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The effect of inlet conditions on the flow and heat transfer in multiple rotating cavity with axial throughflow

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Abstract: This paper discusses experimental results from two different build configurations of a heated multiple rotating cavity test rig. Measurements of heat transfer from the discs and tangential velocities are presented.

The test rig is a 70% full scale version of a high pressure compressor stack of an axial gas turbine engine. Of particular interest are the internal cylindrical cavities formed by adjacent discs and the interaction of these with a central axial throughflow of cooling air. Tests were carried out for a range of non-dimensional parameters representative of high pressure compressor internal air system flows (Re_ϕ up to 5×10^6 and Re_z up to 2×10^5). Two different builds have been tested. The most significant difference between these two build configurations is the size of the annular gap between the (non rotating) drive shaft and the bores of the discs.

The heat transfer data were obtained from thermocouple measurements of surface temperature and a conduction solution method. The velocity measurements were made using a two component, LDA system. The heat transfer results from the discs show differences between the two builds. This is attributed to the wider annular gap allowing more of the throughflow to penetrate into the cavity. There are also significant differences between the radial distributions of tangential velocity in the two builds of the test rig. For the narrow annular gap, there is an increase of non dimensional tangential velocity $V_\phi/\Omega r$ with radial location to solid body rotation $V_\phi/\Omega r = 1$. For the wider annular gap, the non dimensional velocities show a decrease with radial location to solid body rotation.

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Nomenclature

a, b	inner and outer disc radii	r_1, r_o	inner and outer radii for evaluation of average disc heat transfer
A	area	r_s	shaft radius
d_h	hydraulic diameter of the inlet	Re_z	axial Reynolds number(= Wd_h/ν)
G	gap ratio(= s/b)	Re_ϕ	rotational Reynolds number(= $\Omega b^2/\nu$)
Gr_y	Grashof number(= $\Omega^2 r \beta (T_s - T_{in}) y^3 / \nu^2$)	Ro	Rossby number(= $W/\Omega a$)
k	thermal conductivity	s	axial width of cavity
$\dot{m}$	mass flow rate of the throughflow	t	thickness of disc
Nu_{av}	average Nusselt number(= $q_{av} b / \Delta T_{av} k$)	T	temperature
Nu_y	Nusselt number based on y (= $qy / \Delta T k$)	V_z, V_ϕ	axial and tangential velocities
p	pressure	W	bulk mean axial velocity
q	heat flux	Y	distance radially in from shroud(= $b - r$)
r, z, ϕ	radial, axial and tangential coordinates	β	volume expansion coefficient(= $1/T$)

ΔT	temperature difference
μ	dynamic viscosity
ν	kinematic viscosity
ρ	density
Ω	rotational speed of the cavity

Subscripts and superscripts	
av	radially weighted average
in	value at inlet
ref	appropriate reference value
s	pertaining to the disc surface

Introduction

Contemporary gas turbine aircraft engines operate at the limits of technology. Gas temperatures of around 1 800 K at entry to the turbine on a typical civil engine limit the useful life of components due to creep and fatigue. High rotational speeds create significant rotational stresses. These centrifugal loads combine with thermal stresses and create deflections in components that can cause a loss in efficiency or, worse, compromise safety, if due account were not taken of these deflections.

Such engines would not be able to operate without an internal air system. This provides ventilation (mostly cooling, but sometimes components are heated), sealing and purging air to the discs, blades, bearing chambers and various cavities. A schematic diagram of a typical high pressure compressor internal air system is shown in Figure 1. Air, extracted from stages of the intermediate pressure compressor and destined for the intermediate pressure turbine blades and seals, flows through the annular passage between the drive shaft and compressor disc bores. The use of this air, which can be 20% of the main stream flow, represents a parasitic loss to the main cycle, since it is bled from the compressor where work has been done to raise its pressure. Its use may cost up to 5% of the specific fuel consumption in a modern turbofan engine (Rolls-Royce^[1] (2005)). On its route, the air may be heated by both convection and viscous dissipation.

The two factors that motivated this study

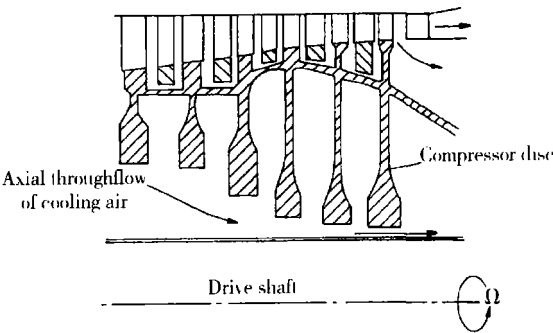


Fig. 1 A schematic diagram of a gas turbine high pressure compressor internal air system

are: firstly, heat transfer from the rotating surfaces inside the compressor creates a temperature rise in the air which needs to be predicted; and secondly, this heat transfer affects the temperature difference between disc bore and rim creating thermal stresses. There is therefore, a need in engine design to have knowledge of heat transfer rates and leakage flows in order to minimise the use of internal system air.

