INVESTIGATION OF THE FLOW AND HEAT TRANSFER IN A LOW PRESSURE TURBINE INTERDISC CAVITY WITH SKEWED RADIAL JETFLOW

Alexander V. Mirzamoghadam
BMW Rolls-Royce AeroEngines
Heat Transfer
Eschenweg 11
15827 Dahlewitz (Berlin)
Germany
(49-33708) 61102
Fax (49-33708) 63011
mirzamal@bmwrr004.mhs.compuserve.com

ABSTRACT

Heat transfer in the low pressure turbine interdisc cavity of an aero gas turbine engine with a closed rotating outer-rim and forced radially outward jetflow directed along the downstream disc-cob front face was partially investigated by experiment and theory as part of an advanced cooling design concept study. Within the interdisc cavity, several metal temperature transient and steady state measurements in the circumferential direction as well as the 2-D axisymmetric plane at rotating speeds of 1500 rpm and 7000 rpm were made. The results are based on matching the measured metal and air temperature at both speed levels as well as the transient behavior between the two speed levels to those predicted by the 2-D axisymmetric transient thermal model. A qualitative description of the 3-D nature of the flow field is given with the aid of CFD studies. The results indicate that the skewed forced jetflow produces a stronger variation to the level of heat transfer at high rotating speed. The jetflow partially penetrates outward through the cavity providing enhanced free-disc forced convection heat transfer (approximately 25%) at high rotating speed to about 70% of the downstream disc radial length only. Toward the rim subcavities, the level of heat transfer drops considerably compared to that of a free-disc and heat transfer along the hot rotating rim and colder flange surfaces are described by flat plate natural convection. The jetflow exits the cavity at the bore of the upstream disc by turning forward within the cavity, substantially reducing the level of heat transfer along the diaphragm of this disc, and providing forced convection heat transfer on the cob whose level is higher than the rest of the upstream disc represented by natural convection. At low rotating speed, a dominating mixed convection mechanism is evident throughout the interdisc cavity with significantly lesser variation in heat transfer. A critical Gr/Re² value of 0.02 was established as the minimum where both natural and forced convection are important. The resulting behavior in local heat transfer coefficient and Nusselt number along the hot bolted rim and the discs are compared with those of previous investigators looking at heat transfer in a rotating cavity.

NOMENCLATURE

A
a
b
C
Cq
CW
CWR
Cw
Gr
k
L
m
n
n*
Nu
Nua
NuA
P
P
R

Surface area
Disc inner radius
Disc outer radius at the rim
Constant in the power law surface temperature distribution
Specific heat at constant pressure
Nondimensional massflow parameter ( = \dot{m} / \mu b)
Massflow parameter based on local radius ( = \dot{m} / \mu R)
Bolt diameter
Rim diameter at bolt location
Radiation overall view factor
Interdisc cavity gap ratio ( = S/b)
Local Grashof number (= \rho g \Delta \rho \bar{R}^3 \bar{V}^2)
Local heat transfer coefficient
Thermal conductivity of fluid
Characteristic length = R for disc and = A/P on rim
Massflow rate
Exponent in power law equation for surface temperature
Number of bolts on flanged rim
Local Nusselt number (= hL/A)
Nusselt number based on local radius
Nusselt number based on A/P
Pressure
Perimeter

Presented at the International Gas Turbine and Aeroengine Congress & Exhibition
Birmingham, UK — June 10-13, 1996
The amount is balanced against hot gas ingress due to rotation at which the superimposed compressor cooling flow is rotor (or rotor-stator) cavities depends on the amount and flow-field and effectiveness of cooling (or heating) in turbine rotor-leads to a reduction in efficiency. In particular, the nature of the mainstream flow, as well as heat conduction from the disc/rim temperature distributions and cooling flow temperature rise. In existing gas turbines, the amount of air required to cool the disc/rim temperature distributions and cooling flow temperature rise. In existing gas turbines, the amount of air required to cool the temperatures.

