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MODELLING FLOW IN INTERNAL COOLING−AIR SYSTEMS OF GAS−TURBINE ENGINES

M. Wilson, H. Karabay, I. Mirzaee and J. M. Owen
School of Mechanical Engineering
University of Bath
Bath, UK

ABSTRACT

Recent computational research has shown that finite−volume, elliptic solution procedures are capable of predicting accurately the turbulent flow and heat transfer in rotating−disc systems with a radial outflow of cooling air, representing conditions which occur in the internal cooling systems of gas−turbine engines. These include idealised systems, such as the rotating cavity and rotor−stator system, and more realistic internal cooling−air systems in which both the disc−cooling flow and the pre−swirl blade−cooling flow are modelled.

This paper describes previous and new computational results for direct−transfer (rotor−stator) and cover−plate (rotating cavity) pre−swirl systems. In the cover−plate system, pre−swirl air flows radially outward as a free vortex between a rotating−disc and a cover−plate attached to it, while in the direct−transfer system the pre−swirl air is confined between inner and outer seals near the top of a rotor−stator system. Adiabatic calculations for both systems show the relative effectiveness of the two configurations, and validation data for flow and heat transfer is available from a purpose−built rotating−disc rig.

Rotating−disc systems involving a radial inflow of air give rise to a combined free and forced vortex flow structure which is difficult to predict with basic two−equation turbulence models. Such flows occur in a rotating cavity in which cooling air enters and leaves through a stationary casing at the periphery of the system. It is shown that a Richardson number correction to the dissipation−rate equation in the turbulence model can be used to give some improvement of the flow predictions.

NOMENCLATURE

a, b	inner, outer radius of disc
A, B	constants in equation (4)
C_{e2}	turbulence model coefficient (equation (6))
C_w	nondimensional mass flow rate $(=\dot{m}/\mu b)$
k	turbulent kinetic energy
$\dot{m}$	mass flow rate
r	radial coordinate
Re_ϕ	rotational Reynolds number $(=\rho\Omega b^2/\mu)$
Ri_{gs}	Richardson−number (equation (5))
s	axial gap between discs
T, T_o	static temperature, total temperature
U_τ	friction velocity $(=\sqrt{(\tau_w/\rho)})$
V_r,V_ϕ,V_z	time−averaged velocities in r,ϕ,z directions
x	nondimensional radius (=r/b)
y	distance normal to wall
y^+	wall−distance Reynolds number $(=\rho U_\tau y/\mu)$
z	axial coordinate
β	swirl ratio $(=V_\phi/\Omega r)$
$\overline{\Delta T}$	non−dimensional temperature rise (Table I)
ε	turbulent energy dissipation rate
λ_T	turbulent flow parameter $(=C_w Re_\phi^{-0.8})$
μ	dynamic viscosity
ρ	density
ϕ	tangential coordinate
τ_w	total wall shear stress
Ω	angular speed of discs

subscripts

b,d,p,s	blade−cooling, disc−cooling, pre−swirl, seal

INTRODUCTION

In air−cooled gas−turbine engines, air bled from the compressor is used to cool the turbine blades and the discs to which they are attached. With the increasing emphasis on engine efficiency and reliability (for both aeroengines and industrial gas turbines), more attention is being paid by designers to the improvement of internal cooling−air systems of engines, and this requires reliable data and better understanding of the flow and heat transfer processes ocurring in these systems.

Owen and Rogers (1989, 1995) give an extensive account of the modelling of flow and heat transfer between a turbine disc and an adjacent stationary casing, and between corotating turbine discs. Many features of these systems can be studied using experimental rigs of simplified geometry (compared with real engine configurations), and with axisymmetric computational models. The results of some recent experimental and computational research programmes have been described by Wilson, Pilbrow and Owen (1995), Wilson, Chen and Owen (1996) and Gan et al (1996) .

In the first part of this paper, the work of Wilson, Pilbrow and Owen (1995) on a direct transfer pre−swirl configuration is extended to introduce more recent work on an alternative cover−plate system. Computations of flow in the new system are described, and the relative merits of the two systems are assessed using adiabatic computational models. In the second part of the paper, the study recently described by Gan et al (1996), of a rotating cavity with a stationary casing and peripheral flow of air, is updated with the use of new experimental data and modification of a low Reynolds−number k−ε turbulence model. Conclusions, and an outline of further work now in progress, are then given.

PRE−SWIRL SYSTEMS

Introduction

Fig. 1a shows a schematic representation of a "direct transfer" pre−swirl rotor−stator system encountered in some engines (Wilson, Pilbrow and Owen 1995). Compressor air is used to cool the turbine disc, and angled pre−swirl nozzles in the stator, located close to the blade−cooling passages in the disc, supply cooling−air rotating at close to the disc speed: this reduces the temperature of the air relative to the disc, resulting in cooler fluid entering the blade−cooling passages. Some of the cooling air is used at the outer seal to minimise ingress of hot gas from the external mainstream, which might otherwise be drawn into the blade−cooling passages. Thermal contamination of the blade−cooling air by the disc−cooling flow is also likely to occur, as illustrated in Fig. 1a.