Previous work, Farthing et al.^[2-3] (1992) and Long^[4] (1994), on rotating cavities with axial throughflow has mostly concentrated on an idealised single rotating cavity. Two discs of outer radius, b , inner radius, a , are separated by an axial gap, s . The rotational speed of the cavity is Ω and the bulk average velocity of the axial throughflow is W . For a fluid of kinematic viscosity ν , the rotational and axial Reynolds numbers, are defined as

$$Re_{\phi} = \Omega b^2 / \nu \tag{1}$$

$$Re_z = W d_h / \nu \tag{2}$$

where d_h is the hydraulic diameter of the inlet. A further non-dimensional parameter, the Rossby number, Ro , is defined as the ratio of the mean velocity of the throughflow to the tangential velocity at the bore radius

$$Ro = W / \Omega a \tag{3}$$

This can also be expressed in terms of the rotational and axial Reynolds numbers as

$$Ro = \left\{ \frac{b^2 Re_z}{2a(a - r_s) Re_{\phi}} \right\} \tag{4}$$

Laser illuminated flow visualisation and laser Doppler anemometry (LDA) was used by Farthing et al.^[2] (1992) to study the flow struc-

ture in heated cavities with $a/b \approx 0.1$. Flow is seen to enter the cavity through a radial arm, bifurcate near the outer radius and form a region of cyclonic flow (with the same sense of rotation as the cavity) and anticyclonic flow (having the opposite sense of rotation). Heat transfer rates were measured by Farthing et al.^[3] (1992) from the discs of a heated rotating cavity with an axial throughflow of air, for $2 \times 10^5 \leq Re_\phi \leq 5 \times 10^6$ and $2 \times 10^4 \leq Re_z \leq 16 \times 10^4$, the cavity had a gap ratio of $G = 0.138$ and $a/b \approx 0.1$. The heat transfer from the discs for a cavity with a surface temperature on both discs that increased with radius was correlated by the expression:

$$Nu_y = 0.0054 Re_z^{0.3} Gr_y^{0.25} \quad (5)$$

Where $y = (b - r)$, is the distance radially inward from the shroud;

$$Nu_y = qy / (T_s - T_m)k \quad (6)$$

and

$$Gr_y = \Omega^2 r \beta (T_s - T_m) y^3 / \nu^2 \quad (7)$$

The significance of Equation (5) is that it suggests the flow and heat transfer in the narrow ($G = 0.138$) cavity occurs as a result of rotationally induced laminar free convection. This finding is confirmed in the work by Long and Tucker^[5] (1994), who made heat transfer measurements from the peripheral shroud of a cavity with a similar gap ratio. For a wider gap ratio ($G = 0.36$, $a/b \approx 0.1$) two mechanisms were identified by Long^[4] (1994) as being responsible for the heat transfer: rotationally induced buoyancy and direct influence of the central throughflow.

An earlier study carried out on an engine representative geometry is reported by Burkhardt et al.^[16] (1992). Their experimental rig comprised five cavities (similar to Figure 1 but having discs with a common hub radius) with $G = 0.256$ and $a/b = 0.286$, and tests were carried out for $1.9 \times 10^6 \leq Re_\phi \leq 5.6 \times 10^6$ and $2.7 \times 10^4 \leq Re_z \leq 9.5 \times 10^4$. The test rig also carried a central drive shaft which could be made to either co-rotate or contra-rotate. The heat transfer from the discs was found to increase when the rotational speed of the shaft approached that of the discs. For the central and outer part of the

discs there was reasonable agreement between measured local Nusselt numbers and those predicted by Equation (5). More recent experimental work on an engine representative test rig was carried out by Alexiou et al.^[7] (2000), on the disc and cone configuration of the current multiple cavity rig. The heat transfer measurements from the inner surface of the cone reveal two regimes of heat transfer. For $Ro / (\beta \Delta T_m)^{1/2} < 6$, the heat transfer is governed by rotation, and by action of the throughflow for larger values of this parameter. In the rotationally dominated regime, the heat transfer from the inner surface of the cone depended on the sense of rotation of the central drive shaft and is consistent with the results of Burkhardt et al. In the throughflow dominated regime the Nusselt numbers show little consistent dependence on the direction of shaft rotation.

