Design and Validation of a Test Rig for Heat Transfer Measurements on a Rotating Cone

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Abstract
The paper presents the setup of a test rig to measure heat transfer coefficients on an actively cooled rear cone of a high pressure compressor. First results of an experimental heat transfer study on two different cooling assemblies are discussed.

To derive local heat transfer coefficients on a rotating cone under varying engine related conditions, a new test rig was built at the Institut für Thermische Strömungsmaschinen in Karlsruhe/Germany. The test rig mainly consists of a rotating cone and an adjacent conical casing. The cooling concept consists of holes uniformly distributed around the circumference of the conical casing. Heat transfer measurements are performed in the gap between cone and casing whereby the gap width can be varied. The geometry of the cooling air supply holes can be modified by inserting different rings in the conical casing which include various cooling air geometries.

The test rig operating conditions can be varied over a wide range of Reynolds numbers and air mass flow rates to cover the load ranges of several jet engines. Experimental results for the heat transfer of two different cooling air inlets are presented. An analysis of the measurement uncertainty is conducted. Errors are less than 10\%.

Therefore, the results facilitate the design and optimization of the secondary air system of a high pressure compressor of a jet engine in terms of compressor rear cone cooling.

Nomenclature

\[ n \quad \text{Normal vector} \]
\[ Nu \quad \text{Nusselt number} \]
\[ \omega \quad \text{rad/s} \quad \text{Rotational speed} \]
\[ p \quad \text{Pa} \quad \text{Pressure} \]
\[ \Pi \quad \text{Pressure ratio} \]
\[ Pr \quad \text{Prandtl number} \]
\[ q \quad \text{W/m}^2 \quad \text{Heat flux} \]
\[ r \quad \text{mm} \quad \text{Radius} \]
\[ Re \quad \text{Reynolds number} \]
\[ \rho \quad \text{kg/m}^3 \quad \text{Density} \]
\[ s \quad \text{mm} \quad \text{Gap width} \]
\[ SR \quad \text{Swirl ratio} \]
\[ T \quad \text{K} \quad \text{Temperature} \]
\[ T^* \quad \text{Dimensionless temperature} \]
\[ \theta \quad \text{Angle} \]
\[ u \quad \text{m/s} \quad \text{Velocity} \]

Subscripts

\[ \text{amb} \quad \text{ambient} \]
\[ f \quad \text{fluid} \]
\[ i \quad \text{inner} \]
\[ is \quad \text{isentropic} \]
\[ jet \quad \text{jet} \]
\[ o \quad \text{outer} \]
\[ \phi \quad \text{circumferential} \]
\[ r \quad \text{rotor} \]
\[ ref \quad \text{reference} \]
\[ s \quad \text{surface} \]
\[ t \quad \text{total} \]
\[ w \quad \text{wall} \]

Introduction
Increasing the efficiency and decreasing pollutant emissions are the main goals in developing future jet engines. A promising way to achieve these goals is a higher engine pressure ratio. This entails rising fluid temperature at the compressor outlet. To guarantee structural and thermal integrity of the compressor rotor, cooling of the compressor becomes necessary. Active cooling of the compressor rear cone, which is usually located between the last stage of the high-pressure compressor and the high-pressure shaft (see Figure 1), is a possible concept to decrease the occurring thermal and structural loads and to increase lifetime.
Cooling air can be taken from the inner diffusor upstream of the combustion chamber, where the static pressure is higher compared to the rear cone cavity. The temperature difference required for cooling the rear cone is provided by a radial temperature gradient in the diffusor. Another purpose of the cooling air is to act as sealing air to avoid hot gas ingestion from the main flow path into the conical rotor-stator gap. The supply of cooling air to the turbine is one of the primary functions of the secondary air system of a jet engine. Therefore, previous experimental and numerical work focused on rotating turbine discs, shafts and cavities with or without pre-swirled air and axial or radial through flow. The current research found in literature is the reduction of the total relative temperature of the cooling air, the optimization of the relative speed between fluid and rotor surface with respect to reduced frictional heating and increased heat transfer. However, only very limited literature is available for active cooling of a rotating cone with pre-swirled air. In order to fill this gap, a test rig is set up at the Institut für Thermische Strömungsmaschinen (ITS) in Karlsruhe, Germany. Main objectives are the determination of heat transfer coefficients along the conical rotor surface in order optimize active rotor cooling. Furthermore, discharge coefficients of different cooling assemblies are measured, enabling the optimization of the secondary air system of the entire engine. The work presented deals with the setup of a test rig for experimental investigation of an actively cooled compressor rear cone. The experiments focus on the heat transfer in the rotor-stator system. Heat transfer coefficients are derived from temperature measurements at the rotor and stator surface using finite element calculations. Moreover, discharge coefficients of the cooling holes, which transfer the cooling air from the diffusor plenum upstream of the combustion chamber into the rotor-stator gap, are determined. In the following, the design of the test facility, the design constraints and the test rig setup will be presented in detail. After a brief description of the measurement technique, a detailed error estimation is presented, followed by the discussion of the results of two different cooling assemblies.