Previous heat transfer studies of flow in a rotating cavity have focused mainly on three scenarios. a) Symmetrically heated rotating discs and rim with forced coolant radial through-flow (symmetrically inward/outward flow); b) Symmetrically heated rotating discs and closed rim with forced coolant axial throughflow; c) Heated and gas-filled (i.e. closed) rotating cavity, with a recent publication by Long et al. (1995) investigating buoyancy dominated flow and heat transfer in an asymmetrically heated rotating cavity. Gan et al. (1993) provides a review of the experimental and theoretical studies related to heated rotating cavities with either radial or axial throughflow. In particular, measurements of the local Nusselt number in the radial direction as a function of disc temperature gradient and rotational Reynolds number were compared against theoretical computations using the integral method and elliptic/parabolic solvers. For radially outward flow and increasing disc temperature toward the rim, they reported that the local Nusselt number increases and then decreases along the disc. Kim et al. (1995) performed heat transfer measurements on the effect of surface heating condition on the local heat transfer coefficient for enclosed corotating discs with axial throughflow. They found that the local Nusselt numbers increase with increasing axial Reynolds number (more jet driven recirculation flow) for a fixed rotational Reynolds number. Furthermore, the effect of the disc heating condition (same on both discs) decreases with increasing rotational Reynolds number for a fixed axial Reynolds number, but increases with increasing axial Reynolds number for a fixed rotational Reynolds number. Their study concluded that the local Nusselt number followed the trend of turbulent natural convection on a vertical hot wall. Bohn et al. (1994) performed studies of heat transfer in closed gas-filled rotating annuli with an imposed axial heat flux (i.e. hot upstream and cold downstream disc walls only) and compared that study with their previous work based on an imposed radial heat flux (i.e. heating the outer and inner cavity drive-arms only). They reported that for the former, the heat transfer strongly depends on the Reynolds number whereas, in the latter case, it is mainly controlled by the Rayleigh number.

A new cooling design concept for the LPT interdisc cavity has been studied which is different from any previous turbine disc cavity cooling system reported in the open literature. Figure 1 illustrates a schematic of the LPT module geometry and secondary air flow system. The nondimensional inner radius ($R^* = a/b$) of discs 1 & 2 is respectively 0.55 and 0.52. The non-dimensional radial length of the cavity, ($b-a)/b$, along disc-1 is 0.45 and for disc-2 it 0.48. The gap ratio ($G$) is 0.26 based on the largest gap between the two discs. The non-dimensional radial length of the flange at the center of the rim is 0.082. The thickness ratio of the cob to diaphragm is 5 for disc-1 and 3.5 for disc-2. The controlling feature of this particular rotating cavity is that the rotating rim is practically closed for through-flow. The rotating rim connects the two discs via a bolted flange around the circumference thereby dividing the rim area into two subcavities. The main forced cooling jetflow which originates from an off-take of compressor air, purges the interdisc and drive-arm cavities before passing through a labyrinth seal located on the forward cob face of disc-1. In addition a small quantity of air enters the disc-2 drive-arm cavity via the drive-arm labyrinth seal and combines with the main jetflow. Allowance is made for some hot gas ingestion flow to mix with the total air flow which together
make-up the entrained boundary layer flow and the core flow in the forward cavity of disc-1. Prior to merging with the annulus gas at the NGV-1 blade platform area, part of this radially outward flow is drawn into the subcavities of the interstage seal via the bucket groove area formed between the blade fir-tree and disc live-rim area of disc-1. These two cavities draw in their respective amounts of ingestion flow based on the cooling flow provided, the hot running gap-size of the interstage seal, and the restriction of the platform regions. Cooling flow is supplied to the rear cavity of disc-2 via the EGV diaphragm. This amount of flow merges with the annulus gas after mixing with hot gas ingestion and exchanging heat with the entire disc-2 rear surface.