Owen and Powell^[8] (2004) made velocity and heat transfer measurements in a single cavity research rig with an annular inlet, $a/b = 0.4$ and $s/b = 0.2$. Tests were carried out for $4 \times 10^5 < Re_\phi < 3.2 \times 10^6$ and $1.4 \times 10^3 < Re_z < 5 \times 10^4$. The radial velocity was found to be approximately two orders of magnitude smaller than the tangential velocity, although the radial velocity is of a similar magnitude to the relative tangential velocity. Further analysis of the velocity measurements revealed behaviour that is consistent with one, two or three pairs of cyclonic and anticyclonic vortices in the flow field. The heat transfer measurements add further support to the existence of two different regimes: a buoyancy induced regime at large rotational speeds and small axial flow, and a throughflow dominated regime at the smaller values of rotational speed and larger values of axial throughflow.

Various numerical studies have been carried out: Long and Tucker^[9] (1994), Long, Morse and Tucker^[10] (1997), Tucker and Long^[11] (1995) and Tucker and Long^[12] (1996), Wong (2002) Sun, Killfoil, Chew and Hills^[13] (2004) and Tian, Tao, Ding and Xu^[14] (2004). In general, these CFD studies have given results that are

qualitatively similar to experimental observations and broadly similar to heat transfer and LDA measurements. However, huge computational resources (months of CPU time) are required for this time dependent 3-D flow and the accuracy of much of the work prior to circa 2000 is constrained by relatively coarse grid sizes and questionable turbulence models. The recent 3D numerical study by Tian et al.^[14] (2004) supports the earlier qualitative flow visualisation work of Farthing et al. Owen et al.^[15] (2006) carried out a numerical simulation and made comparisons with experimental measurements of tangential velocity and heat transfer in a single cavity rig with only the downstream disc heated. The computed flow structure shows the features identified by Farthing et al. and with the exception of one test case there is good agreement between the numerical results and the experimental measurements. Bohn et al.^[16] (2006) make a comparison between experimental observations of flow structure and those computed using a 3D time dependent numerical code. The cyclone and anticyclone pairs observed in the experimental work are also found in the computations. It is also seen that the number of cyclone and anticyclone pairs can change with time. The authors suggest that the cyclone and anticyclone pairs are created by the buoyancy force and their number and strength is due to the Coriolis force. What ap-

pears to be the first reported application of a Large Eddy Simulation to this flow is presented by Sun et al.^[17] (2006) who also make comparisons with measurements of shroud heat transfer and tangential velocity obtained in the test rig used in this paper.

1 Experimental apparatus, procedure and data analysis

A full and detailed description of the rig (in disc and cone configuration), its instrumentation and ancillary equipment is available in the thesis by Alexiou^[18] (2000), so only a brief description is given here. The test rig for the work presented here is shown in Figure 2, the rotor has three internal discs forming four cylindrical cavities, the disc bores are of identical inner radius and the central drive shaft is stationary. This rotor assembly is bolted together at the periphery and is housed within a steel pressure casing. Each cylindrical cavity has an inner radius ratio of, $a/b=0.318$, and a gap ratio $s/b=0.194$ (dimensional details are presented in non dimensional form at the request of our industrial sponsors).

The rotor is independently driven by an A.C. motor up to 10 000 r/min. As in a high pressure compressor, a steel ‘drive’ shaft, runs through the centre of the rig. Two different designs of drive shaft were tested which differ in their outer radius. These will be referred to as

