HIGH SPEED SINGLE CAVITY RIG WITH AXIAL THROUGHFLOW OF COOLING AIR: HEAT TRANSFER AND FLUID PHENOMENA

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ABSTRACT

The understanding of heat transfer and fluid motion in compressor rotor drums is a key know-how for the design and analysis of highly loaded and efficient aeroengines. Since rotor blade tip clearance depends on the thermal growth of the disks, the accurate prediction of the temperature and Nusselt number distribution of these structures is crucial for the turbomachinery performance. Historically the calculation of heat transfer coefficients was done with respect to the inlet temperature of the air entering the cavity. This could lead to misinterpretation of local effects. Therefore, accurate measurements of air and metal temperatures can shed light on the phenomena that impact the disk heat transfer.

This paper presents the results of a single cavity rig that consists of a pair of identical disks, a cylindrical shroud and a stationary inner shaft. The disks and the shroud of the rig can be heated individually to generate an axial gradient to the cavity as found in real engines. Disk and fluid temperature as well as static pressure measurements of the cavity are carried out by a two-sided telemetry system. A new calculation method for the core-swirl ratio distribution from static pressure measurements inside the cavity is presented and compared to laser Doppler anemometry (LDA) investigations from previous work. The operating conditions cover a wide range of cases that are representative of typical high pressure compressor operation. There are additionally new test conditions with asymmetric heating on the outer surface of the cavity.

KEYWORDS
cavity flow, buoyancy, heat transfer, air temperature, swirl ratio

NOMENCLATURE

\[ a \quad \text{bore radius of rotor [m]} \]
\[ b \quad \text{outer radius of the cavity [m]} \]
\[ c \quad \text{regression coefficients [Pa/m}^{n}\text{]} \]
\[ d_h \quad \text{radial clearance of annular bore section [m]} \]
\[ Gr = (1 - a/b)^3 Re_{\phi}^2 \beta \Delta T \quad \text{Grashof number [-]} \]
\[ k = V_{\phi}/\Omega r \quad \text{swirl ratio [-]} \]
\[ p \quad \text{static pressure [Pa]} \]
\[ r \quad \text{radius [m]} \]
$R$

$Re_\phi = \frac{\rho_{bore} \Omega^2}{\mu_{bore}}$

rotational Reynolds number [-]

$Re_z = \frac{\rho_{bore} \bar{V}_z^2 (a - r_s)}{\mu_{bore}}$

axial Reynolds number [-]

$Ro = \frac{\bar{V}_z}{(\Omega a)}$

Rossby number [-]

$s$

axial width of the cavity [m]

$T$

static Temperature [K]

$u, v, w$

radial, tangential, and axial component of velocity in rot. frame of reference [m/s]

$\bar{V}_z$

axial bulk velocity at bore section [m/s]

$V_r, V_\phi, V_z$

radial, tangential, and axial component of velocity in stat. frame of reference [m/s]

$\beta \Delta T = \frac{(T_{sh} - T_{bore})}{T_{bore}}$

buoyancy parameter [-]

$\mu$

dynamic viscosity [m$^2$/s]

$\rho$

density [kg/m$^3$]

$\sigma$

uncertainty in 95% confidence interval [-]

$\Omega$

angular speed of rotor [1/s]

