting the rotational speed, windage losses can be reduced by lowering the pressure and thus the density in the central housing. SNL, B MPC, K IER, and KAERI use this approach and reduce the pressure below the compressor inlet conditions. The resulting leakage flow is returned to the cycle with a CO$_2$ pump. This is not a valid option for the sCO$_2$-HeRo cycle, because it limits the self-sustainability. The design therefore aims for a hermetic design similar to the TIT-design or a direct leakage feedback. The latter allows reducing the pressure in the cavity to compressor inlet conditions.

Since the generator cavity contributes for the greater part to the windage losses, it is used to determine the permissible rotational speed of the shaft. As stated by Wright et al. [3], standard windage models are applicable also for supercritical fluids. Thus, the windage losses in the generator cavity are calculated according to the model of Bilgen and Boulouss [14] for the generator dimensions depicted in Fig. 3. The figure shows the rapid increase of the windage losses $P_w$ with rotational speed $n$ for three different pressure levels of CO$_2$ within the cavity. For the sCO$_2$-HeRo, the pressure range between 78.3 bar and 117.5 bar is desired as it allows feedback of the leakage without an additional pump. The design speed is set to 50,000 rpm leading to windage losses of 1.6 kW at the generator. For the resulting overall dimensions of the rotor including the impellers and shaft, the windage losses are expected to be twice as high leaving a surplus power of around 4 kW. If the speed is increased further, e.g., to 60,000 rpm, the surplus power is reduced to 1 kW, which is considered to be an insufficient margin. By reducing the rotational speed, the point of compressor operation is shifted to lower specific speed and higher specific diameter causing lower efficiency (further to the right in Fig. 2). To reach efficiencies similar to SNL and B MPC, the sCO$_2$-HeRo compressor would, compared to the actual design, need to rotate at a 3.5 times higher speed having a 2.5 times smaller outer diameter. Investigations showed that the windage losses at this high speed exceed the surplus power of the turbine.

Compressor Design

This paragraph contains a short description of the compressor development steps and gives detailed information on the final design including geometry and calculated performance. The boundary conditions are summarized in Table 1. The rotational speed is 50,000 rpm. With help of the Cordier diagram in Fig. 2, the general impeller type is determined to be radial only. Therefore, a two-dimensional radial blading is applied, which is also easy to manufacture. Because the volume flow is small, an unshrouded design requires very small tip clearances to reduce tip leakage losses. Very small clearances raise the risk of rubbing between impeller and casing because of axial displacement of the rotor. Therefore, a cover disk is applied in combination with labyrinth sealing (similar to K IER and KAERI) allowing a significant reduction of clearance losses and simultaneously reducing the risk of contact between impeller and casing. Using a cover disk also reduces thrust forces, because the pressure difference between hub and shroud is reduced. This is especially useful during start-up, when the pressure differences between turbine and compressor can be large. The compressor consists of one stage with a vaneless parallel wall diffuser. The blade angle at the compressor trailing edge is set to $\beta_2 = 90$ deg instead of backward curved blades to increase the circumferential velocity component and be able to reach the required pressure ratio.

Simulations are carried out with ANSYS CFX to study the blading performance. The impeller is modeled together with the vaneless parallel wall diffuser and the volute. This simulation setup neglects leakage effects and friction in the impeller cavities in favor of reduced computational time. For selected working points, the front and back cavities and labyrinth seals on hub and shroud are considered. This allows a comprehensive simulation of the compressor incorporating leakage losses and friction. The computational domain, including the labyrinth seals and cavity, is shown in Appendix A. All simulations have a swirl-free inflow towards the impeller. According to Behafarid and Podowski [15] and Gong et al. [16], assuming sCO$_2$ as incompressible offers a good approximation for the compressor design. Schuster et al. [13] showed that this is also sufficient for the design procedure of the sCO$_2$-HeRo compressor. Therefore, preliminary calculations are carried out by applying constant properties of CO$_2$ for averaged conditions between inflow and outflow. The results are then applied as initial values for simulations considering compressibility and real gas effects using tabulated sCO$_2$ data. More information on the adopted real gas properties are given by Schuster et al. [13]. Due to rapid property changes of sCO$_2$ near the critical point, convergence is hard to achieve. Pressure clipping improves convergence without a considerable influence on the overall performance, since it occurs only in a small region at the blade leading edge. So far, no detailed investigations on rotating stall are carried out.