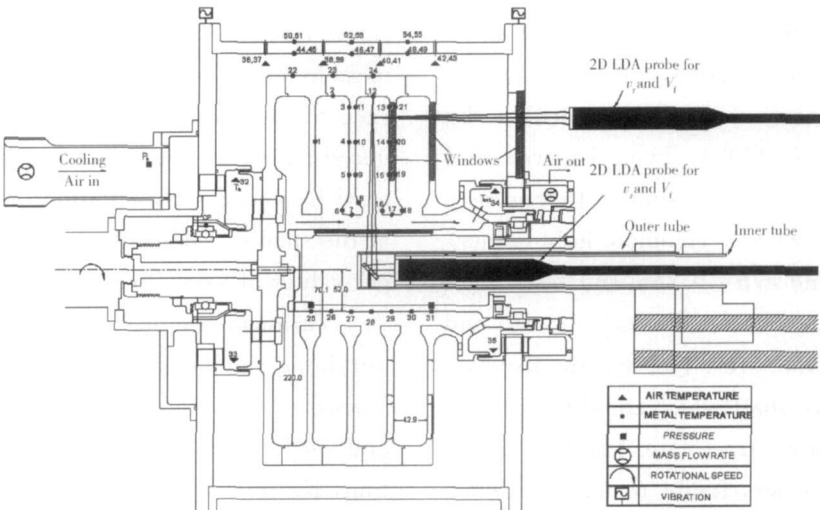


Fig. 2 Multiple cavity test rig

the narrow annular gap ($d_h/b=0.092$) and the wide annular gap ($d_h/b=0.164$). The drive shaft carries a flat window made from 3 mm thick BK 7, Grade A optical quality glass, 20 mm wide and 150 mm long, to provide optical access to the cavities for LDA measurements. Air is supplied to the rig by a single stage 275 kW Howden screw compressor which can deliver up to 1.1 kg/s of air at 4 bar (absolute) and 200 °C. The air flow rate is measured by a single orifice plate at inlet and by another at the exit. The outer surface of the rotor assembly is heated by impingement of hot air obtained from two 10 kW Leister 5000 8D5 heaters.

The surface temperatures of the rotor assembly and relevant air temperatures are measured using calibrated 0.25 mm wire diameter K-type (NiCr-NiAl) thermocouples. The locations of these are also indicated in Figure 2. Of particular interest are the thermocouples on Disc 2 (denoted by metal thermocouples Nos. 3 to 11, on Disc 3 (Nos. 13 to 21). The air inlet temperature is measured by air thermocouples 32 and 33 and the air exit temperature by Nos. 34 and 35. All the rotor instrumentation is led out to a 48-way, Wendon slip ring unit driven from the upstream side of the rig. The signals from all the temperature instrumentation are recorded by a

Solartron Orion data logger, with a sensitivity of $\pm 1\mu\text{V}$ corresponding to $\pm 0.025\text{ }^\circ\text{C}$ for the thermocouples used. However, the overall uncertainty in temperature measurement, which includes the effects of measurement and calibration uncertainty, is around an order of magnitude greater than this figure.

Velocity measurements are carried out using a DANTEC, LDA system. The discs are painted black to reduce reflections and all the windows have an anti-reflective coating. The transmitting lens has a focal length of 250 mm and a beam spacing of 15 mm giving a probe volume length of 4.046 mm, 14 fringes with a spacing of 8.579 μm . For seeding the cooling air flow, atomised oil particles ($\approx 1\text{ }\mu\text{m}$ diameter) are injected through the inner shaft downstream of the rotor inlet using a 6-jet TSI atomiser.

The data from the heat transfer and LDA tests discussed here were acquired under thermal steady state conditions and was confirmed by monitoring a change in surface temperatures of less than 0.2 °C in a period of 5 minutes. The variables investigated, their range, the corresponding range of the relevant non-dimensional groups, together with typical values from an engine, are given in Table 1.

Table 1 Range of experimental conditions

		Narrow Annular Gap Heat Transfer	Narrow Annular Gap LDA	Wide Annular Gap Heat Transfer	Wide Annular Gap LDA	Uncertainty/ Engine Value
Rotor speed	Ω / (rad/s)	$57 \leq \Omega \leq 615$	$123 \leq \Omega \leq 306$	$65 \leq \Omega \leq 608$	$65 \leq \Omega \leq 570$	± 1 rad/s
Cooling air flow rate	$\dot{m}$ / (kg/s)	$0.164 \leq \dot{m} \leq 0.8$	$0.184 \leq \dot{m} \leq 0.62$	$0.16 \leq \dot{m} \leq 0.75$	$0.2 \leq \dot{m} \leq 0.72$	$\pm 3\%$
Inlet pressure	p_{in} / (bar, abs.)	$2.2 \leq p_{in} \leq 3.16$	$2.25 \leq p_{in} \leq 3.0$	$2.3 \leq p_{in} \leq 3.7$	$2.3 \leq p_{in} \leq 3.1$	$\pm 0.1\%$
Inlet temperature	T_{in} / K	$309 \leq T_{in} \leq 324$	$311 \leq T_{in} \leq 319$	$302 \leq T_{in} \leq 339$	$302 \leq T_{in} \leq 339$	± 0.2 K
Disc average surface temperature	$T_{s,av}$ / K	$328 \leq T_{s,av} \leq 351$	$328 \leq T_{s,av} \leq 339$	$321 \leq T_{s,av} \leq 353$	$321 \leq T_{s,av} \leq 353$	± 0.4 K
Shroud average surface temperature	T_{sh} / K	$359 \leq T_{sh} \leq 407$	$361 \leq T_{sh} \leq 376$	$353 \leq T_{sh} \leq 386$	$353 \leq T_{sh} \leq 385$	± 0.4 K
Rotational Reynolds number	Re_ϕ Eqn. 1	$4 \times 10^5 \leq Re_\phi \leq 5 \times 10^6$	$9.5 \times 10^5 \leq Re_\phi \leq 2 \times 10^6$	$4.7 \times 10^5 \leq Re_\phi \leq 5 \times 10^6$	$4.7 \times 10^5 \leq Re_\phi \leq 4.1 \times 10^6$	$[8 \times 10^6]$
Axial Reynolds number	Re_z Eqn. 2	$4.2 \times 10^4 \leq Re_z \leq 1.9 \times 10^5$	$4.8 \times 10^4 \leq Re_z \leq 1.5 \times 10^5$	$43.4 \times 10^4 \leq Re_z \leq 2.0 \times 10^5$	$4.7 \times 10^4 \leq Re_z \leq 1.8 \times 10^5$	$[2 \times 10^5]$
Rossby number	Ro Eqn. 3	$0.29 \leq Ro \leq 8.6$	$1.27 \leq Ro \leq 5.1$	$0.14 \leq Ro \leq 6.8$	$0.27 \leq Ro \leq 6.8$	$[1.5]$
Disc Buoyancy parameter	$\beta \Delta T_{av}$	$0.04 \leq \beta \Delta T_{av} \leq 0.13$	$0.04 \leq \beta \Delta T_{av} \leq 0.08$	$0.017 \leq \beta \Delta T_{av} \leq 0.09$	$0.02 \leq \beta \Delta T_{av} \leq 0.09$	$[0.16]$