**SUBSCRIPTS**

$bore$

bore flow

c

rotating core

$DS$

downstream

$sh$

shroud position

$US$

upstream

**INTRODUCTION**

Rotating cavities are part of compressor rotor drums in jet engines and stationary gas turbines. Air, which is bled off the high pressure compressor stages flows in an annular gap between the disks bore and the rotating inner shaft to cool the highly stressed turbine parts (see figure 1). On this flow path, the air is partially entering these cavities where fluid motion is non-axisymmetric, unsteady and unstable in some operating conditions (Owen 2015). Research on cavity flow and heat transfer was intensively done over the past years. A prime goal is to understand how the operating conditions like rotational frequency, thermal load from the main gas path, temperature and amount of the bled air and also the geometry affects the flow structure and the thermal gradients in the compressor disks. To improve the design of those compressor stages, a small radial clearance between the compressor blades and the outer casing is wanted. For a safe design, it is necessary to calculate the thermal and centrifugal caused strains in all operating conditions. Furthermore, the temperature pickup of the air passing the cavity is of importance to determine the effective cooling temperature of the fluid. The flow in the...
cavity can be categorized in 3 main regions (illustrated in figure 2): a rotational dominated fluid core in the outer domain above the cobs of the disks, a throughflow driven torodial vortex between the cobs of the disks and the flow in the Ekman-layers radial inward on the surface of the disks, where the incoming mass flow is compensated (Farthing et. al. 1992). Examinations with optical measurement techniques observed a rising tendency of the fluid core to reach solid body rotation with decreasing Rossby number and for smaller gaps between the disks (Farthing et.al. 1992, Long et. al. 2007). When the cavity is heated, buoyancy induced flow entrains the cavity by an cold radial arm and cyclonic, anticyclonic vortices are formed in the outer domain. The buoyancy driven flow enhances the heat transfer in the cavity through a more intense radial mixing with cold air from the axial throughflow (Atkins et. al. 2014). Determination of Nusselt numbers was done by measuring heatfluxes with sensors (e.g. Farthing et. al 1992), solving Fourier’s-law with measured temperatures as boundarys (e.g. Günther et. al. 2012) or by using a circular fin equation to model the disk temperature distribution (Tang et. al 2015, Jackson et. al. 2020). In all available examinations the experimental derived heat transfer coefficient was calculated with respect to the temperature in the axial throughflow, which just leads to biased heat fluxes. The latest introduced theoretical model by Tang and Owen (2017) takes the interaction of buoyancy driven flow with the recirculating air from the Ekman layers into account. The driving temperature difference must be known, to improve the calculation of Nusselt numbers. For this, it is necessary to study the fluid temperature distribution inside the cavity. Little research was done to this subject. Tucker and Long (1998) did measurements of the fluid core temperature at three different radial locations in mid axial plane. The uncontroled disks were heated, but the temperature profile was partially differering to compressor rotor disks in real engines.

Figure 2: Theory of possible flow regimes in a rotating cavity with axial throughflow and heated shroud. Adapted from Farthing et. al. (1992) and Jackson et. al. (2020)

This work presents disk as well as fluid core temperatures at engine representative conditions, at a lower level of temperature. A phenomenological analysis of the temperature profiles is presented and explained by the latest theories for buoyancy induced flow. By comparing the dimensionless temperature profiles for different Rossby numbers and heating conditions, the sensitivity of the rig to changes in operating conditions is shown. Furthermore tests with asymmetric heating condition are presented to show the influence on the air temperatures inside the cavity. The radial distribution of the tangential velocity in the core of the cavity is calculated
from pressure measurement, by applying a simplified theory of the fluid motion and is compared to LDA-measurements from Long (2007).

**EXPERIMENTAL APPARATUS AND DATA TREATMENT**

The experimental facility is shown in figure 3. The rig consists of one cavity placed in a symmetrical annular gap section, which allows to change the flow direction and presents similar inlet conditions before the cavity inlet. Analogue to the cooling flow path in real rotor drums, swirl is added to the axial throughflow by rotating orifice disks. The cavity is formed by two identically contoured disks and a cylindrical shroud. To lower heat conduction towards the outer rotor structure, an insulation material is placed between the carrier and the cavity disks. Carrier-disk, shroud and cavity-disks are assembled with a bolted flange joint at \( r > b \). The cavity components of the rotor are manufactured from Ti-6Al-4V titanium alloy Grade 5. The annular bore section is bounded by a central, non-rotating, inner shaft made of steel. The cavity dimensions can be expressed by nondimensional geometry parameters like the gap aspect ratio \( G = 0.26 \), an inner radius ratio \( a/b = 0.27 \) and the bore gap ratio \( d_h/b = 0.14 \). The width between the cobs is half the distance between the disks. To generate thermal gradients on the cavity, the disks can be heated individually by radiation heaters with a total power of 21 kW. In addition to this, the shroud can be heated by hot air which is impinged onto the rotating surface. All massflow rates are determined by standard ISO 5167-1 orifice plated with a relative accuracy < 1.5 %. A detailed explanation of the rig and the facility can be found in Diemel et. al. (2019).