Referring to Fig. 1, five equally spaced discreet stationary jets were to deliver the required main cooling air by splitting the flow roughly 55% and 45% into the interdisc cavity along-side disc-2 and the cavity between disc-2 and the drive-arm, respectively. The orientation of the 45% flow along disc-2 cob center was established from preliminary CFD analysis (see Fig. 2) to move forward and radially outward along the cob and combine with the rest of the 55% forced jetflow. This flow assumption was a simplification of the actual CFD flow field solution which contains vortices around the circumference in the drive-arm corner of the disc. As a thermal model was required for this design study to assess disc life, the complexity of the flow-field particularly in the interdisc cavity and the forward cavity of disc-1 was realized from the onset. Moreover, sensitivity studies of various flow-field assumptions in the interdisc cavity revealed significant variability on predicted disc temperature and temperature gradient. Transient metal temperatures at various locations on the discs as well as steady state air temperatures in the cavities surrounding the interdisc cavity were therefore measured for a square cycle engine run. This cycle consists of a steady state low speed engine run followed by a fast acceleration to the higher speed level. The engine is quickly decelerated back to the low speed after reaching steady state at the high speed condition. The measurement cycle and operating conditions consisted of a rotational disc speed parameter (ω/√ T) and massflow parameter (m/√ T/P, where P is in bar) of respectively 75 and 1.6 at low speed and 276 and 2.89 at the high speed level. The thermal model was then reconstructed in order to reproduce the transient and steady state temperatures, and this paper reports the status of the findings with respect to the resulting massflow distribution and associated heat transfer mechanisms in the interdisc cavity only.

MODEL DESCRIPTION

The process in developing the thermal model involves data input and applying boundary conditions to a 2-D axisymmetric finite element model using the proprietary thermal-mechanical analysis code developed at Rolls-Royce Plc.

The variation of the mean total relative fluid temperature (i.e. the adiabatic disc temperature, Owen & Rogers, 1989) in the boundary layer along the disc is derived based on an energy balance for the fluid in a finite source.

\[ \dot{m}_c \left( \frac{dt}{ds} \right) = \left( \frac{dW_{visc}}{ds} + dW_{ph}\right) - \frac{h}{C_p} \left( T - T_m \right) \]

where, \( \dot{m}_c \) (dWvisc/ds), the frictional heating due to radial flow in a vortex as well as compressive work in units of KJ/mm (Chew 1985) is defined by,

\[ \frac{dT_{visc}}{ds} \left( \frac{\text{KJ}}{\text{mm}} \right) = \left( \frac{\text{KJ}}{\text{mm}} \right) \frac{dR}{ds} \]

and, alternatively, it has been described by the empirical universal windage heat pick-up for a disc, cylinder or cone (Tuley 1978). In Eq. (1), \( \dot{m}_c \) (dWvisc/ds) = \( dT_{visc}/ds \) (KJ/mm) represents an additional heat source term for fluid heating (i.e. as a result of rotating protrusions). The temperature rise per unit length was evaluated using the empirical relation reported by Millward & Robinson (1989). It is repeated here for clarity.

\[ W_{visc} = 2.3 \left( C_{w,R}/Re_a \right)^{1.4-1.05Re_a} \left( \rho/\mu \right)^{0.34} \rho \omega^3 R^3 A_n^{*2} \]

Based on the solid finite element model, Eq. (1) is integrated an element face at a time from the start to the end of the region of interest with user defined massflow rates (in the radial and axial directions which are representative of the effective boundary layer massflow) and inlet temperatures as the initial conditions. The convective heat transfer coefficient in Eq. (1) which varies with time is also user defined.

MODEL VALIDATION PROCESS

The thermal model validation study within the interdisc cavity initially assumed free disc forced convection heat transfer heat transfer (with n = 2 ) for the flow along the boundary layer of each disc, Dorfman (1963) (see Eq. (4)). Dorfman’s solution of the energy and momentum integral equations is based on a power law distribution of surface temperature, i.e. \( T_m = c(R_s)^n \). The factors become respectively 0.0267 and 0.616 for turbulent and laminar flow.

\[ N_u = 0.0197(n+2.6) P_{t+0.6} R e_a \quad R e_a > 3 \times 10^5 \]

\[ N_u = 0.308(n+2) P_{t+3.3} R e_a \quad R e_a < 2.4 \times 10^5 \]

Since the above forced convective heat transfer correlations (Eq. (4)) are based on Reynolds numbers which are rotational speed dependent (and not based on local relative velocity), iterations were performed on the effective fluid temperature in the radial boundary layer based on the assumed effective massflow and heat transfer enhancement or reduction factor. Those rotating surfaces whose temperature is higher than the fluid temperature (i.e. upper disc diaphragm, rim subcavities) induce buoyant currents directed radially inward and averaged natural convection correlations for flow over a heated surface on the horizontal sides of the rim section, Fishbenden & Saunders (1950) (Eq. 5), and along the radial (i.e. vertical) sides of the flange section, McAdams (1958) (Eq. 6), were used. The Grashof number is based on the centrifugal acceleration, and therefore, a similar iteration as for the discs was required for the axial and radial sides of the rim.

\[ N_u = 0.14 \quad R a > 2 \times 10^7 \]

\[ N_u = 0.54 \quad R a > 2 \times 10^7 \]

\[ N_u = 0.129 \quad R a > 10^9 \]

\[ N_u = 0.59 \quad R a > 10^9 \]
As the objective of this paper is to focus on the stabilized fluid flow and heat transfer mechanisms within the interdisc cavity at two different speed levels, little will be reported about the physics associated with the surrounding cavities and during transient mode.