**Experimental Setup**

**Problem analysis**

The geometry of the compressor rear cone consisting of the rotor and the stator with cooling air inserts can be simplified as depicted in Figure 2. In order to be able to describe the problem mathematically, the input parameters were first analyzed using the Buckingham-Π theorem. The resulting dimensionless numbers are given in Table 1. Previous work [1, 2, 3, 4] confirms the application of those dimensionless numbers in order to sufficiently describe the entire system and provide scalability to real engine conditions.

The operating conditions of the test rig are defined by the circumferential Reynolds number $Re_\phi$, and the flow parameter $C_w$. Several different geometries can be obtained by varying the gap width $s$ and the pre-swirl angle $\alpha$ (see Figure 5). The required design parameters were matched to the dimensionless numbers in order to sufficiently describe the entire system and provide scalability to real engine conditions. The operating conditions of the test rig are defined by the circumferential Reynolds number $Re_\phi$, and the flow parameter $C_w$. Several different geometries can be obtained by varying the gap width $s$ and the pre-swirl angle $\alpha$ (see Figure 5). The required design parameters were matched to the dimensionless numbers in order to sufficiently describe the entire system and provide scalability to real engine conditions.

An additional constraint is the reversal of the direction of heat transfer compared to the engine in order to avoid high temperatures close to the rotor bearings.
As a consequence, the rotor inner surface is cooled, whereas the outer surface is heated by the main gas. With respect to the heat transfer phenomena of interest, this does not entail any major difference in heat transfer, as a fully developed turbulent flow regime is expected. To ensure consistency with respect to the real engine, the inlet holes will in the following be referred to as cooling holes.

**Test facility**

Measuring a steady-state temperature gradient across a surface requires two independent fluid cycles. Figure 3 shows a scheme of the infrastructure supplying the test rig with the main hot gas (A) and the coolant (B). For both the hot gas and the coolant path, air is provided by up to three compressors with a capability of $0.7 \times 10^3$ kg/s air mass flow at a maximum pressure of $1 \times 10^6$ Pa. The actual pressure directly upstream of the test rig is controlled by a bypass valve (C) and is held at a constant value of approximately $6.5 \times 10^5$ Pa. The pressures and mass flow rates in the hot gas and the coolant path can be controlled independently by one valve each upstream and downstream of the respective gas path (1). The air mass flow rates entering the test rig are measured by two orifice flowmeters (2). The temperature of the main gas is controlled by an electric heater with an overall power of 64 kW to enable maximum temperature differences between hot gas and coolant of $\Delta T = 150$ K. The heater is connected to a mixer and a manifold (3), from where

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**Figure 3: Scheme of the test facility.**

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**Table 1: Dimensionless properties for a full problem description**