**Instrumentation**

An array of thermocouples is used to investigate the distribution of the cavity metal- and fluid temperatures. A layout of the instrumentation is shown in figure 3. Material temperature
measurement is performed with point welded thermocouples, which are embedded in the surface of the disks. The wires are laid in tangential distance in a groove, which is backfilled with epoxy resin, to avoid conduction errors and flow disturbance. Fluid temperature measurements are done with in-house made Typ-K thermocouples held by a 1.5 mm diameter strut. The length of the strut is 10 mm for all probes in the cavity and varying for those in the annular bore section. The thermoelectric relation, known as the Seebeck-function, for both types of thermocouples was calibrated with two oil filled temperature calibrators. The resulting overall absolute (including uncertainty of the cold junction temperature measurement) and difference temperature accuracy is ±0.34±0.23 K and ±0.23±0.13 K for material- and flow-thermocouples respectively. The temperature of the cold junctions is measured by calibrated resistor thermometers. Disk A is equipped with a high amount of material thermocouples to measure a detailed distribution of the metal temperatures. Metal temperatures measured on Disk B are used to compare and control the thermal fields of both disks. The shroud temperature is measured at 3 axial positions, which are in the center of the cavity and s/4 distance to Disk-A and B. Static pressure is measured by telemetric pressure sensors, located next to Telemetry-A. They are connected to pressure taps at the surface of Disk-A by small tubes. All telemetric sensors are connected to an in-house built digital dual telemetry system. The inlet and outlet condition of the cavity is characterized by measuring static pressure, fluid temperature and the flow-vector at medium channel height. The tangential flow angle and the velocity components are measured by a 3-hole pressure probe used in non-nulling mode, neglecting radial flow components. A detailed description of the calibration and evaluation method is given in Diemel et.al. (2019) and Díaz et.al. (2009).

Calculation of Core-Swirl Ratio

The flow field in a rotating cavity with axial throughflow can be described by the linear equations for inviscid rotating fluids (Owen and Long (2015)). This simplified theory assumes that the nonlinear inertial accelerations are small compared to the Coriolis acceleration. By assuming \( u, v, w \ll \Omega r \), equation (1) is derived from the steady Navier-Stokes equation in cylindrical coordinates. This simplification shows that the pressure in the core of the cavity is independent from the axial coordinate.

\[
\frac{\partial p_c}{\partial r} = 2\rho_c \Omega \left( v_c + \frac{1}{2} \Omega r \right)
\]  

(1)

The taken assumptions could be suitable for the rotating core outside the Ekman-Layer in the cavity. By introducing the core-swirl ratio \( k_c = \frac{v_c}{\Omega r} = \frac{v_c}{\Omega r} + 1 \) and assuming ideal-gas behavior, equation (1) forms to:

\[
k_c = \frac{RT_c}{2\rho_c \Omega^2 r} \frac{\partial p_c}{\partial r} + \frac{1}{2}
\]  

(2)

The calculation of the core swirl ratio from equation (2) is performed with time-averaged sensor readings. The static pressure can be approximated as constant for a radius, so readings from the wall taps on disk A are used. The fluid core temperature is measured by the instrumentation shown in figure 3 and is interpolated by a root function. For radial pressure and its gradient, a best-fit quadratic polynomial regression is used. The uncertainty quantification was done by Gaussian error propagation. For this, it is useful to formulate equation (2) using the fit-functions
with \( p_c(r) = c_3 r^3 + c_2 r + c_1 \) and \( p'_c(r) = 2c_3 r + c_2 \),

\[
k_c = \frac{RT_c (2c_3 r + c_2)}{2(c_3 r^2 + c_2 r + c_1) \Omega^2 r} + \frac{1}{2}
\]