In some other engines, the pre−swirl nozzles are located at a lower radius, as illustrated in Fig. 1b, and the blade−cooling air flows radially outward between the hot turbine disc and a cover−plate attached to it. The cooling air removes heat from the disc before entering the blade−cooling passages, but in this system there is better sealing from the effects of ingress. The flow patterns shown in Fig. 1a and Fig. 1b are based upon the results of axisymmetric computations as described in this paper. The location of pre−swirl cooling systems in engines is illustrated in the hypothetical arrangement shown in Fig. 1c.

Direct transfer system: experimental rig and computations

Wilson, Pilbrow and Owen (1995) described an experimental rig used to perform heat transfer tests for the direct transfer configuration illustrated in Fig. 1a. The steel rotating disc could be heated at its periphery (not shown in the schematic) to about 160°C by electric heaters, and the disc was instrumented with thermocouples and fluxmeters. Sixty pre−swirl nozzles, of 2.9mm diameter, were located at a radius of 200mm on the stator, and sixty blade−cooling holes, of

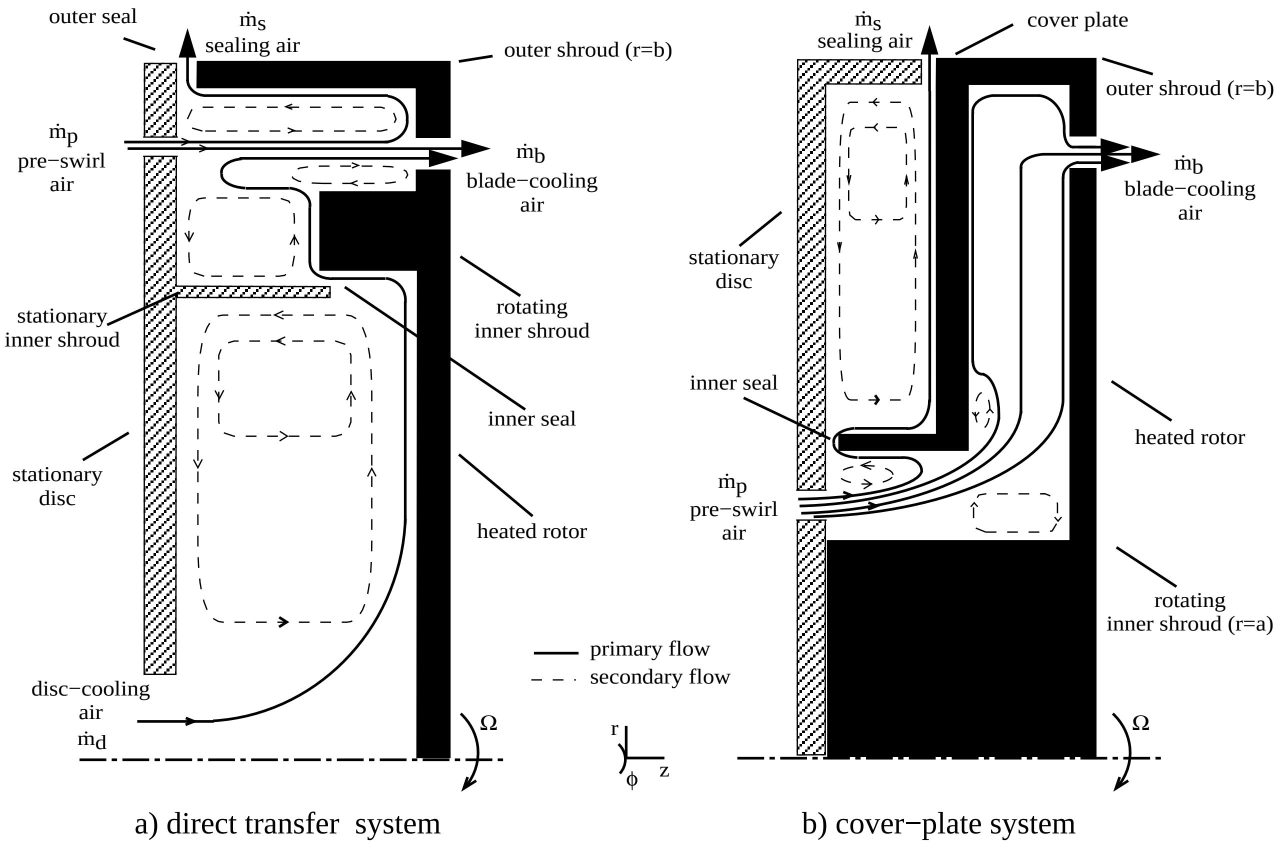


Fig. 1 Schematic diagram of two pre−swirl rotor−stator systems

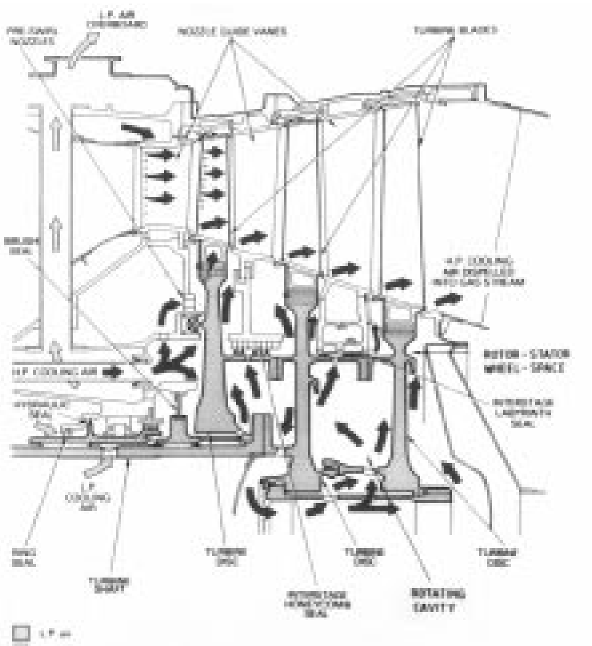


Fig. 1 Schematic diagram of a turbine cooling and sealing system
(from "the Jet Engine" − Rolls−Royce plc)

7.7mm diameter, were located on the rotor at the same radius. Thermocouples located in two of the blade−cooling holes allowed the temperature of the blade−cooling air to be measured. The parameter ranges covered in an extensive test programme were as follows:

$6 \times 10^5 < Re_\phi < 2.4 \times 10^6$
$0.8 < C_{w,p} / C_{w,d} < 16$, $0.03 < \lambda_{T,d} < 0.06$
$0.5 < \beta_p < 3.7$

Wilson, Pilbrow and Owen (1995) also gave results from axisymmetric turbulent flow computations of this system, and made comparisons with experimental results for selected tests. This code was used for all of the computations described in this paper, and some details are given below.

