

OPERATING CHARACTERISTICS OF A HIGH RADIUS PRE-SWIRL COOLING SYSTEM

Z. Peng, P. New, C. A. Long and P. R. N. Childs

Rolls-Royce supported University Technology Centre in Aero-Thermal Systems, University of Sussex, UK

ABSTRACT

An experimental investigation into pre-swirl effectiveness and receiver hole discharge coefficient characteristics for a high radius injection pre-swirl cooling systems was carried out on a physically representative experimental rig with a 450 mm diameter rotor. The receiver holes and pre-swirl nozzle were located at a radius of 181 mm and 180 mm respectively. The experimental work was mainly conducted at 5000-12000 rpm, 4 bar absolute pressure and 1.132 kg/s air supply. The maximum air supply temperature was 190°C.

Pressure and temperature distributions in the pre-swirl system were examined with an emphasis on the velocity effectiveness of the pre-swirl system as a whole and on the discharge coefficients of the rotating 'receiver holes' in the rotor. The results showed that the velocity effectiveness increased with increasing swirl ratio resulting in reduced blade cooling flow temperature. Different seal flow configurations caused very different effectiveness at different speeds, but outflow through the inner and outer seals always gave the highest effectiveness compared other configurations. Increasing the seal flow rate reduced the effectiveness. For the coefficient of discharge, except for the low speed range, it increased with increase in swirl ratio for most speeds.

INTRODUCTION

Pre-swirl cooling systems are widely used in gas turbine engines in order to deliver cooling air to critical components enabling operation at high pressure ratios and turbine entry temperatures. An established method of introducing cooling air to the cooling passages in a turbine rotor is by accelerating it through inclined pre-swirl nozzles to achieve a high circumferential velocity component before the rotating feed holes. Thus the total temperature of the coolant airflow, relative to the rotor, is reduced with consequent benefits to the cooling system. In an ideal design, the pre-swirled coolant flow would penetrate the cavity, enter the blade feed holes in

the rotating turbine disc and then be pumped out through the blade cooling passages without ever coming into contact with the comparatively 'hot' disc cooling air. In practical engine geometries however, where disc cooling or sealing air is used to prevent the life degrading effects of mainstream gas ingestion there will be a complex adverse interaction as the pre-swirl air sweeps the hotter disc cooling air towards and into the rotating feed holes. The effectiveness of a pre-swirl system is described by the maximum achievable temperature decrease and the discharge coefficients of the pre-swirl nozzles and receiver holes. In order to design and build these systems so that they are effective and efficient, significant qualitative and quantitative physical insight is necessary [1,2].

The first to report on the performance of a pre-swirl cooling system was Meierhofer and Franklin [3]. From this work, it is recognized that there are significant potential benefits of cooling rotating components using pre-swirled air and the pre-swirl nozzle effectiveness is one of the main components of the effectiveness of the whole delivery system. Since then a number of researchers have attempted to improve the efficiency of pre-swirl cooling systems and to understand the heat transfer between the cooling air and the turbine disc by experimental investigations or theoretical model developments [4-8]. Recently, CFD has been applied to the analysis of the pre-swirl cooling system and more recently for supporting the design and manufacture of excellent cooling systems [1, 9, 10]. This makes it necessary to conduct some basic investigations into the essential operating and performance parameters for better understanding of the operating characteristics and for the validation of CFD models.

The objective of the work presented in this paper is to experimentally investigate pre-swirl effectiveness and receiver hole discharge coefficient characteristics for a high radius injection pre-swirl cooling system. The measurement of the pressure and temperature distribution in the pre-swirl system was conducted in a purpose-built test rig which had a 450 mm diameter rotor, 13325 rpm top speed and 1.13 kg/s air supply.

The effect of four different seal flow configurations on effectiveness were carried out: no seal flow – inner and outer seals balanced, inflow through the inner seal and out through the outer seal, inflow through both the inner and outer seals, and outflow through both the inner and outer seals. Three speeds were run for each configuration. Based on these tests, the effect of different seal flow rates on effectiveness and the effect of swirl ratio on the coefficient of discharge were established for different speeds.