<table>
<thead>
<tr>
<th>Definition</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$r_s/r_o$</td>
<td>Curvature parameter CP</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Cone angle</td>
</tr>
<tr>
<td>$s/r_o$</td>
<td>Aspect ratio AR</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>Pre-swirl angle</td>
</tr>
<tr>
<td>$\omega r_s / (\omega \cdot r_o)$</td>
<td>Swirl ratio SR</td>
</tr>
<tr>
<td>$c_p T / u_{in}^2$</td>
<td>Total enthalpy of inlet air</td>
</tr>
<tr>
<td>$c_p / \mu / k$</td>
<td>Prandtl number Pr</td>
</tr>
<tr>
<td>$m / (\mu c_p)$</td>
<td>Air mass flow rate $C_W$</td>
</tr>
<tr>
<td>$\rho \cdot \omega \cdot r_s^2 / \mu$</td>
<td>Circumfer. Reynolds number $Re_\phi$</td>
</tr>
<tr>
<td>$h \cdot r_s / k$</td>
<td>Nusselt number $Nu$</td>
</tr>
<tr>
<td>$h \cdot r_s / k_i$</td>
<td>Biot number $Bi$</td>
</tr>
</tbody>
</table>
the hot gas enters the outer settling chamber of the test rig ④. Subsequently, the hot air enters the cooling holes ⑤, is accelerated and reaches the rotor-stator gap ⑥. The air mass flow rate $C_W$ through the cooling holes is controlled by varying the pressure ratio $\Pi$ between settling chamber and inner cavity between $1 < \Pi \leq 2$. While a minor percentage of the hot gas passes the labyrinth seal between outer and inner cavity ⑦, the major part flows downstream along the rotor surface and causes heating of the outer rotor surface. The rotor is driven by an electric motor with a power of 25 kW to reach rotational speeds up to 10 000 rpm. The hot gas exits the test rig through a flow conditioner ⑧ to reduce pressure losses due to dissipation of the swirling flow. The mass flow rate of hot gas at the outlet is measured by an orifice flowmeter ⑨ before leaving the test facility via the above mentioned downstream control valve. The coolant mass flow passes a manifold ⑩ from where it is distributed to the inner settling chamber ⑪. From there, the coolant flows along the inner rotor-stator cavity ⑫ and cools the inner rotor surface to provide a temperature gradient across the rotor structure. Upstream of the rotor-stator cavity the cooling air mixes with the labyrinth seal leakage and exits the test rig via several hoses leading to the collector ⑬. Before the coolant leaves the test facility, the mass flow rate is measured ⑭ including the leakage mass flow from the labyrinth seal. The balance of inlet and outlet mass flow rates enables the estimation of the leakage flow across the labyrinth seal.

**Test section**

The test section is shown in detail in Figure 4. It consists of three basic parts, the rotor (dark grey), the axially traversable stator (light grey) and the fixed stator (grey) and the axially traversable stator (light grey). The red area marks the main gas path including the conical gap, while the blue area represents the coolant path. The geometry of the conical gap was derived from the dimensionless numbers from Table 1. The curvature parameter $CP$, aspect ratio $AR$ and cone angle $\theta$ define the geometry of the rotor with an inner diameter $r_i = 0.1$ m, an outer diameter $r_o = 0.22$ m and a cone angle $\theta = 35^\circ$. The gap width $s$ between rotor and adjacent moveable stator can be varied from 6 to 25 mm in order to match the specified aspect ratios.

A detailed view of the rotor-stator gap and the cooling air inlet geometry is illustrated in Figure 5. The cooling assembly consists of two interchangeable rings with 48 cooling holes equally distributed around the circumference, each with a diameter $d = 3$ mm. The inlet geometries only differ in the inclination angle $\alpha$ of the cooling holes. While the baseline geometry has an inclination angle of $\alpha_1 = 90^\circ$, the second inlet geometry has an inclination angle of $\alpha_2 = 30^\circ$. Hence, with this geometry it is possible to reduce the relative total temperature of cooling air based on the rotor surface speed.