A suitable compressor blading is designed in an iterative way, starting with a conventional blading design with constant blading thickness. This study shows that at the leading edge, the change in $\beta$ needs to be small to reduce flow separation on the suction side. Toward the trailing edge, it needs to decrease strongly to achieve $\beta_2 = 90$ deg. Therefore, flow separation on the suction side toward the trailing edge cannot be avoided. All in all, the study of blades with constant thickness shows that they are not suitable for the design requirements of the sCO$_2$-HeRo compressor, because separation of the flow on the suction side of the blades cannot be avoided. It might even lead to unstable flow conditions. The next step finds a suitable blade thickness distribution to minimize flow separation between two adjacent blades. The final shape of the blading is depicted in Fig. 4 with Table 3 containing the design data. The flow is strongly guided in the first half of the blade channel before the flow area significantly increases. An elliptic contour is introduced at the trailing edge. For definition of the parameters in Table 3, please see the velocity triangles and meridional sketch in Figs. 4 and 5. The exact blading profile can be taken from the coordinates of pressure and suction side in Fig. 12 and Table 4 in Appendix B. Figure 12 depicts the blade marked as 1 in Fig. 4. Table 4 also contains the blade height ($z$-coordinate).

Performance Analysis

Figure 6 shows a contour plot of the velocity within the rotor and diffuser at mid span. The blading is a compromise between good guidance of the flow in between adjacent blades and the reduction of the unavoidable flow separation caused by the thick blades. The flow separations are close to the trailing edge, as indicated. Losses are mainly induced by the flow separations and the small inflow angle into the diffuser $\xi_2$ causing high losses in the...
The resulting design pressure ratio and isentropic efficiency are $\pi = 1.44$ and $\eta = 73\%$, respectively.

Figure 7 gives a glimpse of the compressor performance map. For the design inlet conditions, the pressure ratio $\pi$ is plotted versus the volume flow rate at the inlet $Q$. As a result of the small flow coefficient $\varphi_{in}$, the pressure ratio does not vary much with the volume flow rate $Q$. Further numerical and experimental studies will also focus on changing inlet conditions to present a broad spectrum of operation points for the design evaluation.

Due to time restrictions, the previously described blading geometry is built and will be tested, although it does not fully meet the requirement of $\pi = 1.5$. Further investigations were carried out in order to reach a higher pressure ratio and efficiency. For once the inflow angle into the parallel plate diffuser $\alpha_2 < 20^\circ$ suggests the use of diffuser blading. Different types were investigated, showing that the pressure ratio and isentropic efficiency in the design point could be further increased by using profiled diffuser blading. Investigations are also carried out for the use of splitter blades for the compressor with the goal of reducing the flow separations shown in Fig. 6 and increasing $\alpha_2$. The design improvements can be retrofitted to the machine due to its modular design.

### Mechanical Design

Various challenges occur in the design of a very compact turbo-machine for sCO2, which range from manufacturing strategies to selecting components. Material compatibility, especially for lubricants and polymers, is also of great importance for sCO2.

### Manufacturing of the Impellers

In the design of the sCO2-HeRo, the manufacturing of the impellers is a special challenge because of the small dimensions.
Even small tolerances in the manufactured geometry can cause significant deviations compared to the calculated performance. This holds especially for the joining of the milled blades on the hub with the cover plate, which causes changes in channel height. Thus, two joining techniques are tested.

**Diffusion Welding.** In this process, hub and shroud are pressed against each other at high temperature. To achieve good results, plastic deformation of the parts is required. Tolerances in the used equipment caused unacceptable differences in the channel height of up to 0.1 mm due to the plastic deformation in the welding process. More accurate equipment may reduce the error.

**High Temperature Vacuum Brazing.** In this process, brazing material is applied to the top of the blade or the shroud plate and both parts are joined at high temperature. After joining, no difference in the channel height could be detected. However, due to capillary forces, the applied brazing material tends to flow into the flow channel. This was observed in one of the two manufactured impellers of the compressor while no blocking of the flow path could be found for the other. This is an acceptable result for a first attempt.

**Bearings**

Several bearing types were investigated, from roller bearings over foil bearings to magnetic bearings. Investigation of the rotor dynamic behavior showed the eigenfrequencies of the rotor. Due to the integrated design with two overhung impellers, it has eigenfrequencies distinguished by the oscillation of each impeller. These eigenfrequencies lie close together and exclude a wide range of speeds if the machine would operate above the first critical speed. As the machine should allow a wide range of speeds, it runs below the first critical speed. Therefore, limitations due to eigenfrequencies are avoided but high stiffness of the rotor and bearings is required. For the rotor, this especially means to have the impellers as close as possible to the bearings and have bearings of high stiffness. Furthermore, the project requires to have a self-launching and self-sustainable system. Among the investigated bearings, magnetic bearings were dismissed because of the necessary auxiliary systems. Foil bearings cannot be used because of their limited thrust capacity and their start-up behavior requiring a minimum lift-off speed. This strongly limits the testing of the self-launching capability. Additionally, these bearing types could not provide the intended bearing stiffness for the required sizes. Finally, grease-lubricated (for lifetime) and heat-stabilized hybrid angular ball bearings with an extra-large grease reservoir, supplied by Cerobear, were chosen. It might be that sCO2 dissolves the grease and therefore reduces the service life of the bearings. This will be tested during the measurement campaign.