NOMENCLATURE

A_3	area of one receiver hole
C_d	receiver holes discharge coefficient
C_1	bulk average velocity in the nozzle
C_p	Specific heat capacity
E	pre-swirl effectiveness
M	Mach number
$\dot{m}$	mass flow for receiver holes
R	universal gas constant
Sr	swirl ratio
T_{01}	total temperature in the nozzle reservoir
T_2	static temperature at pre-swirl nozzle exit
T_{2I}	ideal nozzle exit static temperature
T_{03}	total temperature in the rotor receiver holes
$T_{03,rel}$	relative total temperature in the rotor receive holes
U	tangential speed of the rotor
V_3	velocity at the exit of the receiver holes
V_p	pre-swirl velocity (at exit from pre-swirl nozzles)
V_{PA}	actual velocity of pre-swirl coolant
V_{PI}	ideal velocity of pre-swirl coolant
V_z	axial component of V_p
V_ϕ	tangential component of V_p
α	nozzle injection angle
γ	ratio of specific heats (C_p/C_v)
ρ_3	air density at the exit of the receiver holes
ω	rotor rotational speed

Subscripts

I	injection
Ideal	ideal condition
rel	relative to the rotor

PRE-SWIRL SYSTEM DESCRIPTION

A schematic of the pre-swirl system is shown in Figure 1. The pre-swirl supply crosses from the stator to the rotor to cool the rotating turbine blades. The usual definition of the swirl ratio is: the tangential component of the pre-swirl coolant velocity divided by the rotational velocity of the rotor at the radius of the pre-swirl nozzle. The tangential component referred to is $C_1 \cos \alpha$, where C_1 is the bulk average velocity in the nozzle and α is the nozzle injection angle. This is indicated on a vector diagram of ideal pre-swirl nozzle exit velocity in Figure 1.

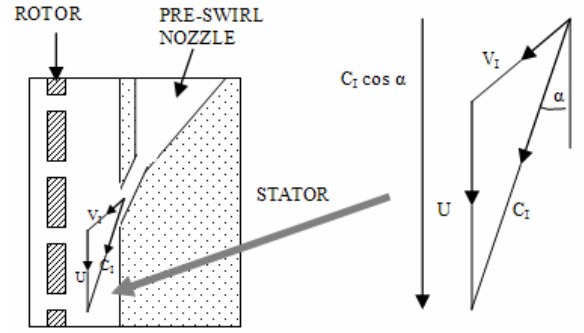


Figure 1 Vector diagram of coolant exit velocity from a pre-swirl nozzle

$$\text{Swirl ratio, } Sr = \frac{C_1 \cos \alpha}{U} \quad (1)$$

$$\text{Pre-swirl effectiveness, } E = \frac{V_{PA}}{V_{PI}} \quad (2)$$

where V_{PA} is the actual velocity of the pre-swirl coolant as it exits the pre-swirl nozzles calculated from the temperature recorded in the receiver holes in the rotor. V_{PI} is the ideal velocity of the pre-swirl coolant as it exits the pre-swirl nozzles calculated from the pressure ratio across the nozzle.

$$V_{PA} = \frac{2c_p(T_{01} - T_{03,rel}) + U^2}{2U \cos \alpha} \quad (3)$$

$$V_{PI} = M \sqrt{\gamma R T_{2I}} \quad (4)$$

The locations of subscripts used above can be determined from Figure 2.

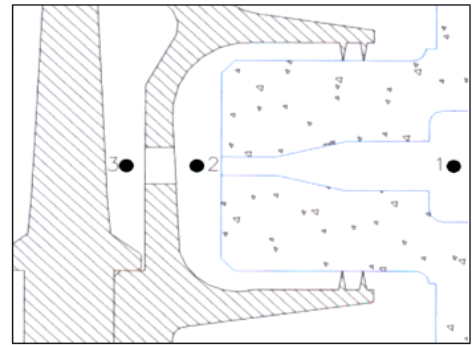


Figure 2 Physical locations of subscripts

For the effectiveness results of a pre-swirl cooling system, Karabay et al. [11] used an ideal adiabatic efficiency and cited a contamination factor ϕ for calculating ideal velocity effectiveness.

$$\text{Contamination factor } \phi = \frac{m_{is}}{m_p} \quad (5)$$

where m_{is} is the inner seal flow and m_p is pre-swirl mass flow.

The ideal velocity effectiveness is

$$E' = (1 - \phi) \quad (6)$$

For the calculation of the receiver hole discharge coefficient, C_d , using the NASA method [12], we have:

$$C_d = \frac{\dot{m}}{\dot{m}_{ideal}} = \frac{\dot{m}}{\rho_{3i} V_{3i} A_3} \quad (7)$$

Where $\dot{m}$ is the measured mass flow (total divided by the number of receiver holes). Subscript 2 and 3 refer to the physical locations indicated in Figure 2.

$$\rho_{3i} = \frac{P_3}{RT_{02}} \left(\frac{P_{02}}{P_3} \right)^{\frac{\gamma-1}{\gamma}} \quad (8)$$

$$V_{3i} = \sqrt{\frac{2\gamma RT_{02}}{(\gamma-1)} \left(1 - \frac{P_3}{P_{02}} \right)^{\frac{\gamma-1}{\gamma}}} \quad (9)$$

$$\text{Where } P_{02} = P_{01} \quad (10)$$

EXPERIMENT DESCRIPTION

TEST RIG

The test rig which was used for the investigation is shown in Figure 3. It mainly consists of a 90 kW, 3000 rpm Bull DC industrial motor with a Gemdrive 90 control system, a 5:1 step up gear box and the pre-swirl disc and chamber housing. Relevant specifications of the test rig are listed in Table 1.

