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Modelling internal air systems in gas turbine engines

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Abstract: Rotating disc systems can be used to model, experimentally and computationally, the flow and heat transfer that occur inside the internal cooling air systems of gas turbine engines. These rotating disc systems have been used successfully to simplify and understand some of the complex flows that occur in internal air systems, and designers have used this insight to improve the cooling effectiveness, thereby increasing the engine efficiency and reducing the emissions. In this review paper, three important cases are considered: hot gas ingress; the pre swirl system; and buoyancy induced flow.

Ingress, or ingestion, occurs when hot gas from the mainstream gas path is ingested into the wheel space between the turbine disc and its adjacent casing. Rim seals are fitted at the periphery of the system, and sealing flow is used to reduce or prevent ingress. However, too much sealing air reduces the engine efficiency, and too little can cause serious overheating, resulting in damage to the turbine rim and blade roots. Although the flow is three dimensional and unsteady, there are encouraging signs that simple 'orifice models' could be used to estimate the amount of ingress into the wheel space.

In a pre swirl system, the cooling air for the gas turbine blades is swirled by stationary nozzles, and the air is delivered to the blades via receiver holes in the rotating turbine disc. Swirling the air reduces its temperature relative to the rotating blades, and the designer needs to calculate the air temperature and pressure drop in the system. The designer also needs to calculate the effect of this swirling flow on the heat transfer from the turbine disc to the air, as this has a significant effect on the temperature distribution and stresses in the disc. Recent experimental and computational studies have given a better understanding of the flow and heat transfer in these systems.

Buoyancy induced flow occurs in the cavity between two co rotating compressor discs when the temperature of the discs is higher than that of the air in the cavity. Coriolis forces create cyclonic and anticyclonic circulation inside the cavity and, as such flows are three dimensional and unsteady, the heat transfer from the discs to the air is difficult either to compute or to measure. The flow also tends to be unstable and one flow structure can change quasi randomly to another, which makes it hard for designers of aero engines to calculate the transient temperature changes and thermal stresses in the discs during take off, cruise and landing conditions. Although recent CFD research has been successful in computing these flows, it will be many years before the designer can rely on computations unless they have been validated on reliable experimental data.

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Nomenclature

a	rotor inner radius
b	rotor outer radius
c_p	specific heat capacity at constant pressure
C_D	discharge coefficient
$C_{p, \max}$	nondimensional pressure difference ($= \Delta p_{\max} / 0.5 \rho W^2$)
C_w	nondimensional flow rate ($= \dot{m} / \rho b$)
$C_{w, \min}$	minimum value of C_w to prevent ingress

g	centripetal acceleration
G	gap ratio ($= s/b$)
G_c	seal clearance ratio ($= s_c/b$)
Gr	Grashof number ($= g L^3 \beta \Delta T / \nu^2$)
h	heat transfer coefficient
k	thermal conductivity of air
K	empirical constant
L	characteristic length
$\dot{m}$	mass flow rate
M	disc moment
Nu	Nusselt number ($= q_w L / k \Delta T$)

$p_{\max}$	nondimensional pressure parameter ($= 0.5 C_{p, \max} Re_w^2$)	Γ	ratio of speed of slower disc to that of faster disc
Pr	Prandtl number($= \mu c_p/k$)	ΔT	temperature difference
q_w	rotor wall heat flux	$\Delta p_{\max}$	maximum circumferential pressure difference in annulus
r	radius	η	cooling effectiveness ($= \dot{m}_o/(\dot{m}_o + \dot{m}_i)$)
r_p, r_b	radii of pre swirl nozzles and receiver holes	$\Theta_{b, ad}$	adiabatic effectiveness of pre swirl system
Ra	Rayleigh number ($= gL^3 \beta \Delta T/\alpha^2$)	λ_T	turbulent flow parameter ($= C_w/Re_\phi^{0.8}$)
Re_z, Re_w	axial Reynolds number ($= WL/\nu$)	μ	dynamic viscosity
Re_ϕ	rotational Reynolds number ($= \rho \Omega b^2/\mu$)	ν	kinematic viscosity
Ro	Rossby number ($= W/\Omega a$)	ρ	density
t	time	Ω	angular velocity of rotor
s	axial space between discs	Ω_c	angular speed of air in cavity
s_c	seal clearance	Subscripts	
T	static temperature	ad	adiabatic
T_o	total temperature in stationary frame	b	blade cooling flow
T_t	total temperature in rotating frame	i	ingress, or isentropic
V_r, V_ϕ	radial, tangential components of ve- locity in stationary frame	max	maximum
V^*	circumferential average radial veloc- ity	min	minimum
W	representative axial velocity	o	superposed outflow
x	nondimensional radius ($= r/b$)	p	pre swirl flow
z	axial distance from disc	s	stator
α	thermal diffusivity	w	rotor
β	swirl ratio ($= V_w/\Omega r$), or volume expansion coefficient	ϕ, r, z	circumferential, radial, axial direc- tion
Φ	nondimensional sealing parameter ($= C_w/G_c Re_\phi$)	∞	value in core at $z/s= 0.5$
γ	ratio of specific heats	1, 2	upstream, downstream locations in stream tube

Introduction

Fig. 1 shows a simplified diagram of the internal cooling-air system of a gas turbine engine. The cooling air for the first-stage turbine blades is extracted from the HP (high pressure) compressor of the engine, and the designer tries to minimise the amount of air extracted without adversely affecting the life of the blades and other critical components. It is worth noting that the mainstream gas temperatures in an engine can be over 1600 °C, which is above the melting point of super-alloys, and the temperature of the HP air can be over 700 °C. Up to 20% of the engine core flow is used for cooling and sealing purposes, and this can cause a 5% loss in the available energy in the fuel.

Rotating disc systems^[1,2], as shown in Fig. 2, can be used to model, experimentally and computationally, the flow and heat transfer that occur inside the internal air systems of gas turbine engines. For example, a rotor-stator system (Fig. 2(b)), incorporating a single disc rotating close to a stationary disc, can be used to represent an air-cooled turbine disc rotating close to a stationary casing. A rotating cavity can be used to represent two co-rotating turbine (Fig. 2(c)) or compressor (Fig. 2(d)) discs. Rotating-disc systems have been used successfully to simplify and understand some of the complex flows that occur in internal air systems, and designers have used this insight to improve the cooling effectiveness, thereby increasing the engine efficiency and re-