RESULTS

The model validation process led to a distribution of fluid energy balance within the interdisc cavity satisfying Eq. (1) and reproducing the measurements at steady state and transient conditions within a specified tolerance of 20 K. The resulting flow distribution and heat transfer were guided by comparing the relative temperature difference between entering coolant flow and local disc temperature. The steady state circumferentially averaged temperature measurements at high and low rotational speed are presented in Fig. 3 for both discs. The measured temperatures indicate a reversal in order between the two speed levels. At high speed disc-1 is hotter whereas at low speed it is cooler than disc-2. Referring to Fig. 3, the radial temperature profiles of both discs within the interdisc cavity limit can be best represented by a cubic polynomial, (see Eqs. 7(a) and 7(b) for the high speed condition).

\[
T_{01}(K) = 1210 - 2356 R^* + 3297.5 R^*^2 + 1421 R^*^3
\] (7-a)

\[
T_{02}(K) = 415.7 + 922 R^* - 1352.6 R^*^2 + 730 R^*^3
\] (7-b)

Comparison of the radial surface temperature behavior of both discs at the two speed levels (Fig. 3) indicates that for disc-1 power law equations can be approximated with lesser error than disc-2.

\[
T_{01}(K) = 731.4 (R^*)^{0.142} \quad \text{high speed}
\] (8-a)

\[
T_{02}(K) = 500.9 (R^*)^{0.276} \quad \text{low speed}
\] (8-b)

Flow Distribution

The effectiveness of the jet cooling system at high power can be realized by drawing a horizontal line on Fig. 3 at the metal temperature of 675 K (which is at close proximity to the average cavity air temperature) and noting the locations on the discs with temperatures below this limit. These locations demonstrate the effect of enhanced cooling and for disc-2, it terminates at about 70% of the radial distance. On disc-1, however, the region is limited to the cob section. Thus, the forced jetflow penetrates the cavity along the radial coordinate and adjacent to disc-2 approximately 70% of the radial distance before beginning the process of joining the circumferential boundary layer flow, by turning forward and flowing out of the cavity along the rear face of disc-1 cob. This flow scenario is similar to the injection of a subsonic normal jet in an axial mainstream gas flow where the jet penetrates the boundary layer a certain distance according to its momentum ratio (i.e. injection to mainstream) before losing momentum and mixing with the mainstream gas flow. The preliminary 3-D CFD solution (see Fig. 2) also confirms the partial penetration of the jet, its skewedness toward disc-2, its turning forward to exit the cavity along the rear cob side of disc-1, and the radial outflow along the diaphragm of disc-1. Moreover, because the jets are discreet and distributed equally around the circumference, pockets of low velocity flow exist between jet locations alongside disc-2 as shown in the 3-D CFD produced radial velocity profile of the cavity at the approximate 70% radial distance, see Fig. 4.

The cooling air massflow was therefore set to full flow along disc-2 radial coordinate (outflow) between \( R^* = 0.61 \) and \( R^* = 0.86 \), and along the rear face of disc-1 cob (inflow). Moreover, as the jetflow turned forward, the situation is analogous to a heated rotating cavity with axial throughflow such that buoyancy induced inflow from the hotter rim section into the core is replaced by cooler upflow along the drive-arm flange connecting the two discs at the center of the rim. This flow field was verified by the preliminary 3-D CFD analysis which revealed pockets of upflow and downflow in the circumferential direction along the rim subcavities. In their study of buoyancy dominant rotating cavity flow, Long et al. (1995) have confirmed the radial inflow and outflow along the hot and cold disc surfaces, respectively. The reduced amount of flow purging the two subcavities of the rim section was arbitrarily set at 10% of the returning jetflow, although, from the preliminary CFD studies, it is not clear how much flow is involved. What is observed, however, is that the subcavity of disc-1 produces a higher and more uniform fluid total temperature whereas, the fluid temperature in the subcavity of disc-2 increases toward the disc. Based on this observation, the 10% upflow along disc-2 drive-arm flange returns as inflow split 5% on each of the buoyancy inducing horizontal and radial surfaces of the rim: whereas, disc-1 subcavity returns the 10% flow moving radially inward along the upper diaphragm.