The casing, consisting of the fixed and moving stator, was designed to withstand the maximum pressure of $1 \times 10^6$ Pa at the maximum temperature of 450 K to ensure a wide range of pressure ratios across the cooling holes. Furthermore, the casing serves as containment in case of rotor failure. To allow optical flow measurements, the casing contains two windows at the inner and outer radius of the measurement cavity.

**Rotor design**

The design of the rotor needs to satisfy different requirements concerning measurement technique, struc-
tural integrity and rotor dynamics. As mentioned above, the outer shape of the rotor was defined by the curvature parameter $C_P$ and the aspect ratio $AR$. To match the Biot number and to reduce the measurement error, a Ti6Al4V alloy, similar to those in jet engine compressor structures, was chosen as rotor material.

A rotor thickness $H$ of 15 mm was found as an optimum to obtain a minimal mass and sufficient stiffness (dynamic behavior), optimal stress distribution (structural integrity) and to maximize the temperature gradient between inner and outer rotor surface (measurement technique). The rotor is balanced by means of a dovetail groove at the outer radius and balancing holes at the inner radius. The rotor is connected to the shaft by a cylindrical clamping sleeve.

Operating conditions

Based on the afore-mentioned design of the test rig and the test facility, a wide range of operating conditions, as displayed in Table 2, is possible.

As no flow transition and a highly turbulent flow is expected for all investigated Reynolds numbers (see [5, 6]) it is assumed that results can be extrapolated to higher Reynolds numbers.

Measurement Technique

Data acquisition

Heat transfer coefficients at a specific surface are calculated by [7]

$$h = \frac{k (dT)}{dh} \left|_w \right. = \frac{\dot{q}_w}{\Delta T_{ref}}$$

where $T_{ref}$ is an arbitrarily chosen reference temperature and $T_w$ is the wall temperature at the surface. In case of one-dimensional heat conduction through the structure, the heat flux through the surface $\dot{q}_w$ can be calculated by

$$\dot{q}_w = \dot{q}_s = k_s \cdot H_s \cdot \Delta T_s$$

Table 2: Operating range of the test rig.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>$AR$</td>
<td>8 … 35</td>
</tr>
<tr>
<td>$SR$</td>
<td>0 … 2</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>0 … 150 K</td>
</tr>
<tr>
<td>$C_P$</td>
<td>0 … 50 x $10^3$</td>
</tr>
<tr>
<td>$Re_\phi$</td>
<td>0 … 10.1 x $10^6$</td>
</tr>
</tbody>
</table>

Paper Figure 5: Detailed sketch of the cooling air inlet geometry.
previous section it was shown, that the heat transfer coefficient for one-dimensional heat transfer can be written as

\[ h = f \left( k_s, H_s, \Delta T_s, \Delta T_{ref} \right) . \]  

(3)

Based on equation (3), an uncertainty analysis has been conducted to determine the uncertainty of the calculated heat transfer coefficients. The analysis was performed in Python using the uncertainties module [8]. The accuracy of relative temperature measurements was set to \(|\Delta T| = 0.5\) K, which was confirmed by measurements in a calibration fluid. The thermal conductivity \(k_s\) of the Ti6Al4V was measured to have a value of approximately 7 W/m/K with a relative uncertainty of \( \pm 0.35\) W/m/K. The thickness of the rotor structure can be determined with an absolute accuracy of 0.1 mm. The temperature differences \(\Delta T_{ref}\) and \(\Delta T_s\) depend on the respective heat transfer coefficients (HTC). Thus, relative accuracy of the temperature measurements changes with the heat transfer coefficients at the inner and outer surface of the rotor structure. Figure 6 shows the measurement error for the previously mentioned accuracies of \(|\Delta T| = 0.5\) K (above) and an additional plot for an improved accuracy of \(|\Delta T| = 0.1\) K (below) to illustrate the potential of an improved accuracy for relative temperature measurements. The range of heat transfer coefficients was chosen based on previously conducted CFD studies [9] at the outer rotor surface and on analytical investigations at the inner surface. The reference temperature \(T_{ref}\) is the total temperature of the inlet air of the respective fluid channel (hot gas at the outer surface and coolant at the inner surface). In the following, heat transfer coefficients are normalized to a reference value \(h_{ref}\).