The pre-swirl chamber is annular in shape with stationary pre-swirl nozzles on one side and the disc with the rotating receiver holes on the other. The receiver holes and pre-swirl nozzles are located at a radius of 181 mm and 180 mm respectively, as shown in Figure 4.

The nozzles are fed with cooling air from a large annular chamber called the pre-swirl reservoir. For all flow conditions, the coolant velocity in this chamber is below 10 m/s, consequently the dynamic contributions to temperature and pressure in this area were minimal. As shown in Figure 1, the schematic diagram shows a tangential view of the pre-swirl chamber for demonstrating the pre-swirl nozzle direction. In reality, they are angled at 20 degrees to the plane of rotation of

the disc (α shown in Figure 1). Various rotor stator gaps from 3 mm to 16.5 mm have been tested.

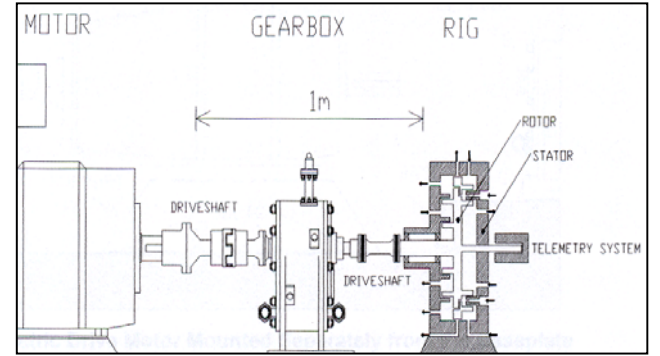


Figure 3 Test rig configuration

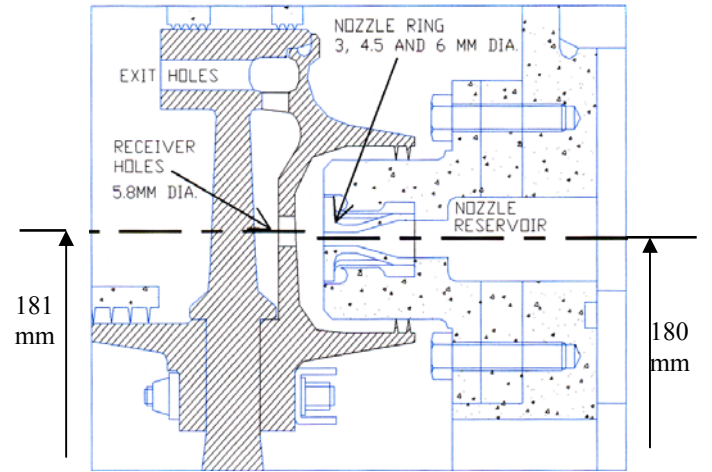


Figure 4 Pre-swirl chamber geometry

Table 1 Pre-swirl rig specifications

Max. Re_ϕ	1.2×10^7
Disc outer rim diameter	450 mm
Max. rotational rated speed	13,325 rpm
Pre-swirl nozzle	total 18 in number, 6 mm in diameter
Total pre-swirl flow area	$0.509 \times 10^{-3} \text{ m}^2$
Receiver holes	total 72 in number, 5.8 mm in diameter
Angle of injection	20° to the plane of rotation
Air supply max. mass flow	1.132 kg/s
Air supply max. temperature	200°C
Air supply min. temperature	ambient
Air supply max. pressure	4 bar (abs.)
Drive motor rating	90 kW
Drive motor max. speed	3000 rpm
Gearbox output:input	5:1
Gearbox rating	225 kW
Thermocouples	K type

INSTRUMENTATION

The arrangement of temperature and pressure sensors in the nozzle ring housing is shown in Figure 5. All thermocouples on the rig and its ancillaries were chromel/alumel K type, with an output of approximately $40 \mu\text{V}/^\circ\text{C}$. The pressure drillings on the inner and outer faces of the nozzle housing were to provide balance data for the pre-swirl chamber inner and outer seals. All static pressure tappings in the rig were 1 mm diameter with the entrance to the hole having been de-burred and the sharp edges broken.