The steady-state temperature distribution in a disc obeys the 2D conduction equation

$$\frac{\partial^2 T}{\partial z^2} + \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} = 0 \quad (8)$$

The method used to obtain the heat transfer results assumes that the disc has an adiabatic mid-plane. The heat fluxes are then obtained from differentiating the radial temperature distribution. This may be seen from integrating Equation (8) over the half thickness of the disc, $t/2$, with an adiabatic condition at $z = 0$ ($q_{z=0} = 0$) and at the disc surface $z = t/2$, $q_{z=t/2} = -k(\partial T / \partial z)_{z=t/2}$, giving

$$q_{z=t/2} = \frac{kt}{2} \left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right] \quad (9)$$

A radiation correction has been applied to all the data. This is based on a grey body method (see Long^[19] (1999)) applied to the same number of surfaces as there are surface grid nodes. The view factors are obtained from the catalogue of Howell^[20] (1982) and with reference to the original articles cited in the catalogue. It is worth noting that typographical errors do exist in some of the view factor correlations and it is worth a cautionary note here to remind the reader of this fact. The radiative heat flux obtained from this correction is typically less than 10% of the measured total heat flux.

The average Nusselt numbers Nu_{av} are defined as

$$Nu_{av} = q_{av} b / \Delta T_{av} k \quad (10)$$

The local temperature difference, ΔT , is the appropriate surface to fluid value ($\Delta T = T_s - T_{ref}$) and,

$$T_{ref} = T_{in} + \frac{\Omega^2(r^2 - a^2)}{2C_p} \quad (11)$$

The thermal conductivity, k , is evaluated at T_{ref} (defined above where T_{in} is given by the temperature of air temperature denote an average over an area, A , between radius r_i and b , e. g. thermocouples No. 32 and 33. The subscript 'av' is used to

$$q_{av} = \frac{2\pi}{A} \int_{r_i}^{r_o} q r dr \quad (12)$$

$$\Delta T_{av} = \frac{2\pi}{A} \int_{r_i}^{r_o} \Delta T r dr \quad (13)$$

The integrals in Equations (12) and (13) were calculated numerically using Simpson's rule.

An uncertainty analysis has been applied to the data presented here. This calculates the Nusselt numbers using measured values of surface temperature that also include a randomly selected component of the estimated uncertainty (± 0.43 K). The process is repeated 100 times and a statistical variation in the Nusselt numbers is obtained. These results suggest a typical variation of $\pm(10 \sim 20)\%$ in the average Nusselt number.

It is worth noting that the method used to obtain the heat transfer rates is strongly dependent on the quality of the surface temperature measurement. Thermocouples were used to measure the surface temperatures because physical access was limited which would make the use of thin film gauges impractical. The temperatures, the range of variation and optical access also made the use of liquid crystals impractical; thermography was not possible because of restricted optical access. While it may be possible to reduce this level of uncertainty by a redesign of the test rig, and not without some element of risk, the results are of significant value in providing data in a region previously inaccessible.