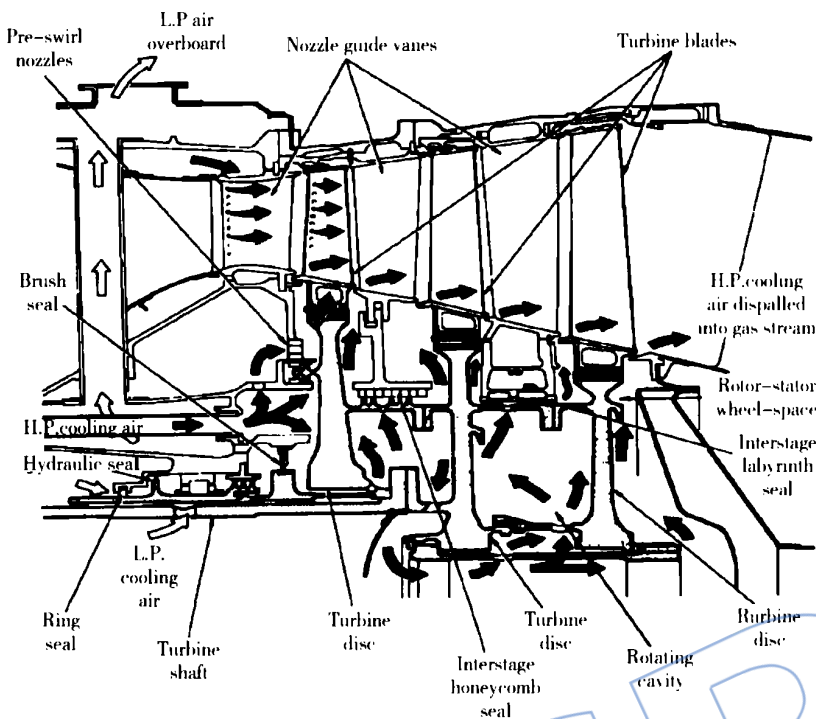


Fig. 1 Schematic diagram of a gas turbine cooling and sealing system (From Rolls Royce "The Jet Engine")

ducing the emissions. In this paper, three important cases are considered: hot-gas ingress in a rotor-stator system; pre-swirl systems; and buoyancy-induced flow in rotating cavities. These cases, which have been the subject of much recent interest in the European gas turbine community^[3], are probably the most important in the design of internal air systems.

Although numerous nondimensional variables are used in the study of flow and heat transfer in rotating-disc systems, the most important flow parameters are the rotational Reynolds number, Re_ϕ , the nondimensional flow rate, C_w , and the turbulent flow parameter, λ , which are defined as:

$$Re_\phi = \rho \Omega b^2 / \mu; C_w = \dot{m} / \mu b; \lambda = C_w / Re_\phi^{0.8} \quad (1)$$

λ , which depends on the other two parameters, is related to the flow rate entrained by a free disc (see Fig. 2(a)) where, from turbulent boundary-layer theory, the entrained flow rate is given by

$$C_w = 0.22 Re_\phi^{0.8} \quad (2)$$

Hence, $\lambda = 0.22$ corresponds to a flow rate equal to the free-disc entrainment rate. In an engine,

Re_ϕ is of the order of 10^7 and $\lambda \approx 0.1$ and 0.4 for sealing and blade cooling flows respectively.

For buoyancy-induced flow, the Grashof number, Gr , is an important parameter, where

$$Gr = g \beta \Delta T L^3 / \nu^2 \quad (3)$$

L is a characteristic length, ΔT is a characteristic temperature difference, and g is the centripetal acceleration, $\Omega^2 r$, where r is an appropriate radius. In the rotating cavities between compressor discs, Gr is of the order of 10^{14} .

1 The ingress problem

Fig. 2(b) shows a schematic diagram of ingress in a simple rotor-stator system with an axial clearance seal, which is used to model the turbine rim seal. Ingress, or ingestion, occurs when hot gas from the mainstream gas path is ingested into the wheel-space between the turbine disc and its adjacent casing. Rim seals are fitted at the periphery of the system, and sealing flow is used to reduce or prevent ingress. However, too much sealing air reduces the engine efficiency, and too little can cause serious overheating, resulting in damage to the turbine rim and blade roots. This is probably the most important of the 'rotating'

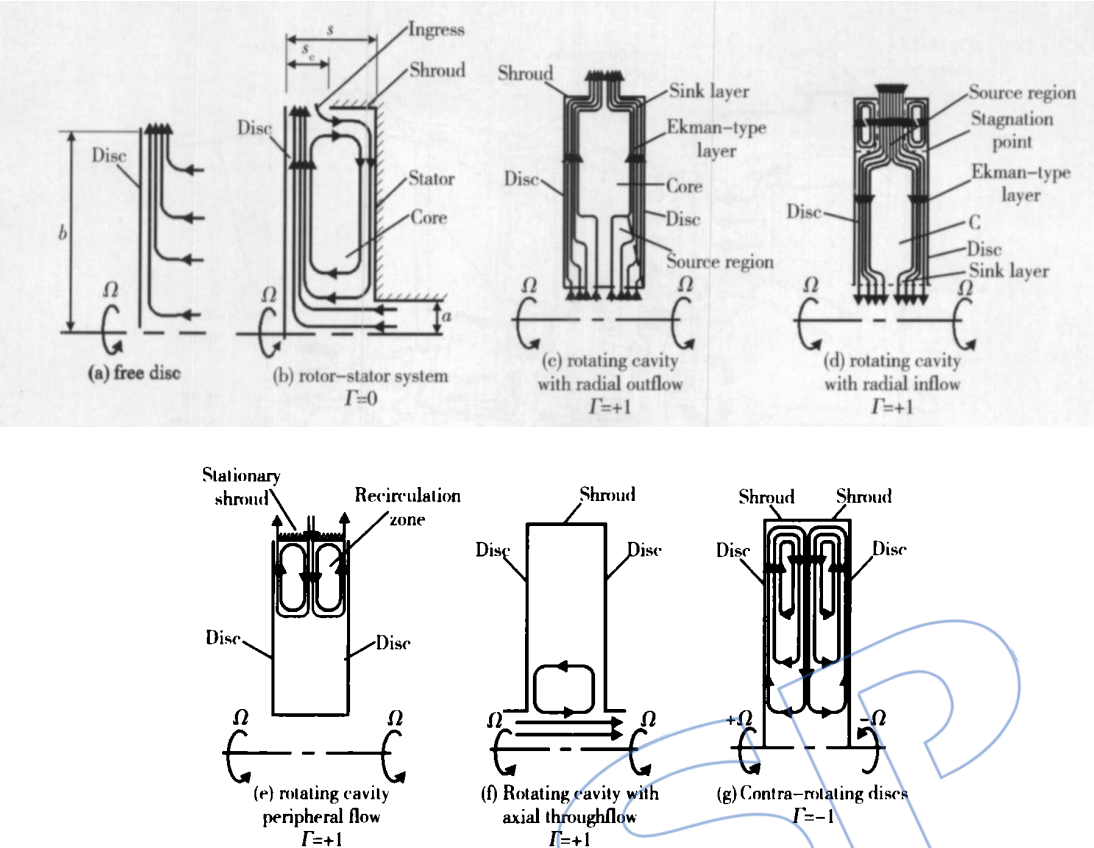


Fig. 2 Rotating-disc systems used to model gas turbine cooling systems

disc problems’ facing engine designers, and considerable international research effort has been devoted to finding acceptable design criteria.

Ingress occurs when the pressure in the external flow near the rim seal is higher than that in the wheel space; this appears to be a necessary but not sufficient condition for the external gas to flow through the seal clearance into the wheel space. The ingested gas then flows radially inward in the boundary layer on the stator, from where it moves axially across the rotating core of air in the wheel space to be entrained into the boundary layer on the turbine disc (or rotor) and mixes with the superposed flow of sealing air. The gas leaves the wheel space via the rim seal through which it entered the system. (Fig. 2 (b) shows the gas entering the system on the stator side of the wheel space; in an engine, the ‘bow waves’ created by the leading edge of the turbine blades can cause the gas to enter on the rotor-side.)