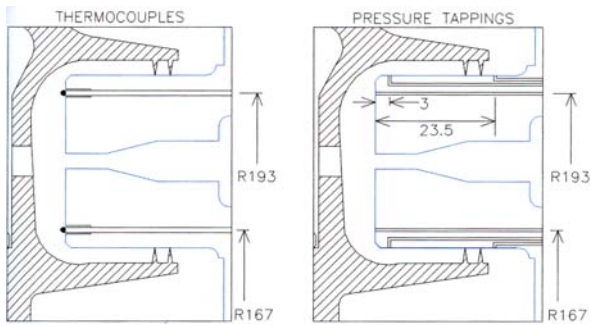


Figure 5 Nozzle ring housing instrumentation

On the cover plate, the instrumentation can be seen in Figure 6. When the cover plate and the rotor are viewed in an axial direction from the pre-swirl nozzles, the instrumentation can be seen to be positioned in three clusters 120° apart.

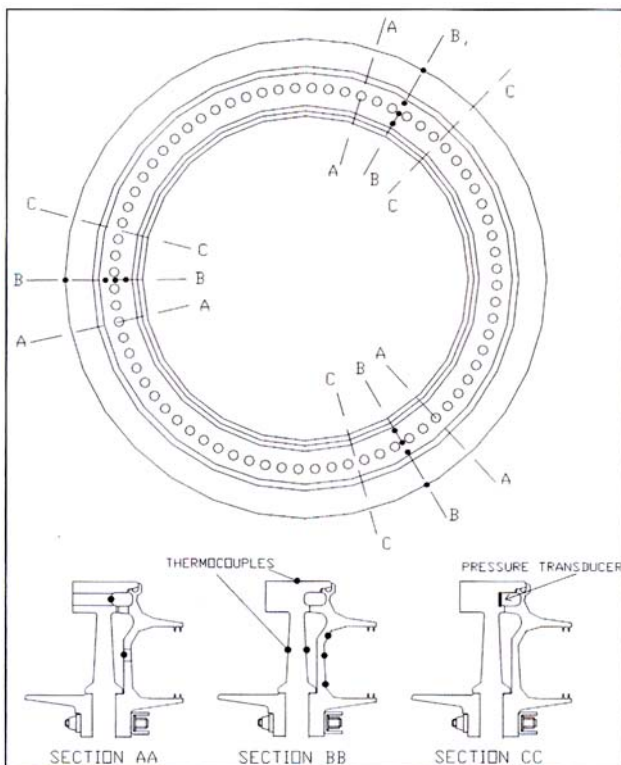


Figure 6 Cover plate instrumentation

In Figure 6, section AA indicates the position of the air thermocouples: three in the cover plate receiver holes and

three at a higher radius in the entrance to the airflow exit holes. The pre-swirl air passed through the receiver holes into an annular chamber, then out of the top of this chamber through 72 passages of semicircular cross section (of 12 mm diameter) in a radial direction. The pre-swirl air finally exited the rotor via 72 exit holes of 6.5 mm diameter. The position of the rotor air thermocouples is indicated in Figures 7 and 8.

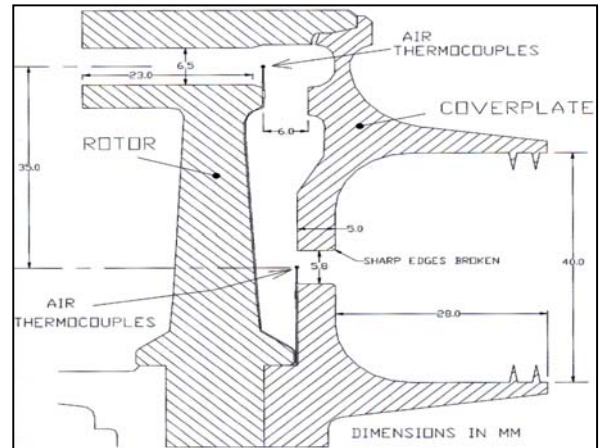


Figure 7 Section AA of the drawing shown in Figure 6, thermocouple installation in the cover plate and air exit holes

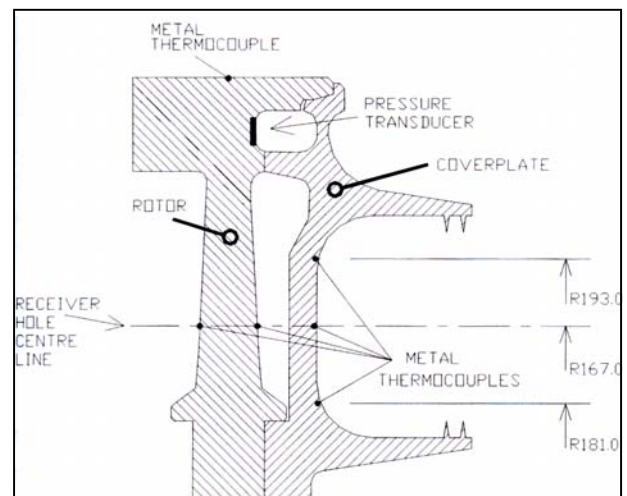


Figure 8 Section BB and CC of the drawing shown in Figure 6, other thermocouples and the rotating pressure transducer locations

The coverplate surface temperature thermocouples are located between the receiver holes on the front face. Additionally the rotor itself has thermocouples on its front face in the annular chamber behind the receiver holes, and on its rear face in the pressure balance chamber (shown in Figure 8). The locations of the rotor pressure transducers are also indicated in Figure 8.