On the diaphragm of disc-1, a smaller amount of air (5%) was allocated as any substantial quantity could only cool this section below the relatively high temperature measured. It is conceivable that the air temperature in the interdisc cavity feeding this section could be raised above the measured disc temperature through ingress of hot interstage seal gas leaking through the rim, a phenomenon observed during radial outflow in a rotating cavity by Owen et al. (1985). However, the maximum leakage inflow is estimated at 4% of the total cavity flow and any simultaneous ingress (due to high speed adverse pressure gradient) would be negligible. Figure 5 illustrates the distribution of effective air massflow in the radial and axial boundary layers of the interdisc cavity based on percentage of jet massflow. The massflow parameter (\( C_{w} \)) distribution based on outer radius and local upflow or downflow massflow rate represents the complex flow field associated with this rotating cavity. Positive massflow along disc-2 signifies radial outflow and a maximum value of 62000 along the diaphragm is reached before its reduction to 6200 near the rim and practically zero outflow on the rim. The radial inflow is designated by a negative massflow parameter along disc-1 cob and results indicate a singularity point on the diaphragm where the local radial massflow switches from inflow to outflow. The free disc entrainment flow parameter defined by Eq. (9) (Owen & Rogers, 1989) is 35000 for reference. In most turbine cooling systems, \( C_{w} \) is less than \( C_{w,tt} \) and the high value of 62000 has not been previously reported.

\[
C_{w,tt} = 0.22 \text{Re}^{0.8}
\] (9)

Heat Transfer

The resulting fluid relative mean temperature distribution along the radial boundary layer of each disc is shown in Fig. 6. At both speed levels, the fluid temperature along disc-1 is higher than disc-2 but the difference diminishes as the rim is approached. Moreover, the lower fluid temperature of disc-1 compared to disc-2 near the rim section indicates that a portion of the returning jet purges the buoyancy driven subcavity of disc-1 and this occurs between the jets in the circumferential direction. The net temperature rise of the fluid in the cavity is respectively 30 K and 25 K at high and low speed. The relatively high air temperature flow injected along the
colder cob section of disc-2 produced slightly negative local heat fluxes at steady state high power conditions.

The exponents in Eq. (8) describing the temperature behavior of disc-1 indicate a departure from the quadratic (i.e. n=2) power law assumption used in the derivation of Eq. (4) for forced convection heat transfer on a free disc. Moreover, the free disc heat transfer model is based on no axial component of fluid velocity. The presence of the skewed jet along the boundary layer of disc-2 compared to its weaker effect on disc-1 does produce axial flow velocity and confirms a greater departure from free disc heat transfer for disc-2. The rotational Reynolds number at the two speed levels is 3.2x10^5 and 3.3x10^5. The flow is turbulent within the 'free disc' boundary layer at both speeds, Owen et al. (1985). The dependence of the heat transfer coefficient factor to the surface temperature exponent, however, is rather weak in turbulent flow, according to Eq. (4). Despite the non-apparent departure from free disc flow, Eq. (4) was used along the radial direction of disc-2 up to R* = 1 with 30% enhancement along the axial cob face, 25% enhancement up to R* = 0.74, and 50% reduction along 0.85 < R* < 1.0, see Fig. 7. The enhancement and reduction in local heat transfer coefficient is attributed to the effect of the forced jet on the radial boundary layer and implies that as the jet loses momentum and the flow is hindered by a closed rotating rim, the level of heat transfer decreases. The thermal response of the rim section produced two levels of heat transfer depending upon the relative accuracy of the matched model to steady state measured temperatures versus accuracy in measured transient behavior. When the low levels of heat transfer depicted by the dashed curves of Fig. 7 (i.e. 75% reduction for both discs near the rim section) are considered along with windage heating (approximately 900 W at high power and 60 W at low power according to Eq. (3)) due to rotating bolts, the transient response between the two speed levels is more closely approached but to the expense of a departure from the steady state levels.