At the University of Sussex, Bayley and

Owen^[4] presented experimental results for a simple rotor-stator system with an axial clearance rim seal (see Fig. 2(b)) in which there was a superposed radial flow of air that discharged through the seal into the atmosphere. Owing to the sub-atmospheric pressure created by the rotating flow in the system, external (atmospheric) air could be drawn into the wheel space. Increasing the superposed flow rate increased the relative pressure inside the wheel space and consequently reduced the amount of ingested air. At sufficiently high superposed flow rates, where $C_w \geq C_{w, \min}$, ingress did not occur.

Bayley and Owen used a simple ‘ orifice model’-treating the seal clearance as an orifice and showed that the minimum nondimensional flow rate to prevent ingress, $C_{w, \min}$, was proportional to the rotational Reynolds number, and to the seal clearance ratio, G_c . Using their measured pressures for $G_c= 0.0033$ and 0.0067 , and $Re_\phi \leq 4 \times 10^6$, they proposed the following criterion:

$$C_{w, \min} = 0.61 G_c Re_\phi \tag{4}$$

This corresponds to $\Phi_{\min} = 0.61$, where Φ is a nondimensional sealing parameter defined in the Nomenclature.

Phadke and Owen^[5], who conducted experiments with a radial clearance seal at Sussex, also found a linear variation of $C_{w, \min}$ with $Re\phi$. Graber et al.^[6] made concentration measurements with CO_2 to determine the cooling effectiveness, η , for several seal geometries. Their rig had an external annulus, and they conducted tests for two different values of swirl ratio ($\beta = 1$ and 2) in the annulus. Although they found that their values of $C_{w, \min}$ were in reasonable agreement with the Sussex correlations, their results showed that Φw was not constant but decreased with increasing $Re\phi$.

Chew^[7] developed a theoretical model, which coupled the boundary layer flow in the wheel space with an orifice model for the seal. As stated in Section 1, boundary layer flow in rotating disc systems depends on λ , the turbulent flow parameter, and Chew used this to correlate his theoretical model with the available measurements. Although his model showed that $\Phi_{\min}$ was not a constant as suggested by Bayley and Owen but depended on λ , his results were in reasonable agreement with the Sussex data. As shown in Fig. 3, Chew's model was also in good agreement with the values of η measured by Graber et al. It should be noted that $\eta = 1$ corresponds to zero ingress, and $\eta = 0$ occurs when the superposed cooling flow is zero. (As stated above, a value of $\lambda = 0.22$ corresponds to the free disc entrainment rate.)

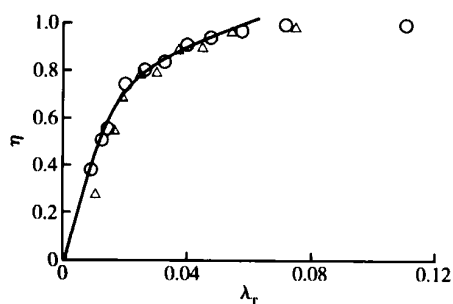


Fig. 3 Comparison between measured and calculated cooling effectiveness

Symbols: experimental data^[6]; line: theory^[7]

Abe et al.^[8], who used a rig with vanes in the annulus upstream of the rim seal, were the first to show that ingress could be dominated by the external flow in the annulus rather than by the rotation of the disc. Phadke and Owen^[9, 10, 11] used flow visualisation, pressure and concentration measurements to determine $C_{w, \min}$ in a simple rotor-stator system with a number of different rim seal geometries, with and without an external flow of air. (It should be noted that there were no vanes or blades in the external annulus of their rig; the circumferential pressure asymmetries referred to below were created by blocking sections of the annulus with wire mesh.) They identified two regimes: rotationally-induced ingress where, with no external flow, $C_{w, \min}$ increased with increasing $Re\phi$; externally-induced ingress where, with non-axisymmetric external flow, $C_{w, \min}$ was independent of $Re\phi$ and increased with increasing Re_w (where Re_w is the axial flow Reynolds number in the external annulus). However, for the case of quasi-axisymmetric external flow (with little or no circumferential pressure variation in the annulus), they found that $C_{w, \min}$ could decrease with increasing external flow. Their measured variation of $C_{w, \min}$ with Re_w is shown in Fig. 4 for an axial clearance seal with $G_c = 0.01$. For $Re_w = 0$, where rotationally-induced ingress occurs, $C_{w, \min} \propto Re\phi$; for large values of Re_w , where externally-induced ingress dominates, $C_{w, \min} \propto Re_w$; for intermediate values of Re_w , a minimum value of $C_{w, \min}$ can be seen to occur.

Phadke and Owen correlated their results for externally-induced ingress, based on flow visualisation for a number of different seal geometries, by:

$$C_{w, \min} = 2\pi K G_c p_{\max}^{1/2} \quad (5)$$

where

$$p_{\max} = 0.5 C_{p, \max} Re_w^2 \quad (6)$$

$C_{p, \max}$ is a nondimensional pressure difference in the external annulus and K is an empirical constant; the data were correlated with $K = 0.6$, as shown in Fig. 5. It should be noted that, although $p_{\max}$ is the controlling parameter, for a

given external geometry or in an engine, $C_{p,max}$ is expected to depend only weakly on rotational speed and flow rate. Hence, $C_{w,min} \propto Re_w$, as Eqs. (5) and (6) show and as Fig. 4 illustrates. Later experiments conducted by Turner et al.^[12], Hamabe and Ishida^[13] and Bohn et al.^[14] also showed that the external pressure asymmetry, generated by vanes or blades in the annulus, plays a dominant role in creating ingress.

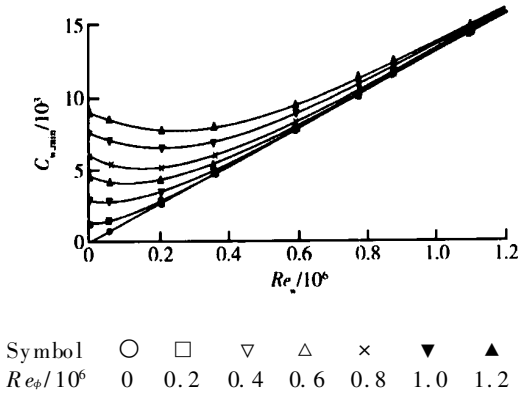


Fig. 4 Variation of $C_{w,min}$ with Re_w for axial clearance seal, $G_c = 0.01$ ^[11]

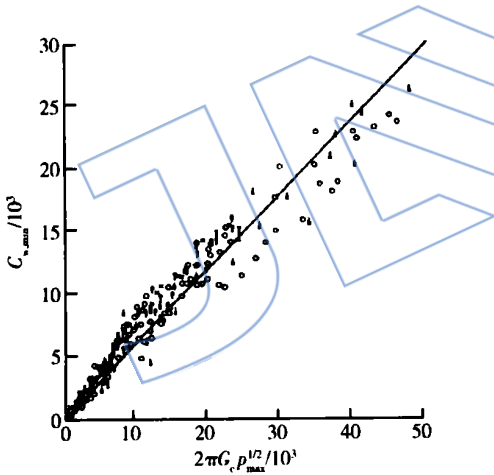


Fig. 5 Variation of $C_{w,min}$ with $p_{max}^{1/2}$
(Symbols represent experimental data;
line corresponds to Eq. (5))

Using concentration measurements in an experimental rig that modelled a turbine stage (with a single row of vanes and blades), Bohn et al.^[15] obtained an alternative correlation for externally-induced ingress in which the cooling effectiveness, η , was correlated in terms of C_w , G_c and $C_{p,max}$. The correlations of $C_{w,min}$ for the four seal geometries shown in Fig. 6 display the linear variation of $C_{w,min}$ with $C_{p,max}^{1/2}$ that was found by

Phadke and Owen, but the value of $K = 0.6$ suggested by the latter authors produces a conservative estimate for $C_{w,min}$. It can be seen from Fig. 6 that the double-radial seal (configuration 4) is the most effective of those tested.