The 3D unsteady flow, due to the stationary pre−swirl nozzles and the rotating blade−cooling holes, was modelled as an axisymmetric, steady system with equivalent flow−area slots on the rotor and stator. Good agreement was obtained between computed and measured flow and heat transfer characteristics in the inner rotor−stator system, however the measured temperature rise of the blade−cooling air was underpredicted by about 30%. Uncertainties in the thermal boundary conditions for the inner seal, and for the pre−swirl chamber itself, made it difficult to account for this underprediction in terms of specific modelling deficiencies, and, due to the relatively complicated geometry, 3D computations have yet to be carried out for direct comparison with these axisymmetric results.

Cover−plate modifications

In a subsequent experimental programme, the rig described above was modified to study a cover−plate system as illustrated in Fig. 1b. The same rotor was used, but a new stator was fitted with 19 angled pre−swirl nozzles at a radius of 90mm, and a cover−plate made from transparent polycarbonate material was attached to the rotor by means of a carbon−fibre ring. A window in the stator allows optical access for LDA measurements of tangential velocities, in the rotating cavity between the cover−plate and the disc, and also for the testing of thermochromic liquid crystal heat transfer measurement techniques. The geometry of the cover−plate system is simple, in the region of the blade−cooling holes, compared with the direct transfer configuration. It is expected that more systematic (axisymmetric and 3D) computations will be carried out for disc heat transfer near the blade−cooling passages, and for the temperature of the blade−cooling air. Many of these experiments and computations have already been carried out, and the results will be reported in due course. Computed results for an adiabatic, axisymmetric model of this new system, using equivalent−flow area inlets and outlets as for the direct transfer system, are described below.

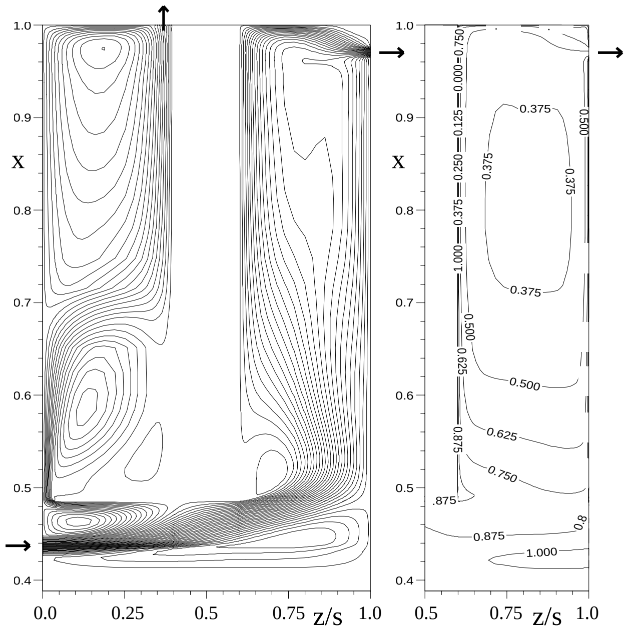
Computational method

Incompressible steady flow has been assumed, and computations carried out using an axisymmetric finite−volume solver for the discretised, Reynolds−averaged, Navier−Stokes and energy equations, using staggered storage locations for axial and radial velocity components in a cylindrical−polar co−ordinate system. The low−Reynolds−number k−ε turbulence model due to Launder and Sharma (1974) was used. This model requires very fine gridding near walls (with y+ < 0.5 for the first grid point), and multigrid convergence acceleration was used for the SIMPLEC pressure−correction algorithm. More details were given by Wilson, Pilbrow and Owen (1995).