MEASURABLE AND CALCULATED PARAMETERS

Measurement of the Pre-Swirl Nozzle Reservoir Temperature T_{01}

T_{01} is the total air temperature in the pre-swirl nozzle reservoir. This reservoir is a large annular chamber with a cross sectional area of 0.0823 m². The maximum possible air mass flow with the supply compressor was 0.3 kg/s at 3 bar absolute at 323 K, which resulted in an axial velocity of approximately 1.5 m/s in the nozzle reservoir. The air temperature T_{01} at this location was measured using three air thermocouples circumferentially spaced at 120°. The average of the three temperatures was used in the calculations; the three readings (after commissioning) were within 1°C of each other.

Nozzle Downstream Pressure P_2

The nozzle downstream pressure P_2 was measured by six static pressure taps, three at a lower radius than the pre-swirl nozzles and three at a higher radius (167 mm and 193 mm respectively). The average value of these six measurements is taken as the measured value of the downstream static pressure P_2 .

Measurement of Receiver Hole Coolant Temperature $T_{03,rel}$

The total temperature of the pre-swirl coolant $T_{03,rel}$ in the blade cooling holes was measured using a bare thermocouple bead placed at the rear of the blade cooling hole. The thermocouple location is shown schematically in Figure 7. The bead diameter was 0.5 mm and was at a radius of 181 mm from rotor centre-line axis.

Calculation of the Ideal Nozzle Exit Static Temperature T_{2I}

The temperature T_{2I} was calculated using the standard isentropic assumptions.

$$T_{2I} = T_{01} \left(\frac{P_2}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} \quad (11)$$

Where P_{01} is the total pressure measured in the pre-swirl nozzle reservoir.

Pre-Swirl Nozzle Exit Mach Number M

The nozzle exit Mach number was calculated using:

$$M^2 = \left[\frac{T_{02}}{T_{2I}} - 1 \right] \frac{2}{\gamma - 1} \quad (12)$$

Where we take $T_{02} = T_{01}$

Ideal Exit Velocity V_{PI}

$$V_{PI} = M \sqrt{\gamma R T_{2I}} \quad (13)$$

Rotational Velocity of the Receiver Holes U

The rotational velocity of the rotor at the radius of the blade cooling holes U , is calculated from the disc rotational speed and the radius of 0.181 m, being the centre line of the holes.

RESULTS AND DISCUSSIONS

Figure 9 shows the experimental conditions for the seal flow arrangements. If seal flows through the inner and outer seals are inflows, they are taken as positive. Otherwise outflows are accounted as negative flows.

In the investigation described in this paper, four different seal flow configurations were examined. They are:

- I. Configuration 1, inner and outer seals balanced – no seal flow, $m_{is} = m_{os} = 0$
- II. Configuration 2, inflow through inner seal and outflow through the outer seal, $m_{is} = 10\%$, $m_{os} = -10\%$
- III. Configuration 3, inflow through both inner and outer seals, $m_{is} = m_{os} = 10\%$
- IV. Config 4, outflow through both inner and outer seals, $m_{is} = m_{os} = -10\%$

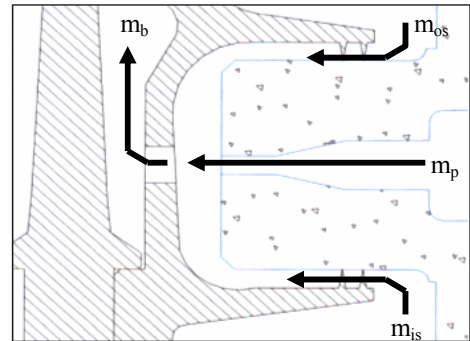


Figure 9 Flows through inner and outer seals

EFFECT OF DIFFERENT SEAL FLOW CONFIGURATION ON EFFECTIVENESS

When the tests for exploring the effects of seal flow configuration on effectiveness were conducted, each rotational speed was held constant while the swirl ratio, Sr , was altered to achieve five different values at each speed. The process was repeated for 5000 rpm, 8500 rpm and 12000 rpm.

Figure 10, shows the results for the seal flow configuration II listed above, inflow through inner seal and outflow through the outer seal, $m_{is} = 10\%$, $m_{os} = -10\%$. It is

thought that this kind of flow distribution is typical of that seen in a real engine.