2 LDA velocity measurements

Simultaneous measurements of the axial and tangential velocity components V_z and V_ϕ were made inside the test rig using the LDA system described in Section 2. Optical access into the rig is provided through mounting the LDA probe inside the non-rotating drive shaft. The LDA beams are directed into the cavities using a mirror also located inside the shaft. The location of the probe volume inside the rig can be changed by moving the mirror and the probe. Axial movement of just the probe changes the radial location of the probe volume within a particular cavity, moving both mirror and probe changes the axial position. For all the measurements presented here, the mean value of velocity is estimated from a minimum sample size of 100, although

typically the sample size was 5 000.

The axial component of velocity measured inside the cavity was found to be close to zero for all the conditions investigated. The measured values of axial velocity were in the annular passage between disc bore and the drive shaft were consistent with the value of the bulk mean velocity. Figure 3 shows the variation of the radial distribution of normalised tangential velocity ($V_\phi/\Omega r$) for the wide annular gap ($d_h/b = 0.164$) tests with $Ro = 0.27, 0.53, 0.94, 1.67$ and 3.6 . These measurements were made in the mid-plane of Cavity 3. However, measurements made in the mid-plane of Cavity 2, gave similar results. Measurements were also made close to the disc surfaces and these too gave similar values. The outer radius of the shaft is located at $r/b = 0.236$ and the bore of the discs at $r/b = 0.318$. For $Ro \geq 0.53$ the tangential velocity exceeds solid body rotation in the region $r/b < 0.4$ and then decreases to $V_\phi/\Omega r = 1$ further into the cavity. As the Rossby number is reduced, there is a tendency for the tangential velocity to behave less like a free vortex (where $V_\phi \propto 1/r$) towards the inner radius. The behaviour of the tangential velocity profile with Rossby number in these tests is consistent with the measurements of Farthing et al.^[2] (1992a), who found that reducing the Rossby number reduces the peak value of the normalised tangential velocity near the disc bore.

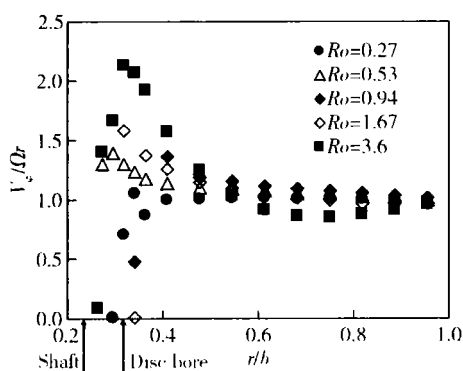


Fig. 3 Wide annular gap ($d_h/b = 0.164$), variation of radial distribution of non dimensional tangential velocity with Rossby number.

A similar plot obtained from measurements made in the test rig with the narrower annular gap ($d_h/b = 0.092$) is presented in Figure 4. It is clear that there is a significant difference between these results and those shown in Figure 3 for the wide annular gap. For the narrow annular gap, the values of $V_\phi/\Omega r$ are seen to be always lower than solid body rotation and increase to $V_\phi/\Omega r \approx 1$, further into the cavity. Since the size of the annular gap tested differs by a factor of almost two, it is not unreasonable to expect that this does have an influence on the flow inside the cavity. In this case a reduction of the annular gap between the shaft and disc bore attenuates the influence of the throughflow on the tangential velocity inside the cavity.

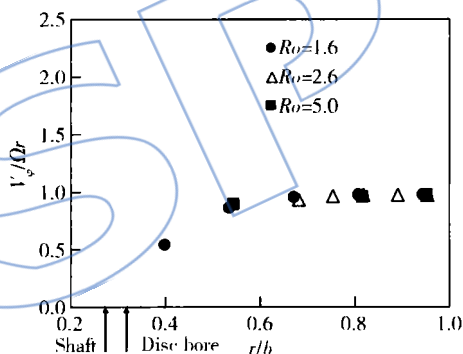


Fig. 4 Narrow annular gap ($d_h/b = 0.092$), variation of radial distribution of non dimensional tangential velocity with Rossby number.

Some support for this can be found in the measurements of Owen and Powell^[8] (2004) in a rig with an annular gap of $d_h/b = 0.026$ ($r_s = 145.2$ mm, $a = 150$ mm, $b = 371$ mm, $s = 75.2$ mm). These also show the behaviour noted in the current narrow annular gap data. The detailed measurements of Pincombe^[21] (1983) display two different types of behaviour corresponding to free and forced vortex flow. For $G = 0.53$, these differences were attributed to a change from Type 1a to Type 2a (non-axisymmetric to axisymmetric) vortex breakdown. This could be of direct relevance to the present measurements; the geometry of the cavity affects the mechanism of vortex breakdown which in turn affects the tangential velocity distribution inside

the cavity.