Referring to Fig. 7, the thermal asymmetry of the rotating interdisc cavity at high speed (Re = 3.2x10^5) is evident. Disc-2 has an increasing heat transfer coefficient up to a maximum value of 640 Wm^-2K^-1 followed by its reduction to 260 Wm^-2K^-1 as the mixed convection region of the rim is approached. Heat transfer for the radial inflow along the rear face cob of disc-1 is defined by the forced convection Nu of Eq. (4). The high temperature noted on the diaphragm of this disc, however, was attributed to the complex hot gas ingestion mixing and rotating protrusion heating of the fluid near the diaphragm in the forward cavity and hence, the heat transfer along the diaphragm within the interdisc cavity along this section should be small. In the absence of any air temperature measurement along the boundary layer, it was postulated that the flow field along the middle section of the diaphragm was caught between an inflow at the cob section and an upflow induced by natural convection near the rim originating from the returning jetflow. The low level heat exchanging process is defined by low massflow (i.e. 5%) and relatively low heat transfer as the radial velocity in this region is expected to be low. Similar disc temperature results are produced whether the convective heat transfer along this section is defined by a 90% reduction of the free disc forced convection Nu or by a 50% reduction in the natural convection Nu prediction according to Eq. (6) and recommended in this study. The maximum heat transfer coefficient occurs near the exit at the cob of disc-1 and has the value of approximately 480 Wm^-2K^-1. The trend in reduced levels of the heat transfer coefficient near the rim section of both discs is somewhat uncertain as illustrated by the dashed curves of Fig. 7 for the transient-based matched model. The expected bell-shaped Nu profile (Fig. 8) for disc-2 varies between 1500 at the cob to 2300 near the radial center of the disc. Morse & Ong (1992) predicted the local Nu for the symmetrically heated rotating cavity with radial outflow (C_w=14000, and Re=3.18x10^6). They reported a similar curve but with a maximum value of 1200. At low speed (Re = 3.3x10^5, C_w = 14700), an averaged Nu based on the length-averaged heat transfer coefficient and disc mean radius for disc-2 and disc-1 is respectively 280 and 190. These results demonstrate the strong dependence of the Nu on the rotational Re, the disc temperature level/radial profile, and the cavity air flow.

The asymmetrical effect of the radial jet on disc heat transfer was transferred to the two subcavities of the bolted flange rim by defining a 50% reduced free disc heat transfer coefficient on the upper diaphragm of disc-2 in concert with full natural convection in both subcavities according to Eqs. (5) and (6). Results are presented in Fig. 9, which illustrates the pronounced asymmetrical heat transfer behavior along the rotating and closed rim at high speed. The presence of the flank leads to a reduction in heat transfer due to the reduced temperature difference between fluid and metal. The averaged Nu along the rim section based on average heat transfer coefficient and a characteristic length of S/2 for the axial section and the radial distance of the flange for the bolted section is shown in Fig. 10. The results at low speed are consistent with those produced by Kim et al. (1993) who considered the effect of increasing the axial flow, the rotational Re, and the disc rim temperature assumption on the averaged Nu. At low speed, the jet massflow is significantly reduced which allows the interdisc cavity heat transfer mechanism along the upper radii of the discs and rim to approach the problem of a rotating cavity with high axial flow. This result is supported by examination of the mixed convection parameter which is defined by Eq. (10) based on the outer radius rotational Re and a Cr based on local radius (A/P on the rim).

Cr/Re = (R*+2)(T_m -T)/T (along the disc radial direction) (10-a)
Cr/Re = (G/2)(T_m -T)/T (along the rim) (10-b)

These equations confirm cavity geometry to be one of the main factors that influences the mechanism of convective heat transfer in a rotating cavity. A deep and narrow cavity would favor forced convection. Figure 11 illustrates the variation of the mixed convection parameter along both discs. Results indicate that a value of 0.02 for this rotating cavity should be used as a guideline to establish the contribution of natural convection to forced convection heat transfer. This implies that at low speed, thermal analysis of the interdisc cavity should be represented by stronger mixed convection along both discs.