**Motor/Generator.** According to the findings of Wright et al. [3], it is not expected that there will be any significant effect of the sCO2 on the insulation of the generator. A four pole asynchronous motor manufactured by E+A is used. The used frequency inverter, supplied by Control Techniques, allows switching from motor to generator operation on the fly.

**Sealing.** The integrated machine design allows building a hermetic turbomachine that has no outgoing shaft. Therefore, O-rings are used for sealing of the casing. Clementoni and Cox [5] say that they found no issues with the sCO2 interaction for O-rings made of ethylene propylene diene monomer rubber (EPDM). Therefore, ethylene propylene diene monomer rubber is used for temperature regimes below 100°C. For higher temperatures, O-rings made from perfluoroelastomer (FFKM) are used.

Efficient sealing of the shaft toward the generator needs to be considered in regard of the efficiency of the compressor and turbine. Because the leakage from the compressor is utilized for generator cooling, a minimum leakage flow is required. The shaft sealing is realized by stepped labyrinth seals with 3 fins on the shroud side and 4 fins on the side of the hub (see Appendix A). The stepped geometry allows easy assembly while reducing the leakage in comparison to a see-through labyrinth seal. Optimized seals with changed chamber geometry resulting in 6 and 7 fins, respectively, are developed as well, but are not yet implemented in the machine. Further optimization of the sealing efficiency on the side of the hub is possible by applying other sealing techniques e.g., dry gas seals. The sealing on the side of the shroud is difficult to optimize, because only labyrinth seals seem applicable. An accumulated leakage mass flow of 0.1 kg/s–0.15 kg/s is expected at the outlet in Fig. 8 depending on the pressure within the central housing.

**Leakage Design Toward the Central Housing**

The leakage control is very important for two reasons. First reason is that even small leakage losses lead to substantial reduction in performance. Second, the leakage is utilized for cooling purposes. For the leakage, basically three design options exist, which are distinguished by the pressure in the central housing and therefore strongly interact with the windage losses (see Fig. 3) and the cooling. First option is to keep the pressure in the central housing subcritical, reducing the friction losses while increasing the cooling of the generator. The compressor and turbine losses are increased and an additional pump for leakage feedback is required. This possibility was applied for the cycles of SNL, BMPC, KIER, and KAERI.

![Temperature distribution in the central housing](image.png)
The second option is to have a pressure in the central housing, which is equal or above the low pressure side of the loop (compressor inlet pressure) allowing direct feedback without any additional pump. The third option is to have an internal leakage within the turbomachine, as applied in the turbomachine of TIT. The leakage flows from the compressor toward the turbine caused by the pressure losses in the heater, the pipe between compressor and turbine and the pressure reduction in the turbine stator. Therefore, this option does not even require the additional pipe and pressure regulation valve from the central housing toward the compressor inlet. The last two options have sCO2 in the central housing, which might reduce the bearing service life. Also, the windage losses increase, while the leakage is reduced. To investigate the effect of sCO2 in the central housing, the testing starts with the option 1 and the leakage flow will be reduced step-by-step.

Cooling of the Generator

The compactness of the turbomachine sets some requirements to the cooling of the machine, in particular of the generator. The required cooling of the generator is realized by a leakage flow through the generator. This avoids external cooling devices and supports the self-sustainable operation. The heat transfer analysis shows that the cooling by the leakage is sufficient when combined with cooling fins in the casing. In fact, the heat-up of the CO2 due to the generator losses is less than 20 °C as visible in Fig. 8, which shows the results of a simplified conjugated heat transfer calculation. The leakage flow efficiently cools the generator and also takes out sufficient heat to avoid a large heat transfer from the turbine to the compressor side.

Final Design of the Supercritical CO2 Heat Removal System Turbine, Alternator and Compressor

Figure 9 shows a picture of the final turbomachine during cold-run air tests. The overall width of the casing is 340 mm and the length of the shaft including impellers is 270 mm. The casing has a maximum diameter of 250 mm, which is required to allow screwing the different parts together. Thermocouples and a vibration sensor monitor the behavior of the machine and especially the bearings. The picture shows an additional cooling jacket that can be used to provide additional cooling and also lead the extracted heat to an outside cooler as not to heat up the room. Figure 10 shows the compressor impeller with a maximum diameter of 40 mm. The right part of the figure presents the stepped geometry of the labyrinth seal.