The shrouds forming the inner seal in the direct transfer model (Fig. 1a), and the cover−plate and seal in the cover−plate system (Fig. 1b), were represented by block obstructions within the computational grid. Wilson, Pilbrow and Owen (1995) used a 115x323 (axial by radial) grid for the direct transfer model; the grid required for the cover−plate configuration was 223x223.

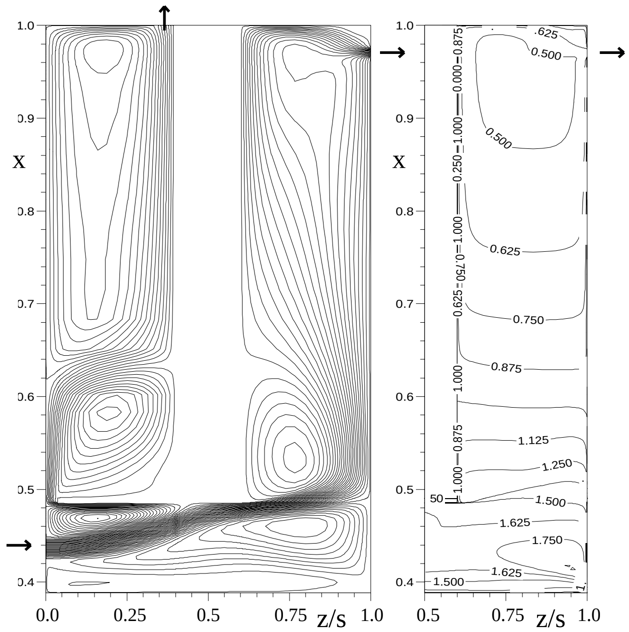
Flowfield in the cover−plate cavity: parametric effects

Computations have been carried out to study the effect of rotational speed, flowrate, and inlet swirl ratio β_p on the predicted flowfield in the rotating cavity between the cover−plate and the disc. Fig. 2 and Fig. 3 show computed streamlines and contours of normalised tangential velocity for, respectively, $\beta_p=1$ and $\beta_p=2$. For both cases, the rotational Reynolds number was $Re_\phi=1.5 \times 10^6$, the blade−cooling air flow rate was given by $\lambda_{T,b}=0.175$, and the sealing air flow rate by $\lambda_{T,s}=0.068$. The non−dimensional turbulent flow parameter λ_T used to characterise these flows is defined in the notation. The width of the axisymmetric inlet slot in the model was varied, so that the inlet flow angle remained constant for different values of β_p. The inlet flow



a) streamlines b) contours of $V_\phi/\Omega r$

Fig. 2 Computed streamlines and tangential velocities for a cover−plate system with $Re_\phi=1.2\text{x}10^6$, $\lambda_{T,b}$ = 0.27 and β_p = 1



a) streamlines b) contours of $V_\phi/\Omega r$

Fig. 3 Computed streamlines and tangential velocities for a cover−plate system with $Re_\phi=1.2\text{x}10^6$, $\lambda_{T,b}$ = 0.27 and β_p = 2

angle was 20° to the tangential direction, corresponding to the nozzles on the experimental rig.

Fig. 2a shows that the pre−swirl air is drawn into the rotating cavity and entrained as a radial outflow onto both the disc and the cover−plate. A small region of recirculating flow is formed in the cavity near the base of the cover−plate, as the flow is diverted from the axial to the radial direction. The predicted contours of tangential velocity (Fig. 2b) show that $V_\phi/\Omega r \approx 0.75$ in the vicinity of this recirculation. In the outer part of the cavity, both streamlines and contours indicate the appearance of a core of fluid, away from the disc surfaces, rotating at around 37% of the speed of the discs. In this axisymmetric model, the flow near the rotor leaves through the lower part of the blade−cooling slot, while the outflow near the cover−plate is turned along the rotating shroud and leaves the system through the upper part of the blade−cooling slot. The flow (including the path taken by the sealing air) in the rotor−stator system between the cover−plate and the stator has been computed for the sake of completeness, but the results will not be described in detail here.

Fig. 3 shows corresponding predictions for the case with β_p=2. Comparing this case with the β_p=1 case described above, the higher swirl ratio at the entrance to the cavity (β=1.4 approx. from Fig. 3b) gives rise to a larger region of recirculating flow near the base of the cover plate. Fig. 3b shows that there is radial inflow near the cover−plate when $V_\phi/\Omega r>1$ inside the cavity (x<0.6 approx.), and that there is radial outflow when $V_\phi/\Omega r<1$ (x>0.6 approx). Fig. 3b shows that a core of fluid, rotating at about 50% of the disc speed, begins to appear at a higher radius than that in Fig. 2b. For both cases, the computed mass−averaged tangential velocity at the blade−cooling slot was around $V_\phi/\Omega r$=0.5.