The effect of radiation heat transfer within the semi-closed rotating cavity was also investigated. With respect to view factor calculation, the hotter rim consists of surfaces in both horizontal and radial directions which are in view of the upper and lower surfaces of both discs as well as the jet surface across the inner radius. Moreover, the higher disc-1 temperature and its direct view of disc-1 causes a cooler surface. As the averaged surface temperature of all surfaces
in view of the rim \( (T_{\text{rim}}) \) is below the rim temperature by approximately 40 K at high power, the maximum amount of radiation cooling from the rim to all other surfaces assumes a one-way radiative heat flux (based on a view factor, \( F_\alpha \), equal to the surface emissivity of the emitting surface) whose heat transfer coefficient is the heat flux gradient with respect to surface temperature. Equation (11) illustrates the relationship of the radiative heat transfer coefficient to the net heat flux versus the maximum (i.e. one-way) heat flux.

\[
h_R = F_\alpha (T_{\text{rim}} - T_{\text{surf}})^2 (T_{\text{surf}} + T_{\text{surf}}) = 4 F_\alpha T_{\text{surf}}^3
\]

At high power, it is approximately 60 W/m²K which is about 25% of the averaged rim convective coefficient. However, the maximum rim surface temperature drop due to radiation cooling was estimated to be 5 K implying that in this case convection is dominant. Similar results were obtained for radiative cooling of disc-1 and radiative heating of disc-2 within the interdisc cavity.

**CONCLUSION**

An investigation of the 2-D flow and heat transfer in the LPT interdisc semi-closed rotating cavity with a forced radial outward skewed jet along disc-2, (practically) no radial outflow through the rim, and a measured disc radial temperature distribution fitted by a third degree polynomial was performed. The two speed levels investigated demonstrate different thermal-fluid behavior within the cavity. At high speed \((Re=3.2\times10^6)\) the temperature level of disc-1 is higher than disc-2. At low speed \((Re=3.3\times10^5)\), however, disc-2 shows a higher temperature. At high speed, enhanced \(Nu\) \((\text{Gr}=2300)\) along disc-2 was attributed to the forced jet. The minimum \(Nu\) of 200 along the diaphragm of disc-1 was due to a low massflow recirculation zone formed by the returning (inflow) jet and the buoyancy induced upflow into the rim subcavity of disc-1. Along the rim section, minimum heat transfer occurs at the bolted flange location where \(Gr = 8.4\times10^{10}\) and the \(Nu\) is approximately 70. The averaged \(Nu\) increases to about 150 \((Gr = 9.0\times10^{10})\) in the subcavity of disc-2.

The mixed convection parameter reveals a value of 0.02 to be considered as the critical value for both natural and forced convection to be important. The penetration of the jet is rotational speed dependent in that at 7000 rpm. the full flow penetration is about 70% along the disc-1 radial height; whereas, at 1500 rpm, the jetflow appears to turn forward at a shorter height and cover more of disc-1 (estimated as 33%). Moreover, thermal radiation cooling and heating of surfaces within the cavity was negligible even at high power.

It is acknowledged that the conclusions drawn from this study depend upon the convective conditions imposed in the surrounding cavities. In particular, the high fluid temperature in the entrained boundary layer flow along the forward cavity of disc-1 requires further investigation related to its source (i.e. high gas ingestion or windage heating due to a rotating protrusion). The present 3-D CFD study has given qualitative insight into the flow pattern and the 2-D profile of the radial velocity. It has yet failed to produce a reasonable heat flux profile especially for disc-1.
Fig. 4 Circumferential Variation of Radial Velocity Near 70% Radial Distance
(Showing bending of jet outflow, core inflow, and pockets of low velocity between jet locations)

Fig. 5 Interdisc Cavity Massflow Distribution in R - X Plane
(% of Main Jet Air)

Fig. 6
RADIAL VARIATION OF FLUID TEMPERATURE ALONG DISC
(Showing a lower temp. for subcavity 1)

Fig. 7
Enhanced and Reduced Heat Transfer on Discs
(Dashed lines show improved transient predictions on rim section)

Fig. 8
RADIAL VARIATION OF NUSSELT NUMBER FOR EACH DISC
(Showing comparison with free disc value)
ACKNOWLEDGMENT

The author acknowledges Dr. S.J. Hiller and H.P. Sheiner for their CFD contributions to this work.

REFERENCES