At the inner surface, dimensionless heat transfer coefficients between 1 and 2.5 are expected. Heat transfer coefficients at the outer surface are in the range of 10 directly beneath the cooling holes and in the range of 1 to 2.5 downstream of the cooling holes. The upper plot in Figure 6 shows isolines of the resulting measurement error of the heat transfer coefficients according to equation (3) for two different temperature measurement accuracies. For the estimated accuracies of \(|\Delta T| = 0.5\) K, geometry and material data, heat transfer coefficients with an accuracy of 8% and lower can be reached in a wide range. Only directly beneath the cooling holes, much higher thermocouple accuracy is needed to reach overall accuracies of approximately 10%. Figure 6 also shows that the measurement accuracy can be increased considerably by increasing the heat transfer at the inner rotor surface.

The second target value to be determined from experimental investigation is the discharge coefficient \(c_D\) of the cooling holes as a function of the inclination angle \(\alpha\). The discharge coefficient is defined as

\[ c_D = \frac{m}{m_i} = f \left( m_i, p_{\text{in}}, p_2, T_1, ..., d \right) \]  

(4)

where the isentropic mass flow rate \(m_i\) is a function of the inlet total pressure \(p_{\text{in}}\), the inlet temperature \(T_1\), the outlet pressure \(p_2\) and the cooling hole diameter \(d\). The mass flow rate \(m\) can directly be measured with an orifice flowmeter (see Figure 3 and Figure 4) with a relative uncertainty of \(|\Delta m/m| = 1.5\%\) based on calculations according to the ISO standard 5167. Pressure is measured with an uncertainty of \(|\Delta p/p| = 0.05\%\), temperature with an absolute uncertainty of \(|\Delta T| = 1.5\) K. The cooling hole diameter can be determined with an absolute accuracy of 0.05 mm. With these values, a relative uncertainty of \(|\Delta c_D/c_D| = 2.5\%\) can be achieved.

Results and Discussion

Data Analysis

As heat transfer coefficients cannot be measured directly, experimentally obtained data has to be processed in an appropriate way in order to derive reliable results. A scheme of the data analysis is displayed in Figure 7. The first step is to measure temperatures at the inner and outer surface of the rotor. The temperature measurement locations are depicted in Figure 4. Obviously, there are areas on the surface where no temperature information is available, namely at the
outer diameter next to the dovetail groove and the inner diameter next to the shaft. In the finite element model, an adiabatic wall is assumed at the respective surfaces. A previously conducted sensitivity study showed that the temperature distribution on the conical surface, which is of unique interest, is not affected by the choice of these boundary conditions.

In a second step, the locally measured temperatures are interpolated linearly to derive a global temperature distribution on the conical rotor surface. The interpolated temperatures are set as boundary conditions in the finite element model. As a result of the finite element calculation, the temperature distribution and hence the heat flux within the rotor structure is derived.

In the last step, the surface normal heat flux is extracted from the calculated results. By applying equation (1) and referring the surface heat flux to a characteristic temperature difference, heat transfer coefficients are calculated.

**Experimental heat transfer results**

Two different inlet geometries were investigated as depicted in Figure 5. The operating conditions for both cases are summarized in Table 3. For a single operating point approximately 20 minutes are necessary until steady-state conditions are reached. Afterwards, about 100 values are recorded and averaged. The measured temperature distribution along the inner (triangles) and outer rotor surface (squares) is shown in Figure 8, were the temperature \( T \) is normalized to the total temperature of the inlet air \( T_1 \) and the ambient temperature \( T_{amb} \) according to

\[
T^* = \frac{T - T_{amb}}{T_1 - T_{amb}}.
\]

The solid lines represent the interpolated temperatures at the inner surface, whereas the dashed lines stand for the interpolated temperatures at the outer surface. Applying these temperature distributions as boundary conditions for the finite element calculation, the heat transfer coefficients in Figure 9 are obtained. As the heat transfer coefficients at the inner rotor surface are not of particular interest, only heat transfer coefficients at the outer surface are illustrated.