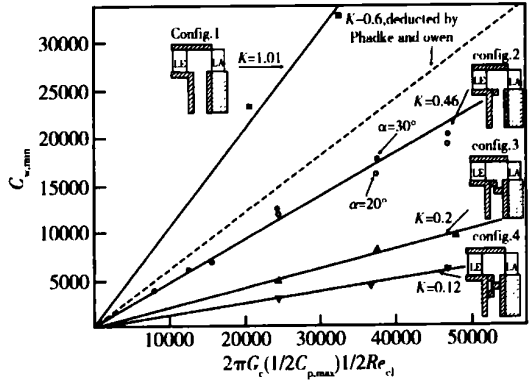


Fig. 6 Variation of $C_{w,min}$ with $C_{p,max}^{1/2}$
(Symbols represent experimental data;
lines correspond to correlations.)

Johnson et al.^[16] estimated the cooling effectiveness using an orifice model, in which the seal clearance was treated as an orifice with an unknown discharge coefficient, C_D . They compared their estimates with measurements in an experimental rig (see Bohn et al.^[17]) that incorporated a turbine stage with 16 vanes and 32 blades. (Bohn et al. noted in their paper that, because of the high pressures upstream of the rotating blades, the ingress had its maximum value in front of the blades, and the negative radial velocities at these positions could be up to twenty times greater than the positive velocities associated with the superposed flow of sealing air.) For the external circumferential pressure distribution in their model, Johnson et al. used the values obtained from 2D time-dependent CFD, which allowed the effects of the vane wakes and the blade bow waves to be taken into account. Fig. 7 shows the variation of the theoretical and experimental values of η with the velocity ratio, $V^* / \Omega b$, where V^* is the modelled circumferential average velocity through the seal clearance; it can be shown that $V^* / \Omega b \propto \Phi$. Their orifice model, with $C_D = 0.4$, provides a good fit to the experimental data; the choice of C_D will depend on the system geometry.

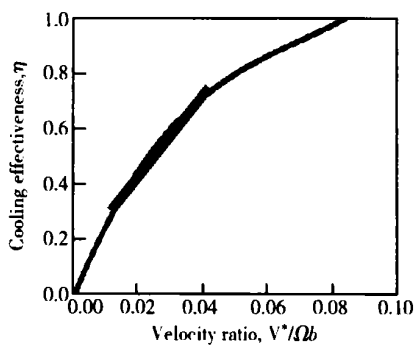


Fig. 7 Variation of cooling effectiveness with velocity ratio^[16] (Grey band represents range of experimental data; curve corresponds to orifice model.)

CFD has been used with some success for the ingress problem, but care must be exercised in computing these unsteady 3D flows. Although some measurements have shown that the nonaxisymmetric pressure distribution created in the annulus can be attenuated after the ingested flow has passed through the rim seal, the combined experimental and computational study of Cao et al.^[18] has shown that the swirling ingress can destabilise the rotating flow in the wheel-space. The instabilities can create a series of cells with a circumferential periodicity different from that of the vanes or blades. (These cells may be related to those found in buoyancy-induced flow, in which a circumferential distribution of pressure is needed to create the necessary Coriolis forces.) The authors suggest that a ‘segmental domain’ in CFD (where computations are conducted over an angular segment corresponding to, for example, the vane pitch) may be unable to capture these instabilities. However, their computations for a 90° segment showed largely good agreement with those for the full 360° domain.

The above description of the ingress problem is an attempt to simplify what is a very complex phenomenon. As well as the papers cited above, Refs. [19–32] contain further experimental and computational studies of the ingress problem. In particular, the paper by Johnson et al.^[21] discusses additional ingestion mechanisms to those referred to above.

2 Pre-swirl systems

A simplified diagram of the so-called direct

transfer pre-swirl system is shown in Fig. 8 where the blade-cooling air is supplied to the rotating high-pressure blades by stationary pre-swirl nozzles. The nozzles swirl the air, and this reduces the work done by the rotating turbine disc in accelerating the air to the disc speed and consequently reduces the total temperature of the air entering the receiver holes in the disc.

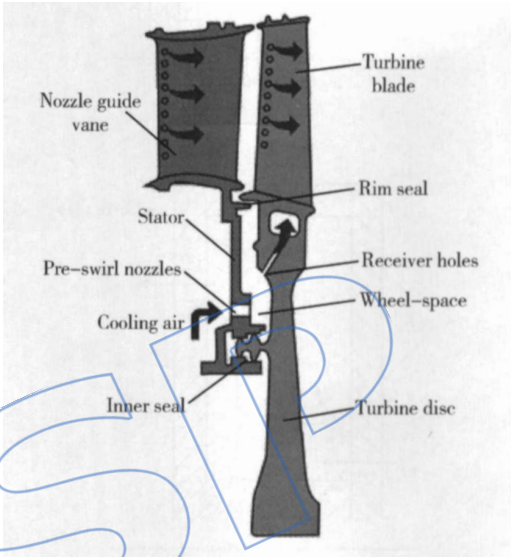


Fig. 8 Simplified diagram of direct transfer pre-swirl cooling system

Fig. 9 shows a schematic diagram of the two pre-swirl systems that are in common use: the direct transfer system, where the air from the pre-swirl nozzles (which are usually located at the same radius as, or at a smaller radius than, the receiver holes), and the cover-plate system, where the air flows radially outward in a cover plate attached to the turbine disc. As shown below, the cover-plate system has a higher effectiveness but the extra weight and stress associated with the cover plate can be a disadvantage. The designer is interested in calculating the pressure drop and cooling effectiveness of the pre-swirl system, and there is also a need to understand the heat transfer between the cooling air and the turbine disc.