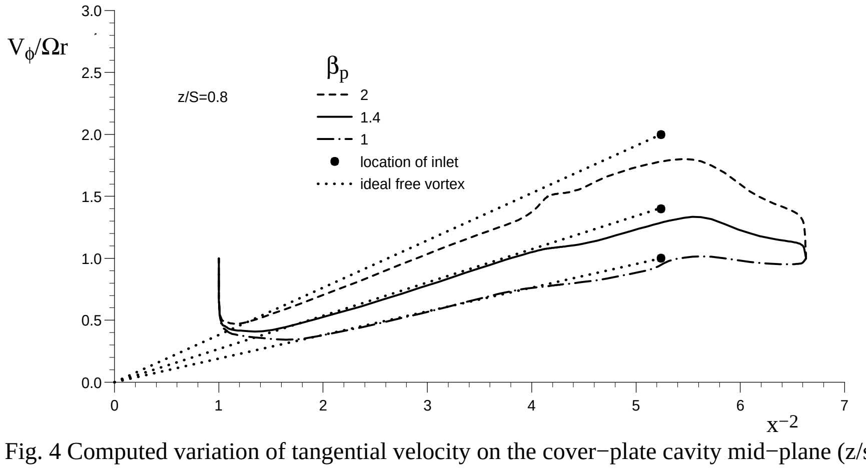


Fig. 4 Computed variation of tangential velocity on the cover−plate cavity mid−plane (z/s=0.8) for $Re_\phi=1.2\text{x}10^6$, $\lambda_{T,b}$ = 0.27 ; effect of inlet swirl ratio β_p

Free−vortex behaviour in the cover−plate cavity

For free−vortex flow between the pre−swirl nozzles and the blade−cooling passages:

$$rV_{\phi} = \text{constant} = r_p V_{\phi,p} = \beta_p \Omega r_p^2 \quad (1)$$

or

$$V_{\phi}/\Omega r = (\beta_p x_p^2)\, x^{-2} \quad (2)$$

This "ideal free−vortex" condition from equation (2) is shown in Fig.4 for three inlet swirl ratios, β_p=1, 1.4 and 2, together with the computed results on the cavity mid−plane, z/s=0.8. (i.e. mid−way between the cover−plate and the disc). The two extreme values for β_p correspond to the cases described above. For β_p=1 and β_p=1.4, the computed mid−plane results follow closely the ideal free−vortex behaviour. For the highest inlet swirl ratio, β_p=2, the computed results depart from the ideal behaviour, indicating that significant losses are associated with the flow turning into the cavity. In the outer part of the cavity, the "core rotation" effects described above can be seen in more detail for x^{-2} < 1.8 (corresponding to x > 0.74), and viscous effects become important close to the outer shroud. The centreline of blade−cooling slot is at x^{-2} = 1.06. Further parametric investigations showed that there were no significant effects of rotational speed, or flowrate, on these computed results, so that the departure from ideal free−vortex behaviour depended principally upon the prescribed inlet swirl ratio.

Results of adiabatic computations

Computations, at $Re_{\phi}=1.2\times10^6$ and assuming all surfaces to be adiabatic, have been carried out in order to compare the relative effectiveness of the two different pre−swirl systems (Fig. 1) under idealised conditions. For these calculations, the total inlet flowrate and inlet total temperature were assumed constant between the two systems. An inlet swirl ratio βp=0.93 was used for the direct transfer computation, and the same inlet tangential velocity was used in the cover−plate calculation, giving a swirl ratio β_p≈2.07 for the lower−radius inlet in this case. The flow−rates of the blade−cooling air and sealing air leaving the system were the same for both models, these being $\lambda_{T,b}$=0.27 and $\lambda_{T,s}$=0.09 respectively. Flow rates were prescribed at boundaries so that, for the direct transfer system, $\lambda_{T,p}=\lambda_{T,b}$ and $\lambda_{T,d}=\lambda_{T,s}$, and for the cover−plate system, $\lambda_{T,p}=\lambda_{T,b}+\lambda_{T,s}$.

The computed blade−cooling air temperature rise for each system (relative to the total temperature at inlet) is given in Table I. The predicted blade−cooling air temperature T_b includes the work done on the fluid in bringing it into solid body rotation, in order to simulate the conditions in the blade−cooling holes. The "ideal" values given in Table I were deduced from the steady−flow energy equation, giving:

$$\overline{\Delta T}_{ideal} = 1 - 2(r_p/r_b)^2\beta_p \quad (3)$$

(This corresponds to the result derived by El−Oun and Owen (1989), and used by Wilson, Pilbrow and Owen (1995), where r_p = r_b for the direct transfer system; the recovery factor is taken as unity here).

$\overline{\Delta T}$	direct transfer β_p = 0.93	cover−plate β_p = 2.07	cover−plate β_p = 4.6
computed	− 0.216	+ 0.162	− 0.844
ideal	− 0.860	+ 0.162	− 0.860

Table I Non−dimensional blade−cooling air temperature rise in pre−swirl systems: $\overline{\Delta T} = C_p(T_b - T_{o,p})/0.5\Omega^2 r_b^2$

For the direct transfer system result given in Table I, the computed temperature rise, ΔT=−0.22, indicates that the pre−swirl is effective in *reducing* the blade−cooling air temperature relative to the disc. The difference between the computed result and the ideal result is due to the mixing between the pre−swirl air and the disc−cooling air (which entered the system with zero swirl). This mixing reduces the effectiveness of the pre−swirl, but it is not accounted for in equation (3).