(3)

where \( c_1 \) to \( c_3 \) are the regression coefficients. Hence the uncertainty can be calculated by:

\[
\sigma_{k_c} = \sqrt{\left( \frac{\partial k_c}{\partial c_3} \sigma_{c_3} \right)^2 + \left( \frac{\partial k_c}{\partial c_2} \sigma_{c_2} \right)^2 + \left( \frac{\partial k_c}{\partial c_1} \sigma_{c_1} \right)^2 + \left( \frac{\partial k_c}{\partial p_c} \sigma_{p_c} \right)^2 + \left( \frac{\partial k_c}{\partial \Omega} \sigma_{\Omega} \right)^2 + \left( \frac{\partial k_c}{\partial T_c} \sigma_{T_c} \right)^2}
\]

(4)

The uncertainty analysis shows that the biggest contributors are the goodness of the fit and the uncertainty of the pressure measurement. The pressure readings have to be corrected by the influence of the radial acceleration on the sensor membrane, temperature drift and the air column pressure rise in the connecting tube from the sensor to the wall tap. A detailed description of the corrections can be found in Uffrecht et. al. (2008).

**Temperature Distribution**

The presented metal and fluid temperatures are averaged values in steady state condition of minimum 5 minutes at a sampling frequency of \( \approx 6 \) Hz. Steady state condition was achieved when all metal temperatures of the cavity parts had a minimum to maximum bandwidth lower than the relative temperature accuracy of the used thermocouples in a period of 10 minutes. Typically this was achieved about 1-1.5 hours after heating up the cavity. All metal and fluid temperatures are plotted in non-dimensional form, defined as

\[
\Theta = \frac{T(r) - T_{bore}}{T_{sh} - T_{bore}}
\]

(5)

where \( T_{bore} \) is the axial throughflow temperature at mid radius of the annular section and \( T_{sh} \) is the average of the 3 axial spaced shroud temperatures. Due to redundancy of these temperatures the values are mean of 2 thermocouples at the same location, tangential mirrored. To quantify the axial-symmetry of the metal temperatures thermocouple readings from the same radius with an angle difference of 9° and 180° are compared. In all studied cases the circumferential disturbance in steady state condition was < 0.5 K. Since the velocity of the flow in the bore section and in the cavity are < Ma 0.2, there is no need to take recovery effects on the air temperature probes into account. Due to high temperature difference between the walls and the fluid in the cavity, the readings of the air thermocouples are potentially influenced by strut conduction, especially at higher radius (r/b > 0.5) where the fluid core reaches solid body rotation and the temperature difference between wall and fluid is the highest. To apply a correction of the probe, the heat transfer distribution along the strut should be known. Experimental and numeric investigations of the probe conduction error for these probes, showed an error up to 15% of the temperature difference between wall and fluid at low relative velocities (3-5 m/s).

**EXPERIMENTAL RESULTS**

The testing conditions are expressed by non-dimensional parameters. Tests were performed at two levels of axial Reynolds number (low, high) in a range of \( 2 \times 10^4 < Re_z < 5 \times 10^4 \) with a variation of Rossby numbers in a range of \( 0.2 < Ro < 1 \). The heating conditions were adjusted to reach certain levels of buoyancy parameters up to 0.3, which lead to Grashof numbers > \( 1 \times 10^{12} \). Asymmetric heating conditions were applied to the cavity, where the upstream disk
(Disk B) was cooler than the downstream disk (Disk A). The heating condition is expressed with an buoyancy parameter for the shroud, upstream (US) disk and downstream (DS) disk. The buoyancy parameter for asymmetric heating condition (see legend fig. 5) is noted in the form of $\beta_0 \Delta T_{sh} | \beta_0 \Delta T_{DS} | \beta_0 \Delta T_{US}$, expressed by eq. (6). For symmetric heating condition only $\beta_0 \Delta T_{sh}$ is used.