The heat transfer coefficients depicted in Figure 9 are based on the total inlet temperature of the fluid as reference temperature. The radial coordinate \( r \) ranges from the inner radius of the rotor \( r \approx 40 \) mm to the outer radius of the rotor \( r \approx 220 \) mm. Heat transfer coefficients are displayed for the conical part of the rotor from 100 mm to 220 mm (see also Figure 2).

In case 1 with an inclination angle of 90° the heat transfer coefficients are globally higher compared to case 2 with an inclination angle of 30°. Heat transfer coefficients directly opposite the cooling air inlet \( r \approx 200 \) mm are generally higher for both cases with a local maximum at \( r \approx 210 \) mm. The influence of the cooling air inlet has disappeared at \( r \approx 180 \) mm. Further downstream, the Nusselt number nearly remains constant for both cases 1 and 2. As expected from the

![Figure 7: Scheme of the calculation of heat transfer coefficients.](image)

![Figure 8: Temperature distributions for cases 1 and 2 at inner and outer rotor surface.](image)

**Table 3: Test conditions**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \alpha )</td>
<td>90°</td>
<td>30°</td>
</tr>
<tr>
<td>( AR )</td>
<td>10.46</td>
<td>10.46</td>
</tr>
<tr>
<td>( SR )</td>
<td>0</td>
<td>1.32</td>
</tr>
<tr>
<td>( \Delta T )</td>
<td>108 K</td>
<td>106 K</td>
</tr>
<tr>
<td>( C_w )</td>
<td>( 34.5 \times 10^7 )</td>
<td>( 32.72 \times 10^7 )</td>
</tr>
<tr>
<td>( Re_\phi )</td>
<td>( 3.6 \times 10^6 )</td>
<td>( 4.5 \times 10^6 )</td>
</tr>
</tbody>
</table>
CFD study [9], the heat transfer coefficients downstream of the cooling air inlet are in the range of 1 to 1.5 for case 1. For case 2 a heat transfer coefficient in the range of 0.7 is found in this region. Heat transfer coefficients at the inner rotor surface are in the range of 1.5 to 4 (not depicted in Figure 9). Figure 6 shows an excellent accuracy for these values. In case of a thermocouple uncertainty of 0.5 K a relative error for the heat transfer coefficients of less than 10% can be expected.

By improving the thermocouple accuracy to 0.1 K, the accuracy of heat transfer coefficients could be increased considerably to values in the region of 5%.

Conclusion

A newly designed test rig for experimental investigations of heat transfer coefficients in a conical rotor-stator system with active cooling has been introduced. The main focus of the experiments is to determine heat transfer coefficients on the surface of a rotor inside of a conical gap for a wide range of operating conditions similar to those in jet engines. Error estimation showed that the test rig has the capability to determine the target values, namely heat transfer coefficients and discharge values of the cooling air holes with the specified accuracy of 10%. First measurements for two different cooling hole geometries confirmed the results from the error estimation. It has been shown, that a further increase of the temperature measurement accuracy would lead to a higher accuracy of the derived heat transfer coefficients. Furthermore, the results for both cases showed a region with locally high heat transfer coefficients directly opposite of the cooling air inlet. Heat transfer coefficients further downstream are considerably lower. In a next step, more experiments will be run in order to receive comprehensive data depending on several geometries and operating conditions.

The data obtained from the rig will be used to understand the physical phenomena regarding flow field and heat transfer on a rotating compressor rear cone with active cooling. The ultimate objective is to use an improved understanding for the optimization of the cooling concept based on the desired operational parameters.

Acknowledgement

The authors would like to gratefully acknowledge the European Commission for supporting the present work within the Seventh Framework Programme project LEMCOTEC funded under the grant agreement 283216. Furthermore, the authors would like to thank MTU Aero Engines for the collaboration within the LEMCOTEC project.

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