For the cover−plate system with β_p = 2.07, the computed and ideal temperature rise in Table I agree. For free−vortex flow, the tangential velocity of the air at the centreline radius of the blade−cooling passages is given, from equation (2), by $V_{\phi}/\Omega r$=0.42. In the computation, Fig. 4 shows that, although the flow deviates from free−vortex behaviour in the lower part of the cavity for this inlet swirl ratio, the tangential velocity is raised above the "ideal free−vortex" level by viscous effects in the outer part of the cavity where the blade−cooling passages are located.

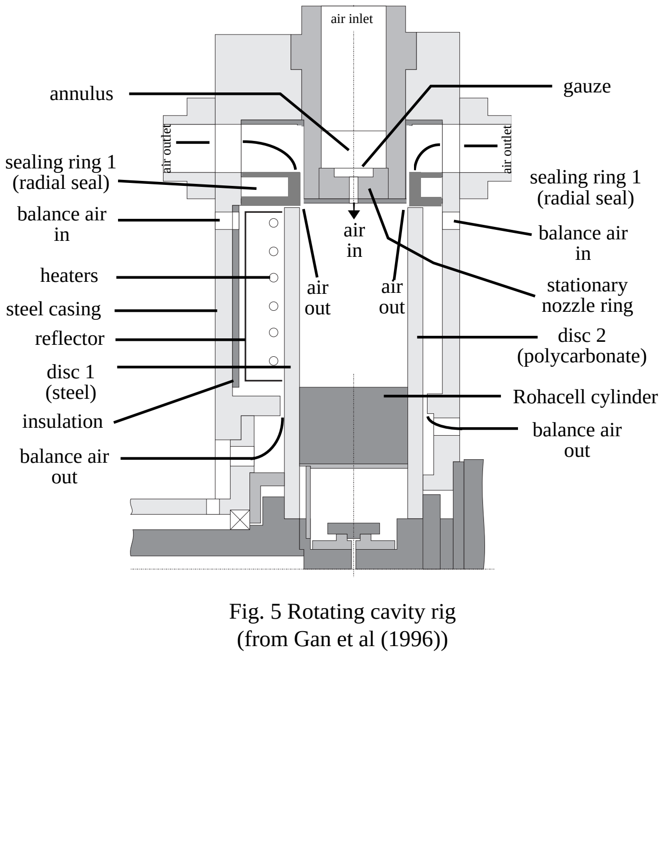


Fig. 5 Rotating cavity rig
(from Gan et al (1996))

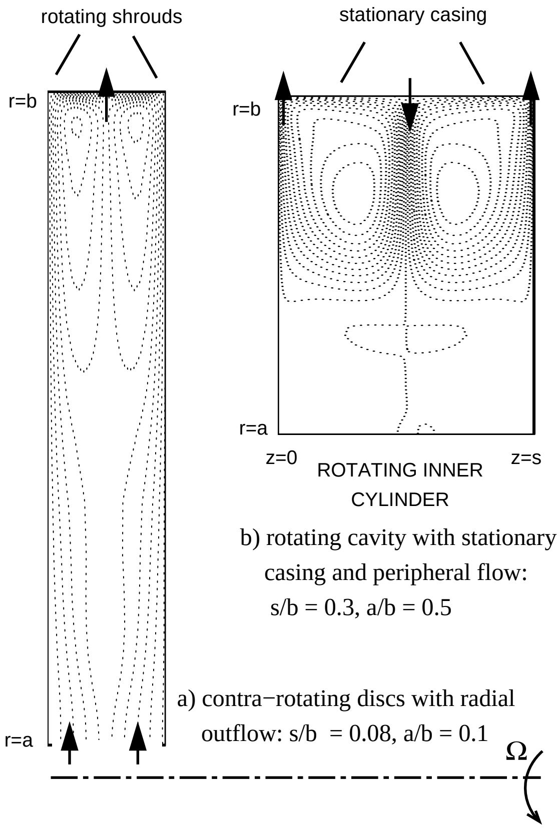


Fig. 6 Geometries studied with the rotating cavity rig, and computed streamlines for $Re_\phi \approx 1.2 \times 10^6$

The results for the cover−plate system in Table I for the case when β_p = 4.6 show that, under these conditions, the same ideal effectiveness is indicated by equation (3) as for the direct transfer system, since for these two cases the term $2(r_p/r_b)^2\beta_p$ is identical. The computed result is close to the ideal. The interpretation of pre−swirl effectiveness through analysis of adiabatic systems will be developed in a publication now in preparation (Karabay et al (1997)).

ROTATING CAVITY WITH A STATIONARY PERIPHERAL CASING

Gan et al (1996) used a rotating cavity rig, as shown in the schematic diagram reproduced in Fig. 5, to study a situation encountered in some engines where cooling air flows radially inwards, through stationary nozzles, into a rotating cavity formed by two discs and a rotating inner cylinder. Air leaves the system through clearances between the discs and the stationary casing.