$$\Delta T_{sh} = T_{sh} - T_{bore} \quad \Delta T_{DS} = T_{DS,(r=b)} - T_{bore} \quad \Delta T_{US} = T_{US,(r=b)} - T_{bore}$$ (6)

### Disk and Core Temperatures

At first, the tests with symmetrical heated disks are discussed. In figure 4-a, the dimensionless metal temperatures for the downstream disk are presented for the high level of axial Reynolds number. All presented curves are showing changes in curvature, where the cob radius of the disk begins ($x = 0.54$). The distribution of dimensionless temperature for the low heating condition are showing no significant differences by a decrease of Ro. At higher radius ($x > 0.6$) the temperatures are slightly lower for the case with low Rossby number compared to those at high Rossby number. When comparing the temperature curves at high heating condition the effect of Rossby number is more significant. The temperature profile at radii $x > 0.54$ is more curved for low Ro compared to the test with high Ro. This implies a weaker radial mixing at the same heating condition. In general it can be seen that the temperature profiles are more curved for tests with higher heating condition, which is related to buoyancy induced mixing. The same effects are also described by Atkins (2014). When looking at the fluid temperatures measured in a distance of 10 mm from the disks and shroud (fig. 4-b) it can be seen that these are lower for higher heating condition compared to lower heating condition, which supports the theory of buoyancy induced mixing. For all test cases the readings of the probe located next to the shroud

![Figure 4: Comparison of downstream disk temperatures (a) and fluid temperatures (b) for a symmetrical heated cavity at constant $Re_{z, high}$ for two levels of Rossby number and $\beta_0 \Delta T_{sh}$.](image)
are lower than those of the probe which is mounted at the upstream disk at almost same radius. Since the shroud is hotter than the disk at the radius where the probe is mounted, it indicates that the probe at the shroud is cooled more intense. This can be explained by the optical investigations from Farthing et. al. (1992). When the cavity is heated from the shroud they observed, that cold fluid from the axial throughflow is entering the cavity in a radial arm that flows to the shroud (see illustration of flow regimes in figure 2). The air temperatures measured at the lowest radius seem to be more influenced by the Rossby number than by heating condition. Figure 5 shows disk and fluid temperatures for a symmetrical and asymmetrical heated cavity for two levels of Rossby number. In both cases the axial Reynolds number was $Re_{z,low} \approx 0.5 \cdot Re_{z,high}$ and the flow configuration was unchanged to figure 4. The dimensionless tem-