Meierhofer and Franklin^[33], who were the first to measure the effect of pre-swirl on the temperature drop in a direct-transfer system, showed that swirling the air could significantly reduce the total temperature in the receiver holes

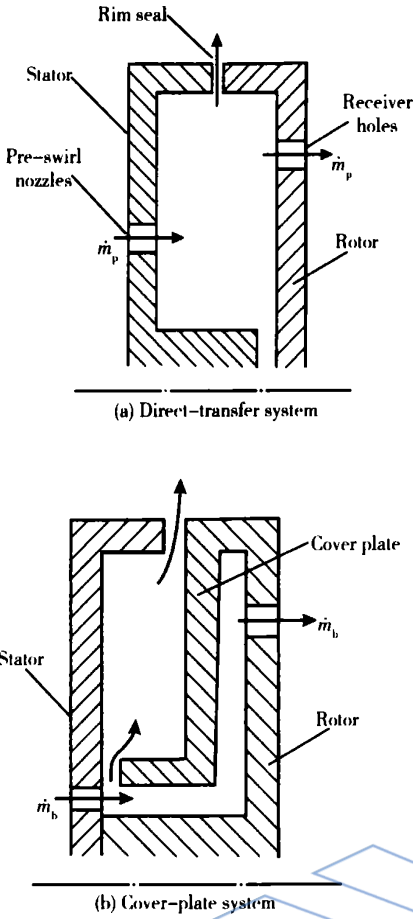


Fig. 9 Schematic diagram of direct transfer and cover plate systems

of a turbine disc. El-Oun and Owen^[34] developed a theoretical model for the so-called adiabatic effectiveness, $\Theta_{b,ad}$, based on the Reynolds analogy. The model, which was in good agreement with the temperatures measured on their rotating-disc rig, showed that $T_{t,b}$, the total temperature in the receiver holes, decreased monotonically as β_p , the pre-swirl ratio, increased even when β_p was significantly greater than unity.

Geis et al.^[35] made measurements of the adiabatic effectiveness, which showed that the measured values of $T_{t,b}$ were significantly higher than the values predicted from their ideal model. (It should be pointed out that their pre-swirl ratio was based on isentropic values rather than on measurements.) Chew et al.^[36] made numerical simulations of both the ‘Karlsruhe rig’, used by Geis et al., and a ‘Sussex pre-swirl rig’. The computations were in good agreement with the results of both rigs, and the low adiabatic effec-

tiveness of the Karlsruhe rig was attributed to the geometry of the pre-swirl chamber; in particular, the Karlsruhe rig had a much larger stator area, which reduced the effective swirl ratio and consequently reduced the effectiveness. Chew et al.^[36] and Farzaneh Gord et al.^[37] independently derived theoretical models for the adiabatic effectiveness of a direct transfer system.

Dittmann et al.^[38] measured the discharge coefficients for the pre-swirl nozzles and receiver holes in a direct transfer system, and Yan et al.^[39] also measured the discharge coefficients for the receiver holes. Heat transfer in a direct transfer rig was studied experimentally and computationally by Wilson et al.^[40] using fluxmeters to determine the local Nusselt numbers, and their axisymmetric CFD results gave reasonable predictions of the velocity and temperature in the core but under-predicted the measured Nusselt numbers.

Numerous experimenters have used thermochromic liquid crystal (TLC) to determine heat transfer coefficients on purpose-built test sections. A common technique is to solve Fourier’s transient conduction equation to calculate h for a semi-infinite solid exposed to a step change in air temperature. As it is virtually impossible to achieve a step change in the air temperature of pre-swirl rigs, Newton et al.^[41] developed the so-called ‘slow transient’ technique, and Lock et al.^[42,43] used this technique to measure the local Nusselt numbers, on the rotating disc of a direct transfer rig. Lewis et al.^[44] used CFD and simple analytical techniques to explain the flow physics of direct transfer systems, and their computed results for C_D , the discharge coefficient for the rotating receiver holes, and for $\Theta_{b,ad}$, the adiabatic effectiveness, are discussed below.

There has been less published on cover plate systems, and the interested reader is referred to Refs. [45–50]. It should be noted that the flow structure between the cover plate and the rotating disc is similar to that in a rotating cavity with a radial outflow of air (see Fig. 2(c)), but owing to the large flow rates involved the source region tends to

fill the cavity. Free vortex flow occurs outside the boundary layers on the rotating surfaces, and a theoretical description of these flows is given by Karabay et al.^[45].

As there is some confusion in the literature about C_D and $\Theta_{b,ad}$, it may be helpful to discuss both these parameters. The adiabatic effectiveness, $\Theta_{b,ad}$, is defined as:

$$\Theta_{b,ad} = \frac{c_p (T_{o,p} - T_{t,b})}{0.5 \Omega^2 r_b^2} \quad (7)$$

where $T_{o,p}$ is the total temperature (in the stationary frame of reference) of the air leaving the pre-swirl nozzles and $T_{t,b}$ is the total temperature (in the rotating frame) of the air in the receiver holes; for a given value of $T_{o,p}$, $T_{t,b}$, decreases linearly as $\Theta_{b,ad}$ increases. The effectiveness depends strongly on the inlet pre-swirl ratio, β_p , and Karabay et al.^[49] derived a theoretical value for $\Theta_{b,ad}$ using the First Law of Thermodynamics in which the work done on, or by, the air is proportional to the moment required to change the tangential velocity of the air from $\beta_p \Omega r_p$, at the pre-swirl nozzles, to Ωr_b in the receiver holes.

Their equation, which was derived for a cover-plate system in which there is no stator to reduce the swirl, is given by

$$\Theta_{b,ad} = 2\beta_p \left(\frac{r_p}{r_b} \right)^2 - 1 \quad (8)$$

It should be noted that $\Theta_{b,ad} \propto \beta_p$: $\Theta_{b,ad}$ increases, and consequently $T_{t,b}$ decreases, linearly with β_p even when $\beta_p > 1$. Karabay et al. also used CFD to compute $\Theta_{b,ad}$ in a cover-plate system for $0 \leq \beta_p \leq 3$, and their computations were in good agreement with this equation.

Farzaneh-Gord et al.^[37] modified equation (8) for a direct-transfer system to account for the moment on the stator, M_s . Their model gives

$$\Theta_{b,ad} = 2\beta_p \left(\frac{r_p}{r_b} \right)^2 - 1 - \frac{M_s}{1/2 \dot{m}_p \Omega^2 r_b^2} \quad (9)$$

where the effect of M_s (a positive quantity) is to reduce $\Theta_{b,ad}$.

Fig. 10 shows a comparison between the computed values of $\Theta_{b,ad}$ for a direct-transfer system and the theoretical values for $r_p/r_b = 0.8$. The computations are in excellent agreement

with the theoretical values of $\Theta_{b,ad}$ based on equation (9); the values of M_s in this equation were obtained by computing the tangential shear stress over all of the stationary surfaces. The theoretical values of $\Theta_{b,ad}$ calculated using equation (8) for a cover-plate system are significantly higher than those for the direct-transfer system.

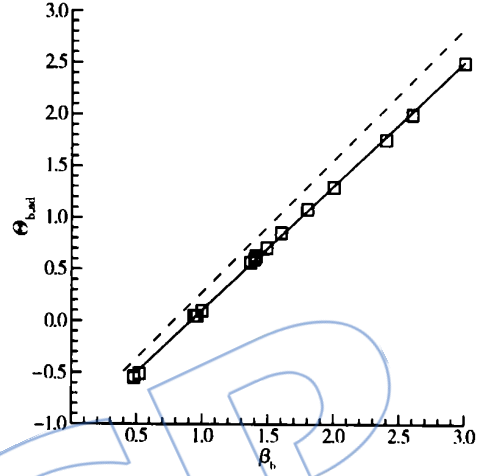


Fig. 10 Variation of $\Theta_{b,ad}$ with β_p for pre-swirl systems with $r_p/r_b = 0.8$, $0.8 \times 10^6 < Re_\phi < 1.2 \times 10^6$, $0.12 < \lambda_T < 0.38$. Dotted line: Eq. (8); solid line: Eq. (9); symbols: computations^[44].