In Figure 10, the trend of the effect of Sr on effectiveness for different speeds is presented. If the speed is constant, the effectiveness increases monotonically with increase in the swirl ratio. Also, for the same swirl ratio, a higher speed produced a lower effectiveness.

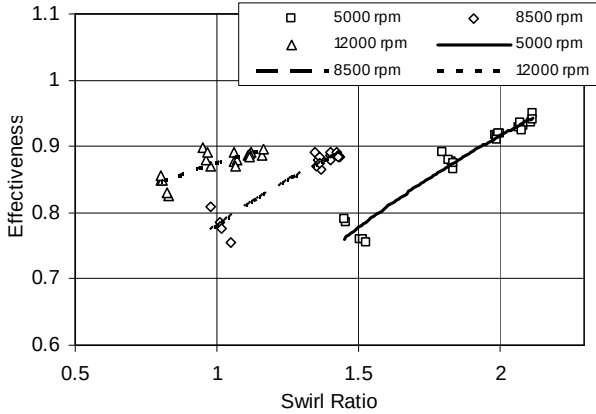


Figure 10 Effect of swirl ratio on effectiveness with inflow through the inner seal ($m_{is} = 10\%$) and outflow through the outer seals ($m_{os} = -10\%$)

In Figure 11, results for four different seal flow configurations for a constant speed of 5000rpm were plotted together. It can be seen that the effectiveness keeps increasing from seal flow condition I to II, to III and then to IV.

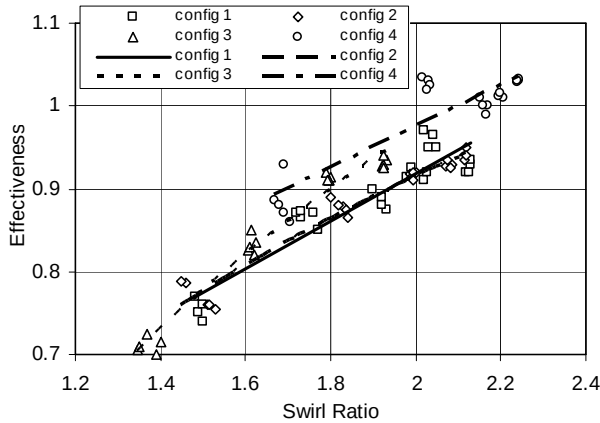


Figure 11 5000 rpm different configurations.

But if we continue to examine the effect of increased speed, such as 8500 rpm shown in Figure 12 and 12000 rpm shown in figure 13, it can be seen that the trend of the effect of different seal flow arrangement is different for different speeds.

The effectiveness under the seal flow configuration I keeps increasing from low speed to high speed compared to

configuration II and III. Or it can be said that effectiveness for configuration II and III decrease compared to the configuration I. Particularly the configuration III under low swirl ratio conditions has the greatest reduction with increase of speed. Only configuration IV kept the highest effectiveness for different speeds.

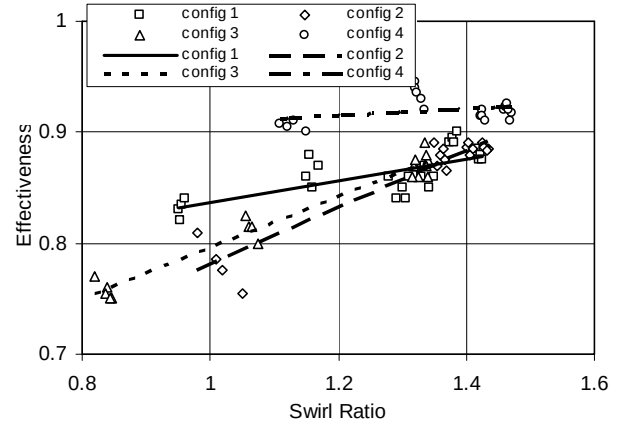


Figure 12 8500 rpm different configurations.

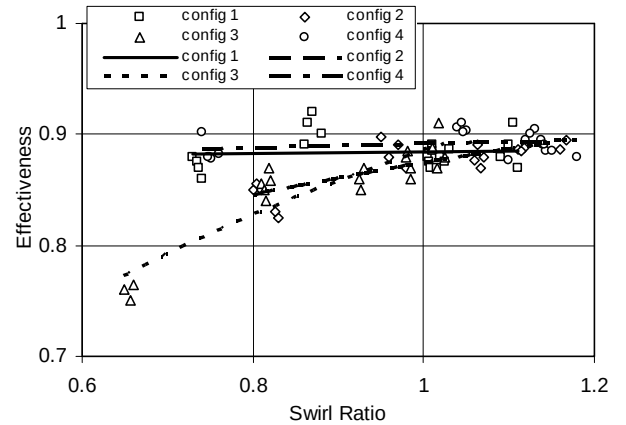


Figure 13 12000 rpm, different configurations.