3 Disc heat transfer

Figure 5 shows the measured and fitted surface temperature distributions for $d_h/b = 0.164$ at $Re_\phi = 1.95 \times 10^6$ and $Re_z = 8 \times 10^4$ ($Ro = 0.79$). A profile of the disc shape is included under the horizontal axis for reference. The fitted temperature distributions shown in Figure 5 appear to be a reasonable fit to the measured data. It should be noted that the temperatures measured at the disc bore and on the disc cob ($r/b = 0.32$ and $0.32 < r/b < 0.4$, respectively) are not used in obtaining the radial temperature fit. This is because of the difficulty in prescribing how the surface temperature will vary over a region of varying disc thickness. It is evident from Figure 5 that there is only a very small temperature difference across the axial thickness of the disc. For this reason, and considering the adiabatic mid-plane assumption implicit in the conduction solution method, it is probably unwise to draw too much from the local Nusselt numbers. But the average Nusselt numbers are expected to provide reasonable estimates of the heat transfer from the discs.

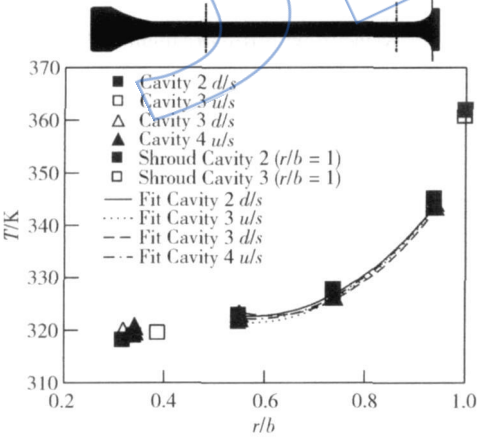


Fig. 5 Measured and fitted surface temperatures for $d_h/b = 0.164$ with $Re_\phi = 1.95 \times 10^6$, $Re_z = 8 \times 10^4$, $Ro = 0.79$.

The variation of average Nusselt number, Nu_{av} , with axial Reynolds number, Re_z , for $d_h/b = 0.092$ (narrow annular gap) and for various values of rotational Reynolds number, Re_ϕ ($Re_\phi =$

$0.5 \times 10^6, 10^6, 2 \times 10^6$ and 4×10^6), is shown in Figure 6. Values of Nusselt number are shown for both Disc 2 and Disc 3 and there appears to be little significant difference between these. There also appears to be a clear trend increasing the axial Reynolds number increases the average Nusselt number. For the larger values of Re_z ($Re_z > 10^5$) and the lower values of Re_ϕ ($Re_\phi < 10^6$), i.e. the largest values of Rossby number, this increase is more noticeable. There is some evidence, at $Re_z < 10^5$, to suggest that increasing the rotational Reynolds number increases the average Nusselt number and the results for $Re_z > 10^5$ suggest the opposite is true as increasing Re_ϕ actually decreases the Nusselt number. This is consistent with the findings of Bohn et al.^[22] (1994), who also noted the attenuating effect of the Coriolis acceleration.

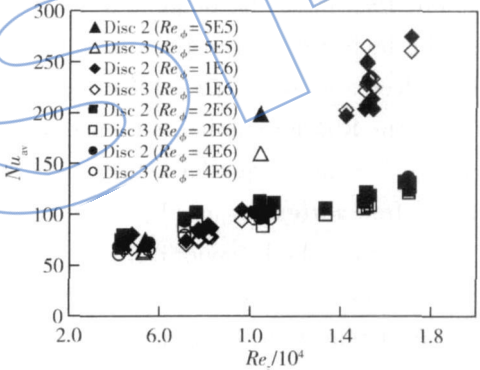


Fig. 6 Narrow annular gap ($d_h/b = 0.092$), variation of Nu_{av} with Re_z for various values of Re_ϕ .

A corresponding plot, but for the wide annular gap data, is shown in Figure 7. The overall impression is that for the same value of Re_z and Re_ϕ , the average Nusselt number from the disc surface is increased by increasing the size of the annular gap between the shaft and disc bore. The behaviour shown in Figure 6 where at the larger values of Re_z increasing Re_ϕ actually decreases the Nusselt number does not seem to occur with the wide annular gap. This and the higher overall level of heat transfer must be caused by a difference in geometry. Equal values of Re_z imply that the average velocity of the throughflow is lower in the wider annular gap. Consequently there will be a greater tendency for the flow to

penetrate the cavity and interact with the disc surfaces. If this is the case, then a greater sensitivity to the rotational Reynolds number may be expected to be shown in Figure 7 compared to Figure 6. It is possible that this is reflected in the data but the results are far from conclusive.