The experimental rig shown in Fig. 5 had been modified from that used to study the flow and heat transfer between contra−rotating discs, as described by Chen, Gan and Owen (1995). Fig. 6 shows schematics of the different rig geometries for the two studies, together with computed streamlines. For the case of pure contra−rotation (Fig. 6a), Chen, Gan and Owen (1995) showed that heat transfer rates were lower than those obtained for the corresponding rotor−stator system (tested on the same rig). These findings were confirmed computationally, and good agreement was obtained between predicted and measured distributions of flow and heat transfer (using a computer code very similar to that described above).

For the "peripheral flow" configuration, Fig. 6b, the radial inflow of cooling air is progressively entrained into the boundary layers flowing outward on the discs. The shear between the rotating fluid and the stationary casing causes a radial inflow between the discs even when there is no cooling flow (λ_T=0). Gan et al (1996) showed that the penetration of the recirculation region increases with increasing $|\lambda_T|$ (by convention, $\lambda_T < 0$ for radial inflow), and decreases with increasing Re_ϕ. Although there are superficial similarities between the recirculating flows in the cases shown in Fig. 6a and Fig 6b, Gan et al (1996) found that the Launder and Sharma turbulence model gave inferior predictions of the measured peripheral flow behaviour, compared with the good agreement previously obtained for contra−rotating discs and rotor−stator systems.

Fig. 7, reproduced from Gan et al (1996), shows the prediction of tangential velocity on the mid−plane between

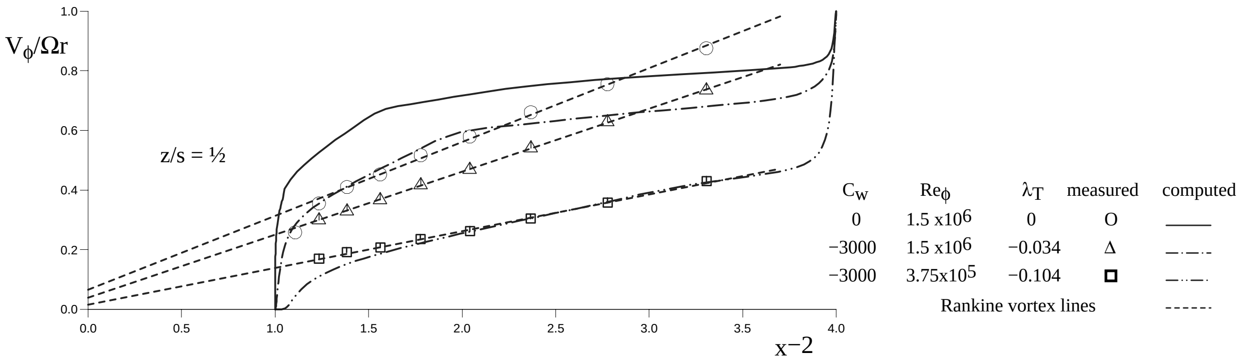


Fig. 7 Comparison between computed and measured variation of $V_\phi/\Omega r$ with x^{-2} in the mid−plane (from Gan et al (1996)

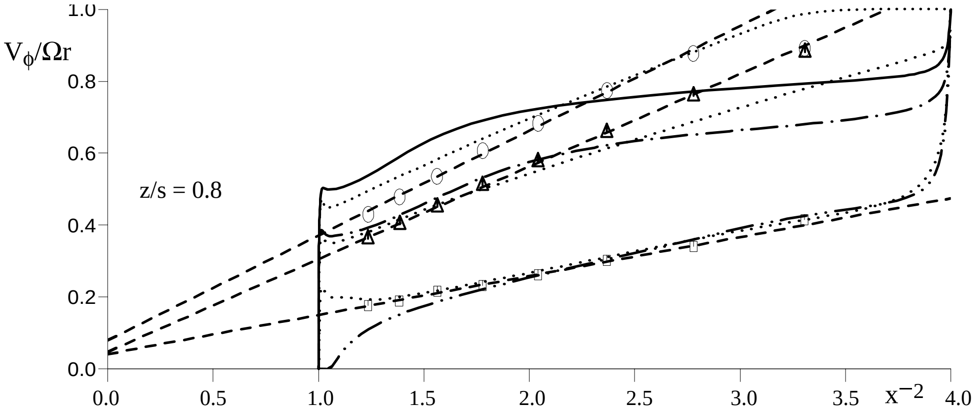


Fig. 8 Comparison between computations and smooth−surface measurements of $V_\phi/\Omega r$ against x^{-2}: effect of Richardson−number correction

KEY: as Fig. 7, with: ····· Launder and Sharma model with Richardson−number correction

the discs for λ_T = 0, −0.034 and −0.104. Although reasonable agreement was found for the prediction of radial velocity distributions in these cases, Fig. 7 illustrates that the Launder and Sharma model fails to predict the underlying "Rankine−vortex" behaviour (see Owen and Rogers (1995)) of the flow indicated by the experimental data, for which:

$$V_\phi/\Omega r = Ax^{-2} + B \tag{4}$$

where A and B are constants, but different for the three sets of measurements. For the largest of the three $|\lambda_T|$ values, the recirculations were predicted to fill the cavity. In this case the measured (and computed) flow exhibits behaviour close to that of a free vortex (as discussed above in relation to the cover−plate cavity), and the tangential velocity data are reasonably well predicted. For the two other cases shown in Fig. 7, forced−vortex type flow is predicted for the inner part of the cavity (for $x^{-2} > 2$ approx.), away from the boundary layer on the rotating inner cylinder.