If take the seal flow condition I as a baseline, in the seal flow condition II (Configuration 2) it must be assumed that mixing of the different flows in the pre-swirl chamber occurred, though the magnitudes of this pair of flow rates are equal. Disc cooling flow is used to purge the low radius area inside the inner seal. The only exit path for this flow is through the pre-swirl chamber. Some of the purge flow (which will be at a higher temperature than the pre-swirl) could be mixing with the pre-swirl coolant, reducing its effectiveness.

For configuration III, with inflows through both inner and outer seals both with 10% flow rates; effectiveness at high swirl ratios apparently appears unaffected compared to the results of configuration II. Effectiveness does however taper off rapidly to a lower value at low swirl ratios than indicated in the results of configuration I and II. Allowing air to enter the pre-swirl chamber reduces the pressure ratio across the

pre-swirl nozzle giving a lower coolant velocity and correspondingly reduces the swirl ratio.

For configuration IV of the seal flow arrangements, one reason for always having a high effectiveness compared to other configurations is that the pre-swirl coolant does not suffer from contamination by seal flows. The pressure ratio across the nozzles is significantly increased due to the coolant in the pre-swirl reservoir having a reduced restriction with respect to leaving the chamber. Hence for a given nozzle reservoir pressure, the swirl ratio is higher with outflow through both seals than with other seal flow configurations.

In Figure 14, all data with different seal flow configurations and different speeds are plotted together. Although effectiveness for different speeds display apparent differences for different speeds, the total trend of swirl ratio with speed are similar for all seal flow arrangements.

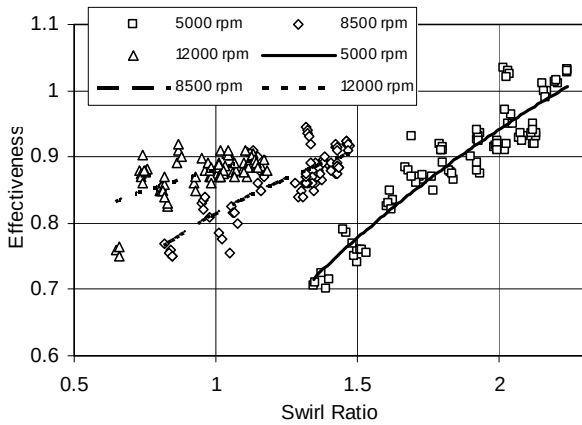


Figure 14 Effects of swirl ratio on effectiveness with all different seal flow

EFFECT OF DIFFERENT SEAL FLOW RATE ON EFFECTIVENESS

In the above section, the effect of the different seal flow configurations was presented. In this part, we discuss the influence of seal flow rate on effectiveness. The experiment was carried out with a 6.5 mm rotor stator gap, 5000-11000 rpm rotor speed, 1.1-3.3 bar (absolute) pre-swirl nozzle chamber pressure.

The seal flow arrangement was similar to configuration II but with 5% outflow rate through the outer seal. The flow through the inner seal was an inflow with the flow rate changed from 0 to +30%. As mentioned in the last section, this seal flow arrangement is typical of that found in a real engine.

Figure 15 shows the measured values of effectiveness obtained from tests at 5000 rpm. It can be seen that lower values of inner seal leakage result in a higher effectiveness

than with high values of inner seal leakage. This must be due to contamination of the pre-swirl coolant by the inner seal flow.

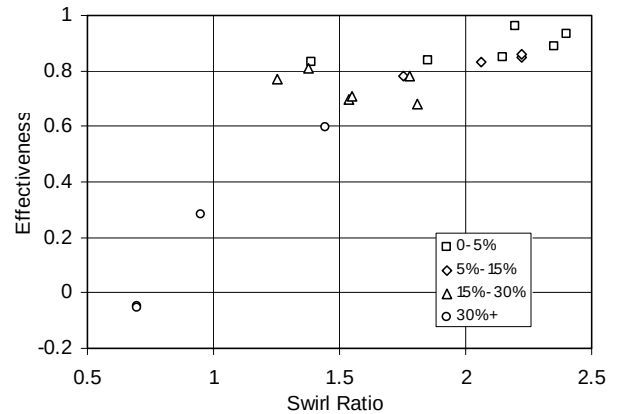


Figure 15 Variations of effectiveness with different inner seal flow rate (5000 rpm)

If all the data achieved for different speeds is presented in one figure - Figure 16, the trend is obvious. Higher inflow rate through the inner seal always results in a lower effectiveness, the trend becoming significant with increase of inflow rate.