Figure 5: Comparison of down- and upstream disk temperatures (a,c) and fluid temperatures (b,d) for a symmetrical (red) and asymmetrical (blue) heated cavity at constant $Re_{z,low}$ for two levels of Rossby number. Buoyancy parameter is notated as $\beta \Delta T_{sh} \mid \beta \Delta T_{DS} \mid \beta \Delta T_{US}$ and refers to eq (6).
perature profiles for both Rossby numbers shows a higher temperature at the inner radius of the disk compared to the tests with higher axial Reynolds number (fig. 4-a). This is plausible because the higher throughflow is resulting in a stronger heat sink at the cylindrical bore face. The test cases with Ro = 1 (fig. 5-c) shows an almost linear distribution for \( x > 0.54 \) with higher temperatures at \( x > 0.7 \) compared to the tests at Ro = 0.9 in figure 4-a. The upstream disk of the symmetrical heated test case in figure 5-c was 1 K hotter at highest radius and 4 K hotter in the middle of the cob. By comparing the probe readings in 5-d, it can be noticed that also the fluid temperatures of the upstream side were hotter, than those measured from the downstream disk. This could be explained by a toroidal vortex between the cobs which is heating the upstream disk with the recirculating air from the Ekman layers. The same mechanism is described by Jackson et. al (2020) and Günther (2012). When the upstream disk was less heated, the measured air temperatures between the cobs (\( x = 0.29 \) and \( x = 0.36 \)) are showing lower temperature differences between up- and downstream side. Furthermore the fluid temperature on the downstream side at \( x = 0.36 \) is still cooler than the reading of the probe located on the upstream side. This supports the theory that a toroidal vortex is fed by fresh air from the axial throughflow on the downstream side and the hotter air from the Ekman layers is mixing in, where the radius of the cobs ends. The fluid temperatures measured at the downstream disk at higher radii seem to be unaffected by the lower temperature of the upstream disk. The reading of the probe at highest radius on the upstream disk is showing 10% cooler temperatures than in the symmetrical heated case. The disk and fluid temperature distribution for the symmetrical heated test case at Ro = 0.19 is shown in figure 5-a,b. No significant temperature differences between up- and downstream disk and fluid temperatures occurred in this test case. When the upstream disk was cooler than the downstream disk the readings of the air temperature probes on the upstream side showed slightly lower temperatures, compared to the symmetric heated case, but since the probes are potentially influenced by strut conduction these differences cannot be clearly evaluated. As Atkins (2014) showed in Computational-Fluid-Dynamics simulations and Farthing et. al. (1992) examined for low Rossby numbers, the toroidal vortex degenerates or disappears completely and a thin shear layer is forming between the axial throughflow and the cavity. This would explain the different behavior of the air temperatures between the test cases in figure 5-a,b to c,d. The same trend in air temperature at low radii with Rossby number can be seen in the symmetrical heated tests of figure 4.

**Static Pressure and Velocity Distribution**

The tangential velocity in the annular bore before the cavity, measured by 3-hole pressure probes, and the distribution inside the cavity, calculated by equation (2) from static pressure and temperature measurement, is presented in figure 7. Static pressure measurements from the downstream disks surface is presented in figure 6 and shows higher overall pressure gradient for lower Rossby numbers, which is caused by higher rotational speed of the rotor. The velocities are normalized by the local rotor speed. For the values in the annular bore, the rotor speed at the inner radius of the disk is used. The results are compared to LDA measurements from Long et. al. (2007). This rig had an inner radius ratio of a/b = 0.318, a gap ratio of s/b = 0.195, with an wide and narrow gap ratio of \( d_{w}/b = 0.164 \) \| \( d_{n}/b = 0.092 \) respectively. The testing conditions of the presented cases are analogous to those in figure 4 and the axial Reynolds number is about half of that in the investigations of Long. The calculated core-swirl ratio curves show an increase with decreasing Rossby number. All test points show an increase of swirl ratio with respect to radius and reaching unity at \( x^* \approx 0.55 - 0.65 \).
The tests of Long in wide gap configuration (fig. 7) show a decrease in swirl ratio from the bore face further into the cavity (negative gradient) and reaching solid body rotation at $x^* \approx 0.75$. The peak value of tangential velocity at the inner radius of the disk is caused by the acceleration of the axial throughflow by a toroidal vortex located between the cobs of the disks (Frathing et. al. (1992)). This peak is lowered with decreasing Rossby number. In all examined test cases (also theses not shown here) this peak in tangential velocity does not occur for the calculated velocities and the gradient of the curves is always positive. Since this flow phenomena is related to the toroidal vortex between the cobs of the disks it is obvious that the simplified theory of the linear inviscid equations can not capture this complex fluid motion because the static pressure should not independent from the axial direction anymore. The swirl distribution of the investigations done with narrow annular gap configuration (fig. 7) by Long (2007) are showing an increase in swirl ratio (positive gradient), starting with values lower than rotor speed between the cobs and reaching solid body rotation.