<u>Effect of Surface Roughness and Richardson Correction</u>

Gan et al (1996) suggested a number of possible explanations for the poor predictions shown in Fig. 7. In the experimental rig, Rohacell material was used to insulate cylindrical surfaces for the purposes of heat transfer tests, and this gave rise to surfaces which were not aerodynamically smooth. Subsequently, several of the experiments were repeated after the stationary casing had been smoothed by varnishing and polishing the insulating material. By these means, the average roughness was reduced to approximately 8µm, compared with the 30µm reported by Gan et al (1996). The new measurements are described below.

For cases of combined free and forced vortex flow, Richardson−number based corrections have been proposed for the source term in the ε equation of the k−ε turbulence model, and several formulations have been reported in the literature (see Sloan et al (1986)) for different flow situations. Bohn, Johann and Kruger (1995) included such a correction in the computation of a rotor−stator system with a mainstream. In the present work, and following testing of a number of previously−proposed modifications, a gradient Richardson number for swirl, defined as:

$$Ri_{gs} = \frac{k^2}{\varepsilon^2}\frac{V_\phi}{r^2}\frac{\partial}{\partial r}\left(rV_\phi\right) \tag{5}$$

was used to modify a source term coefficient in the transport equation for ε as follows:

$$C_{e2,eff} = C_{e2}(1 - C_{gs}Ri_{gs}) \tag{6}$$

For the present study, the value of the coefficient C_{gs} was optimised numerically, and a value C_{gs}=0.7 was obtained. In the basic model, C_{gs}=0 and C_{e2}=1.92 is the value usually taken for this empirical coefficient. In the modified model, the variation of the coefficient using equation (6) was constrained by $0.1 < C_{e2,eff} < 2.4$ (Sloan et al (1986)).

Fig. 8 shows the "smooth−surface" experimental data, as described above, in comparison with the results of the basic and modified Launder and Sharma models. The computational model was otherwise unaltered from that described by Gan et al (1996) and the same 91x91 grid was used. The new data were taken at an axial location z/s=0.8, however little variation is expected between this and the location z/s=0.5, as used in Fig. 7 (Gan et al (1996)). The basic model computations are in slightly better agreement with the new experimental results in the region $x^{-2} > 2$, corresponding to the outer part of the cavity. However the smooth surface data are again well represented by a Rankine−vortex curve, and the basic turbulence model still fails to capture this essential feature of the flow.

The results of the modified turbulence model given in Fig. 8 show that the introduction of the Richardson−number term leads to generally better agreement with the data for these three cases. For λ_T=0, the predicted tangential velocity distribution shows improved agreement with the

Rankine−vortex character of the measurements, and the revised computations are closer to the measured velocities. This is the expected effect of the Richardson−number correction, since it acts to adjust the relative importance of free and forced vortex regions in flows where these regimes are combined.

For λ_T=−0.034, Fig. 8 shows less good agreement between the computations and data, although there is still some improvement using the modified model. This suggests that the Richardson−number correction may not be a universal cure for deficiencies in predicting these flows. As described above, the case λ_T=−0.104 exhibits behaviour (both measured and computed) close to that of a free vortex. In this case there is no significant effect of the correction, since, for free−vortex flow, $\partial/\partial r(rV_\phi)=0$ and $Ri_{gs}=0$ from equation (5). For this reason also, the modification used here would not be expected to affect the cover−plate calculations described above, and negligible effects have been observed in calculations of other rotor−stator, rotating cavity or contra−rotating−disc flows with either inflow or outflow.

CONCLUSIONS

An axisymmetric finite−volume multigrid elliptic solver, incorporating the Launder and Sharma low Reynolds−number k−ε turbulence model, has been used to compute the flow and heat transfer in rotating−disc systems, which are idealised models of the cooling−air systems found in gas−turbine engines. For rotor−stator systems, rotating cavities and contra−rotating discs, where results have been reported extensively in the literature, the agreement between computations and experimental measurements is mainly good. In this paper, the solver is used to compute the flow in pre−swirl rotor−stator systems and in a rotating cavity with a stationary peripheral casing. The computations for the pre−swirl system are encouraging but those for the cavity reveal a fundamental deficiency in the turbulence model.

For the rotating cavity, the stationary casing creates recirculating flow in which the core forms a Rankine (or combined free and forced) vortex. The basic Launder and Sharma model fails to produce the Rankine−vortex flow shown by the experimental data. A Richardson−number correction to the ε equation produces improved computations for the cavity, and is not expected to affect adversely the computations for other rotating−disc systems. However, the correction cannot be regarded as a universal cure for the deficiencies of the basic model, and the use of anisotropic, low Reynolds−number differential stress models may be required for more robust calculation of this type of flow.

Experimental and computational work on the pre−swirl and rotating cavity problems is continuing, and in particular the validation of computations of heat transfer will be reported in due course.

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