It should be pointed out that both the computed and theoretical values of $\Theta_{b,ad}$ for the direct-transfer system depend, explicitly or implicitly, on the computed value of M_s . It is, therefore, uncertain that the theory will agree with experimental estimates of $\Theta_{b,ad}$, the measurement of which is nontrivial: the actual (not the isentropic) value of β_p must be known; the bulk-average relative velocity and total temperature inside the receiver holes must be measured accurately; and the rotating disc should be made from a thermal insulator to reduce the heat transfer to the cooling air. These three conditions are seldom, if ever, achieved in rotating-disc rigs.

The discharge coefficient for the receiver holes, C_D , is defined as:

$$C_D = \frac{\dot{m}_b}{\dot{m}_i} \quad (10)$$

where $\dot{m}_i$ is the isentropic mass flow rate and $\dot{m}_b$ is the mass flow rate of the blade-cooling air supplied through the receiver holes. The isen-

tropic flow rate can be calculated from the First Law of Thermodynamics for an adiabatic stream tube, taking into account the rate of work done by or on the air from stations 1 to 2 in the stream tube. The equation derived by Yan et al.^[39] is:

$$\frac{\dot{m}_i}{A_2} = \rho_{0,1} \left(\frac{p_2}{p_{0,1}} \right)^{\frac{1}{\gamma}} \left\{ \left(\frac{2\gamma}{\gamma-1} \right) \frac{p_{0,1}}{\rho_{0,1}} \left[1 - \left(\frac{p_2}{p_{0,1}} \right)^{\frac{\gamma-1}{\gamma}} \right] + 2\Omega(r_2 V_{\phi,2} - r_1 V_{\phi,1}) - V_{\phi,2}^2 \right\}^{1/2} \quad (11)$$

The first term inside the curly brackets is the standard result for compressible flow in a stationary nozzle; the second term represents the work resulting from the change of angular momentum of the air; the last term is due to the fact that the air in the receiver holes has an absolute tangential, as well as an axial, component of velocity. It should be noted that research workers who fail to define C_D correctly sometimes measure discharge coefficients that exceed unity, which is nonsensical.

Lewis et al. based $p_{0,1}$ and $V_{\phi,1}$ in Eq. (11) on their pitot-tube measurements at $r = r_b$ and $z/s = 0.5$, and they assumed that p_2 was the atmospheric pressure at outlet from the receiver holes and that $V_{\phi,2} = \Omega r_b$. Fig. 11 shows a comparison between the computed and measured variation of

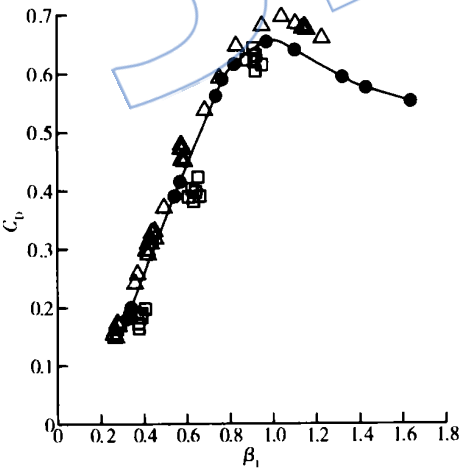


Fig. 11 Variation of discharge coefficient for the receiver holes, C_D , with swirl ratio, β_1 , for a direct transfer system with $r_p/r_b = 0.8$, $0.8 < Re_\phi/10^6 < 1.2$, $0.12 < \lambda_T < 0.38$. Open symbols are experimental values; solid symbols are computed values^[44].

C_D with β_1 , and both the computations and the measurements show that C_D reaches a maximum value at $\beta_1 = 1$. This shows that the minimum loss will occur when the core of fluid at the radius of the receiver holes is in synchronous rotation with the holes; for values of core rotation greater or less than this, the value of C_D will be lower than the maximum value. However, to achieve a value of $\beta_1 = 1$ it is necessary that β_p must be much greater than unity; in the computations, $\beta_1 = 1$ when $\beta_p = 1.8$. In an engine, the maximum value of C_D will depend on the design of the pre-swirl system.

Lewis et al.^[44] compared their computed Nusselt numbers with the measurements of Lock et al.^[42] on the rotating disc of their pre-swirl rig. The computed and measured values of Nu showed that for the smaller flow rates (that is, for the smaller values of λ and β_p) there is a viscous regime in which the heat transfer is controlled by the boundary layer flow on the rotating disc, except near the receiver holes where large values of Nu occur. For the larger flow rates, in the so-called inertial regime, the pre-swirl flow impinges on the disc creating large values of Nu at this radial location.

Fig. 12 shows a comparison between the measured and computed contours of Nu for the rotating disc; Fig. 12(a) corresponds to the viscous regime, and 12(b) and (c) to the inertial regime. (It should be noted that a steady 3D CFD code, in which the discrete pre-swirl nozzles were modelled as a circular slot, was used for the computations, and the measurements were made using TLC.) In Figs. 12(b) and (c), there is good correspondence between the computed and measured contours near the receiver holes, and in all three cases the Nu contours show quas-axisymmetry at the smaller radii. The small region of high heat transfer, measured around the ‘9 to 11 o’clock’ position near the holes, is captured by the computations. This region, which could result in high thermal stresses in the turbine disc, was attributed to the two routes by which the flow can enter the receiver holes: a ‘direct route’