Figure 6: Pressure distribution for two levels of $Ro$ and $\beta \Delta T_{sh}$ at constant $Re_{z, high}$

Figure 7: Comparison of swirl ratio distribution at constant $Re_{z, high}$, for two levels of Rossby number and $\beta \Delta T_{sh}$ with anemometry data $\triangle$ from Long (2007) in WG = wide gap, NG = narrow gap configuration. Solid lines represent calculated swirl ratio from pressure readings and $\bigcirc$ from 3-hole probe measurements. Uncertainty bars indicate measurement location.
further into the cavity, comparable to the presented test cases. Similar velocity distributions were found by Owen et. al. (2006). This major difference in swirl distribution found by previous research workers shows, that the forming of the vortex flow between the cobs is strongly influenced by the gap ratio of the annular bore. Therefor, the presented swirl distributions may not seem unrealistic.

When the disks are heated stronger only small difference in swirl ratio occur in the outer part of the cavity $x^* > 0.5$. Furthermore it can noticed that the pressure in this part of the cavity is noticeable lower for the test with lower Rossby number when $\beta \Delta T_{sh}$ is increased (see fig. 6). This correlates with the fluid temperature change shown in figure 4-b. The calculated uncertainty of the swirl ratio is shown at the pressure measurement position and becomes larger with decreasing rotational frequency. This is explained because uncertainty of the telemetric pressure measurement, as well as the quality of the fit becomes dominant compared to the weak centrifugal caused pressure rise (see figure 6 and equation (4)).

CONCLUSIONS

Steady state measurement of disk and core temperatures as well as static pressure measurement were collected with an single cavity rig at high pressure compressor representative conditions. The tests were conducted at throughflow rates of $2 \times 10^4 < Re_{\phi} < 5 \times 10^4$, rotational Reynolds numbers $Re_{\phi} < 3.5 \times 10^6$ and heating settings of $\beta \Delta T < 0.3$. This leads to Grashof numbers up to $1.3 \times 10^{12}$ and Rossby numbers $0.2 < Ro < 1$. Additionally, tests with asymmetric heating conditions were studied, where the downstream disk was hotter than the upstream disk.

The analysis of the dimensionless temperature profiles for disk and fluid temperatures in the cavity showed similar behavior as previous experimental investigations. Two dominant mechanisms were found to explain the changes in temperature distributions by varying the test conditions. The first is the radial penetration of a vortex in the cob zone, dominated by the axial throughflow and the second is increased radial mixing by buoyancy induced flow proportional to $\beta \Delta T$ and Grashof number. The first mechanism is lowered as Rossby number is decreased. By analyzing the fluid temperatures in the cavity, compared to the metal temperatures, it can be assumed that heat is transferred from the downstream disk to the upstream disk by vortex driven flow between the cobs and recirculating air from the Ekman layers. This theory is also supported by the latest examinations of Jackson et. al. (2020). The fluid temperatures measured from the downstream disk of the cavity found to be lower than those on the upstream side for high Rossby numbers. For low Rossby numbers no significant deviations in fluid temperature between up- and downstream side occur. When applying an asymmetric heating condition to the disks of the cavity, the core fluid temperatures are lower on both sides for high Rossby numbers and stays almost constant for low Rossby number.

A new method for calculating the core-swirl ratio distribution from static pressure and fluid temperature readings inside the cavity was introduced. It uses the linear inviscid equations for rotating fluids derived from the Navier-Stokes equations. The comparison of the calculated swirl ratios to previous experimental investigations (Long (2007), Owen (2006)) shows reasonable agreement. A rising tendency of the fluid core to reach solid body rotation with decreasing Rossby numbers was observed. The method showed no significant dependency of the buoyancy forces to the calculated swirl ratios. The advantage of this method for calculating swirl ratios in the cavity from pressure measurement is, that it can be applied to test conditions with high centrifugal accelerations, where particle image velocimetry or laser Doppler anemometry fails.
Future work will focus on the analysis of the temperature and heatflux distribution of the disks to calculate Nusselt numbers and the validation of the calculation method for the swirl ratio in the cavity with CFD calculations.

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