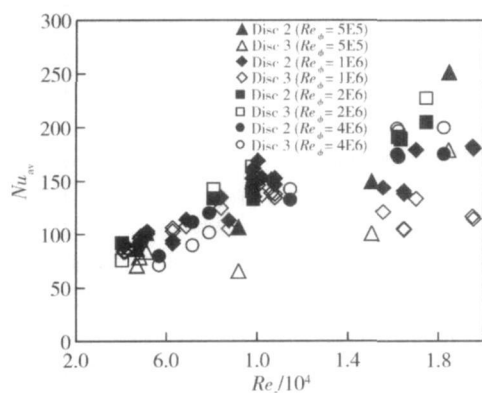


Fig. 7 Wide annular gap ($d_h/b = 0.164$), variation of Nu_{av} with Re_z for various values of Re_ϕ .

From the preceding figures, it is apparent that a simple plot of average Nusselt number against axial Reynolds number does not adequately collapse the heat transfer data. A number of attempts have been made at correlating the average Nusselt numbers and the most promising of these uses the parameter $Nu_{av}/Re_\phi^{1/2}$ plotted against the Rossby number, Ro . It is worth noting here that a previously used parameter, the buoyancy number, and the Rossby number differ through the inclusion of the dimensionless temperature $\beta\Delta T_{av}$ in the buoyancy number. The Rossby number has been chosen for use here as this was found to give a small reduction in the scatter of the data. The parameter $\beta\Delta T_{av}$ was varied ($0.04 < \beta\Delta T_{av} < 0.13$ for $d_h/b = 0.092$ and $0.017 < \beta\Delta T_{av} < 0.09$ for $d_h/b = 0.164$) but not in any systematic sense. Further work is therefore required to establish the relevance of this parameter in characterising the disc heat transfer results.

Figure 8 shows the variation of $Nu_{av}/Re_\phi^{1/2}$ with Rossby number for both sets of data. For clarity only the results for Disc 2 are shown but as previously noted those for Disc 3 are similar. For $Ro < 4$, the increase in heat transfer which

results from a wide annular gap between shaft and disc bore is easily seen in this figure.

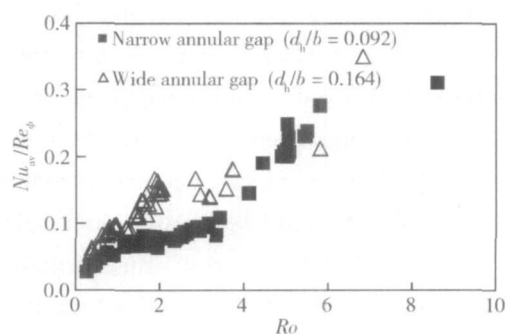


Fig. 8 Correlation and comparison between average Nusselt numbers for the narrow and wide annular gaps

4 Conclusions

This paper has presented results from tests carried out on a heated multiple rotating cavity test rig. This is designed to be a 70% full scale replica of a high pressure compressor internal air system of an axial gas turbine engine. The main interest is the flow in the cavities formed between the adjacent heated disc surfaces and the interaction of this flow with the central through-flow of cooling air. Two different build of the test rig have been investigated, these differ by the size of the annular gap between the central drive shaft and the disc bores.

Measurements of heat transfer and flow velocity were made inside the rig for a range of non-dimensional conditions that are representative of those found inside an engine (values of rotational Reynolds number up to $Re_\phi = 5 \times 10^6$ and axial Reynolds numbers up to $Re_z = 2 \times 10^5$). Heat transfer measurements from the disc surfaces were made using a conduction solution method. Measurements of tangential velocity inside the cavities were made using an LDA system, with optical access being provided through the drive shaft.

The main conclusions from this work are summarised below:

Although it was not possible to make measurements inside the disc boundary layers, the tangential velocity does not appear to vary across

the axial width of a cavity. The axial velocities inside the cavity are close to zero. For the wider annular gap between shaft and disc bore, the maximum value of $V_{\phi}/\Omega r$ occurs close to the inner radius of the cavity and its magnitude increases with increasing Rossby number. The non-dimensional tangential velocity decreases from this maximum value to solid body rotation $V_{\phi}/\Omega r = 1$, as $r/b > 0.6$. For the narrow annular gap, the non-dimensional tangential velocities tend to not show a maximum until solid body rotation is achieved near the outer radius. This difference in radial variation in tangential velocity is attributed to a difference in the influence of the central throughflow on the flow inside the rotating cavities and a narrow annular gap attenuates this interaction.

The disc heat transfer also exhibits a dependence on the size of the annular gap. There is a stronger dependence on axial Reynolds number compared with rotational Reynolds number; increasing Re_z tends to increase the average Nusselt number.

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