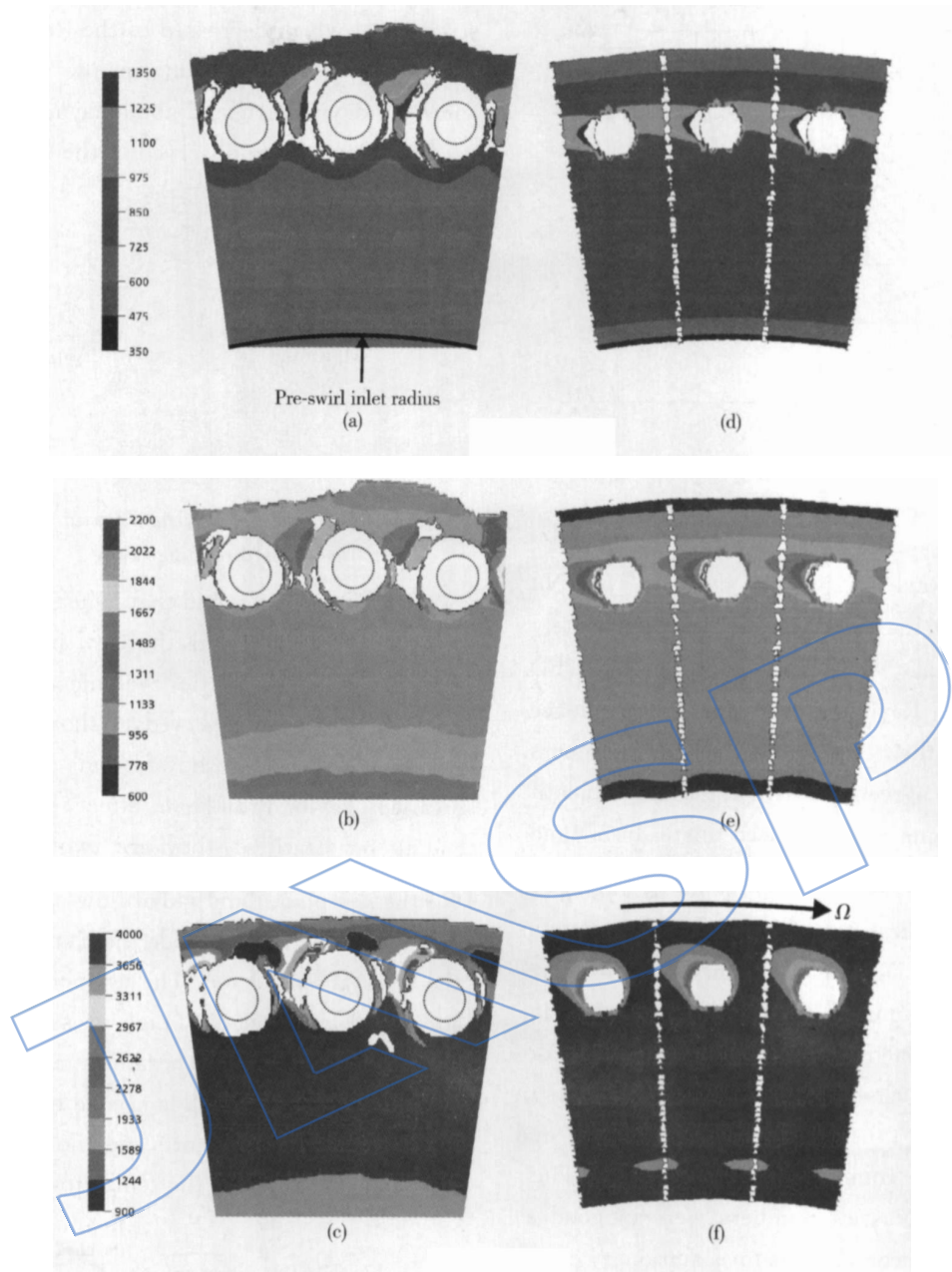


Fig. 12 Experimental^[42] (left) and computational^[44] (right) contours of Nu for direct transfer system with $r_p/r_b = 0.8$ and $Re_\phi = 1.2 \times 10^6$. (a) $\beta_p = 0.5$, $\lambda_T = 0.13$; (b) $\beta_p = 1.0$, $\lambda_T = 0.24$; (c) $\beta_p = 1.5$, $\lambda_T = 0.37$

from the preswirl nozzles to the holes, and an ‘indirect route’ from the rotating core.

3 Buoyancy-induced flow

Figure 13 shows a simplified diagram of a high pressure compressor rotor through the centre of which is an axial flow of air that is used, downstream, for turbine cooling. When the cooling air has a lower temperature than the rotating surfaces (as is the case for steady state engine operation), buoyancy-induced flow can occur in the cavity between the discs.

The main features of the flow in this so-called open system can be modelled by the simplified rotating cavities shown in Fig. 2(f). Most of the experimental work to date has been conducted with either a central inlet or with an annular inlet for the cooling air.

Bohn et al.^[54-55] made heat transfer measurements in a sealed rotating annulus (with $a/b = 0.35$ and 0.52) where the heat flow could be either axial (from a hot to a cold disc) or radial (from a hot outer cylinder to a cold inner one). For the radial case, they correlated their Nusselt

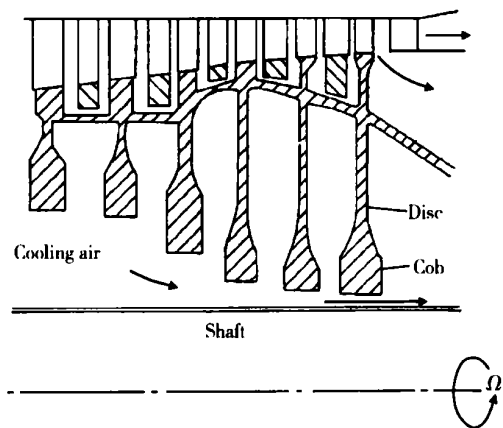


Fig. 13 Simplified diagram of high pressure compressor rotor with axial throughflow of cooling air

number, for Rayleigh numbers up to 10^{12} , by $Nu \propto Ra^c$, where the constant c was approximately 0.2. Sun et al.^[55] used an unsteady 3D CFD code, with no Reynolds averaging or turbulence model, and their computed Nusselt numbers were in good agreement with the correlations of Bohn et al. King et al.^[56] used unsteady 2D and 3D CFD codes, with no Reynolds averaging, to compute the heat transfer in a sealed annulus (with $a/b = 0.5$) with a radial heat flow for $Ra \leq 10^9$. Their computed 2D streamlines (see Fig. 14) showed that Rayleigh-Bénard convection occurred, with cyclonic and anti-cyclonic vortices in the $r-\phi$ plane; the vortices could vacillate and their number could change with time. Their time-average Nusselt numbers were in good agreement with correlations for a stationary cavity but were significantly higher than the correlations of Bohn et al.; the reason for this overestimate of Nu was not understood.

The Sussex group^[57-63] made measurements and computations for an open cavity (with $a/b \approx 0.1$) with an axial throughflow of cooling air. There was no central shaft or cylinder, and the outer cylindrical surface and/or the discs could be heated. In particular, Farthing et al.^[57] carried out flow visualization and LDA (laser Doppler anemometry) studies in both isothermal and heated cavities. In isothermal cavities^[63] with large gap ratios ($G > 0.4$), the axial throughflow generated a powerful toroidal vortex, the

strength of which decreased as the Rossby number Ro decreased. Depending on the values of G and Ro , axisymmetric and non-axisymmetric vortex breakdown were observed in the central jet.

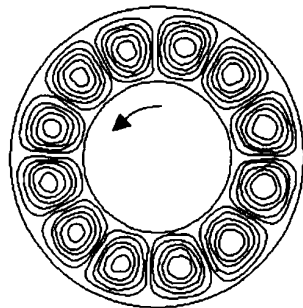


Fig. 14 Computed Rayleigh-Bénard vortices in a closed rotating cavity^[56].

Farthing et al. found that, when one or both of the discs were heated, the flow in the cavity became non-axisymmetric: cyclonic and anti-cyclonic vortices were observed, as shown schematically in Fig. 15. They showed that, owing to the so-called Taylor-Proudman effect, these flows tend to be stratified: they are two-dimensional (in the $r-\phi$ plane) and radial flow can only occur in the rotating core, outside the Ekman layers on the discs, if $\partial p / \partial \phi \neq 0$. The cyclonic (low pressure) and anti-cyclonic (high pressure) circulation shown in Fig. 15 generates a circumferential pressure gradient, which is needed to produce the radial flow inside the core, and the 'radial arm' convects cold air from the centre to the periphery

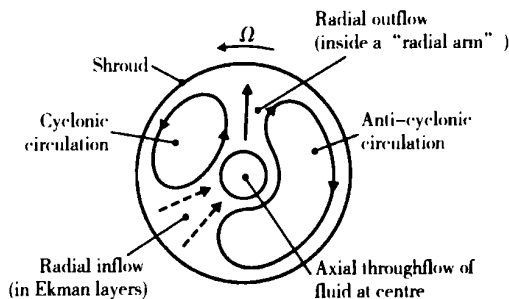


Fig. 15 Schematic of flow structure in the core of a heated rotating cavity with an axial throughflow of cooling air^[57].

of the cavity. In their experiments, the core of fluid precessed at an angular speed (Ω_c say) slightly less than that of the discs; the ratio of Ω_c / Ω decreased as the temperature difference between the discs and the cooling air increased.