First tests of the machine included an overspeed test of the impellers at 120% of design speed and a pressure test of the casing followed by cold-run tests with air at pressures up to 40 bar. The air tests verified that the turbomachine can reliably reach its design speed and that all control signals are correct. Vibrations were monitored to check the vibration level. All tests were successfully conducted. Performance testing with sCO2 is done at Centrum výzkumu Rež (CVR) in Rež, Czech Republic.

Summary and Future Work

This paper describes the development of the turbomachinery for the sCO2-HeRo project—a promising approach to bring nuclear power plant safety to an even higher level. The challenges in designing small turbomachinery for carbon dioxide are explained and are also applicable for other carbon dioxide cycles.

The proposed design allows a self-sustained operation without the need of supplementary aggregates. The announced challenges regarding lubricant and O-ring material will be verified by performance test. A suitable compressor blading design is presented and the geometry is fully described. This gives the possibility to validated calculation tools once the performance tests are completed, and hence allows further detailed analysis and optimization of this geometry and future geometries with validated design tools.

Funding Data


Nomenclature

- $c_m$ = meridional speed (m/s)
- $D$ = diameter (m)
- $h$ = channel height (m)
- $k$ = blocking factor
- $L$ = length (m)
- $m$ = mass flow (kg/s)
- $n$ = rotational speed (1/s)
- $P$ = pressure (Pa)
- $Q$ = power (kW)
- $Q_0$ = volume flow (m²/s)
- $q$ = blade angle (deg)
- $s$ = blade thickness (m)
- $t$ = gap width (m)
- $T$ = temperature (°C)
- $u$ = circumferential speed (m/s)
- $v$ = specific flow work (J/kg/°C)
- $\delta_{opt}$ = blade angle (deg)
- $\eta$ = efficiency (%)
- $\pi$ = pressure ratio
- $\rho$ = density (kg/m³)

"Transactions of the ASME"
\[
\sigma_{\text{opt}} = \left( \sqrt{\frac{\varphi_{\text{opt}}}{\psi_{\text{opt}}}} \right)^{3/4} = n \times \left( \sqrt{Q_0/(2 \times y)} \right)^{3/4} \times 2 \times \sqrt{\pi},
\]

specific speed 
\[
\varphi_{\text{opt}} = \left( \frac{c_m}{n} \right) = \left( 4 \times \frac{Q_0}{y} \times D^3 \times n \right), \text{ flow coefficient}
\]

\[
\psi_{\text{opt}} = \left( 2 \times y/n^2 \right) = \left( 2 \times y/\pi^2 \times D^2 \times n^2 \right), \text{ head coefficient}
\]

**Subscript**

- \( C \) = compressor
- \( T \) = turbine
- \( 0 \) = inlet
- \( 1 \) = impeller inlet
- \( 2 \) = impeller outlet
- \( 3 \) = diffuser outlet

**Abbreviations**

- sCO2 = supercritical CO2
- TAC = turbine, alternator, compressor system

**Appendix A: Cavity and Labyrinth Geometry**

Figure 11 contains a sketch of the compressor with dimensions for the impeller cavities and labyrinth seals in the detailed views A and B, respectively. The inflow is a nozzle reducing the diameter from 15.8 mm at the compressor inlet to 12 mm at the impeller. The cavity has an outer diameter of 41 mm and width of up to 4 mm. At a diameter of 84 mm, the flow exits the diffuser into the volute with a constant width of 7.5 mm. The height of the volute increases linearly from 0 mm to 8.5 mm with one revolution. Then the flow enters into a circular pipe with diffuser, which increases the diameter from 7.5 mm to 15.8 mm. Detail B shows a chamber of the stepped labyrinth seal. As all chambers are equal, dimensions are only given for one. Each chamber is 3.5 mm high and 5 mm wide. The clearance is 0.1 mm with a fin being 0.25 mm wide at the tip.

**Appendix B: Compressor Blade Profile**

Figure 12 and Table 4 contain the coordinates to the blading curves on pressure and suction side on the side of the cover disk. As a two-dimensional blading is applied, replacing the z-coordinate with 0 mm gives the curve on the side of the shroud. The axis of rotation is the z-axis.
### Table 4 Blade curves in coordinates (continued)

<table>
<thead>
<tr>
<th>Curve on pressure side</th>
<th>Curve on suction side</th>
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Trailing edge

Leading edge

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### References