Owen and Powell^[64] made measurements in an open cavity, where $a/b = 0.4$ and $G = 0.2$, in which one of the discs was heated. LDA and heat transfer measurements were made for $1.4 \times 10^3 < Re_z < 5 \times 10^4$ and $4 \times 10^5 < Re_\phi < 3.2 \times 10^6$. Spectral analysis of the LDA measurements revealed a multicell structure comprising one, two or three pairs of vortices. At the smaller values of Re_z , the measured Nusselt numbers were consistent with buoyancy-induced flow; at the larger values of Re_z , the effect of the axial throughflow became dominant.

Tian et al.^[65] computed the flow and heat transfer in a rotating cavity based on the geometry of Farthing et al. Their 3D, steady, turbulent computations showed that the flow structure comprised two parts: Rayleigh-Bénard convection at the larger radii, and forced convection in the central region. The computations suggested that there is a critical Rayleigh number above which the flow becomes unstable and time-dependent.

Johnson et al.^[66] investigated the stability characteristics of variable-density swirling flow in rotating cavities. Using a narrow-gap approximation for inviscid flow, they produced criteria for the necessary and sufficient conditions for stability. When the rotating surfaces are colder than the “cooling air” (such as at take-off in an aeroengine) the flow is stably stratified: the axial throughflow cannot penetrate very far into the cavity and the resulting convective heat transfer is relatively low. Conversely, when the rotating surfaces are hotter than the air (at cruise and landing), the flow is unstable, the axial flow can readily enter the cavity, and the heat transfer is increased as a consequence.

Three recent computational studies have thrown more light on the flow in open cavities. Sun et al.^[67] used large eddy simulation (LES) to compute the flow and heat transfer in a cavity with a central axial throughflow, and they compared their results with available experimental data and with an unsteady Reynolds-average Navier-Stokes (RANS) model. The LES model,

although computationally demanding, produced better agreement than the RANS model with the measured values of velocity and heat transfer. For both models, computations for a 120° segment of the cavity captured the large-scale flow structure and gave similar results to computations for the full 360° .

Bohn et al.^[68] used an unsteady 3D in-house CFD code to compute the flow structure that was measured in an experimental open rotating cavity. Flow visualisation in the experimental rig revealed that the flow structure changed periodically with time with the regular appearance of one, two or three pairs of vortices, and - as found in the experiments discussed above - the whole flow structure rotated with a core rotational speed, Ω_c , less than that of the cavity, Ω . The computed flow structure also showed the same periodic patterns, and the quantitative agreement with the measured periods was excellent. (It should be noted that the authors included a gravitational vector in the buoyancy term of their numerical model; without this term their computations failed to produce the periodic flow structure. In this author's opinion, the gravitational term only triggered the unstable flow, and any other non-axisymmetric term could have the same effect. As the magnitude of the centripetal acceleration of the cavity is of the order of 10^4 times that of the gravitational acceleration, g , it seems improbable that the g term could have a significant effect on the flow after the instability has been triggered.)

Owen et al.^[69] compared their computations with the experimental results of Owen and Powell. A commercial 3D unsteady CFD code (Fluent) with a $k-\epsilon$ turbulence model (similar to that used in the RANS model of Sun et al.) was used for the computations. The computations were compared with three experimental cases: for Exp. 2, $Re_\phi = 0.43 \times 10^6$, $Re_z = 0.303 \times 10^4$, and $Ro = 0.674$; for Exp. 5, $Re_\phi = 1.57 \times 10^6$, $Re_z = 0.164 \times 10^4$, and $Ro = 0.100$; for Exp. 6, $Re_\phi = 1.63 \times 10^6$, $Re_z = 0.295 \times 10^4$, and $Ro = 0.173$. Fig. 16 shows the computed temperature

contours at $z/s = 0.5$ for Exp. 2 and 5. For Exp. 2, the axial throughflow of cooling air creates a circular ring of cold fluid at the centre of the cavity. 'Radial arms', similar to the one shown in Fig. 15, transport some of the fluid from the centre to the periphery of the cavity, creating a thin layer of cold air adjacent to the unheated shroud. For Exp. 2, two distinct 'radial arms' are clearly visible, and this is in agreement with the two-cell structure that was observed experimentally. To the right of each radial arm, the circulation is cyclonic, which corresponds to a low-pressure region; in this region, hot air flows axially from the heated disc into the core. To the left of each radial arm, the circulation is anticyclonic, which corresponds to a high-pressure region where the core air flows axially towards the disc. The contours for Exp 5 in Fig. 16 are more complex: several large and small radial arms can be seen, and to the right of each one is a high-temperature region. For this case, the experimental observations were unable to determine the number of cells.

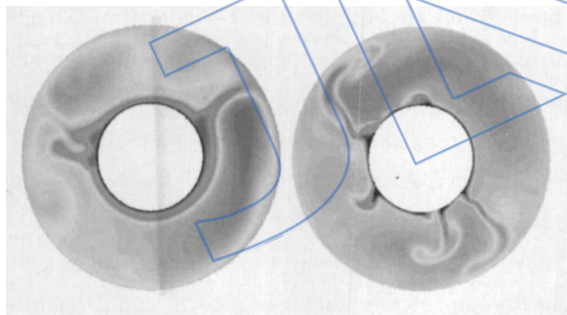


Fig. 16 Computed temperature contours at $z/s = 0.5$ for open rotating cavity^[9] Exp. 2 on left; Exp. 5 on right. (Rotation of cavity is clockwise.)

The computed values of tangential velocity in the core agreed reasonably well with the measurements for Exp 2 and 6 but not for Exp. 5, whereas the computed Nusselt numbers for the heated disc agreed reasonably well for Exp. 2 and 5 but not for Exp. 6. The reason for this poor agreement was not known, and the authors concluded that more work is required before the engine designer can depend solely on CFD codes to predict buoyancy-induced flow in rotating cavities.

4 Concluding comments

The above examples show how the study of simple rotating-disc systems can give useful insight into the more complex problems that occur in the internal cooling-air systems of gas turbine engines. By the use of analytical techniques, simple theoretical models, CFD and experiments, these complex flow structures can be understood both qualitatively and quantitatively. Although the three cases chosen are different in many respects, they share some common characteristics, particularly in the case of the two rotor-stator systems. In particular, the rotating cores, outside the boundary layers on the solid surfaces, show evidence of Taylor-Proudman flow where the axial variations of velocity are small and 2D stratified flow tends to occur in the $r-\phi$ plane.

When, owing to buoyancy effects or to ingress, radial flow occurs in the core, the resulting flow structure always becomes three-dimensional and unsteady. It is, therefore, dangerous to assume that apparently axisymmetric steady boundary conditions necessarily produce axisymmetric steady flows: using steady axisymmetric CFD can create a self-fulfilling prophecy in which the computed flow is steady and axisymmetric but the real flow is not. Although there have been many advances in the use of CFD, it will be many years before the engine designer can rely on computations unless they have been validated on reliable experimental data.

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