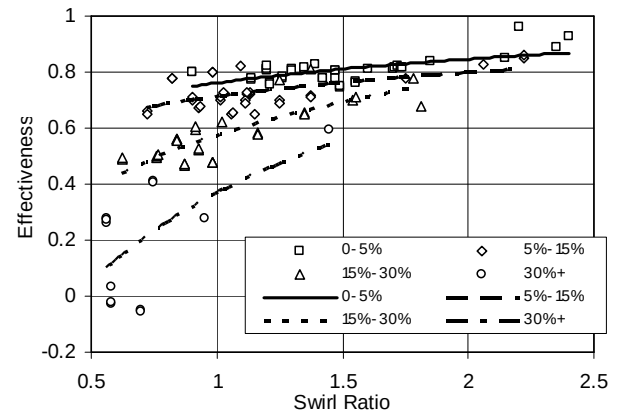


Figure 16 Variations of effectiveness with different inner seal flow rate under all four different speeds

Any seal flow entering the pre-swirl chamber will be rotating in the direction of rotor rotation due to its passing between a stationary and rotating surface. It was assumed that the swirl of this seal flow had a magnitude of 0.5ω (ω is the rotor rotational speed). For nearly all the tests carried out, this is below the swirl ratio. Hence an exchange of momentum must take place between the seal inflow and the pre-swirl coolant, resulting in a reduction in swirl of the main body of flow. The stator surface in the pre-swirl chamber will further reduce the magnitude of the swirl due to viscosity. To achieve a 'true' swirl ratio of unity, a pressure ratio across the nozzle to achieve a theoretical $Sr > 1$ is required. Furthermore, when a 'true' swirl ratio in excess of unity is achieved, effectiveness continues to increase with increasing swirl ratio, due to the

over swirled coolant doing work on the rotor and hence reducing the temperature.

DISCHARGE COEFFICIENT AND SWIRL RATIO RELATIONSHIP

The coefficient of discharge of the receiver holes in the rotor cover plate is of interest to the pre-swirl system designer. One of the design parameters of the pre-swirl system is the pressure ratio available to accelerate the coolant out of the pre-swirl nozzles. If the coefficient of discharge value of the receiver holes is low then a high percentage of the available pressure may be lost across the receiver holes.

The effect of the swirl ratio on the coefficients of discharge is shown in Figure 17. Four speeds were tested for investigating this relationship. Except for 5000 rpm, where the coefficient of discharge displays a nonlinear variation with swirl ratio, at other higher speeds the coefficient of discharge increases markedly with increase in swirl ratio.

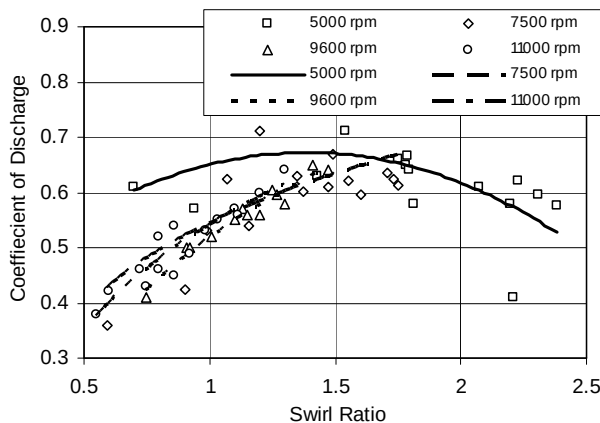


Figure 17 Cd vs. swirl ratio for the four different speeds

At high and low swirl ratios the relative velocity of the pre-swirl coolant to the rotor will be at a significant angle to the axis of the receiver holes, causing separation at the hole edges and thus a low Cd. This is backed up by Wittig et al [2] who state that lower Cd values are achieved with higher rates of rotation (axial flow was used in their rig, i.e. no swirl), this was considered to be due to a large re-circulation eddy being produced at the entrance of the hole due to the incidence angle of the velocity relative to the rotor. McGreeham and Schotsch [13] also stated that a higher relative velocity angle resulted in a lower Cd value. This argument indicates that a maximum Cd value should be achieved when the relative velocity of the coolant flow is perpendicular to the receiver holes. One might assume that this was the case at a swirl ratio of unity, but the value of core rotation just before the receiver hole is not necessarily the same as the swirl ratio. This is due to viscous drag on the stationary surface and contamination flow from the pre-swirl chamber inner and outer labyrinth seals (that itself may also have a different rotational velocity than the swirl ratio).

CONCLUSIONS

An experimental investigation into pre-swirl effectiveness and receiver hole discharge coefficient characteristics for radius injection systems has been carried out. The effects of rotational speed and swirl ratio for different seal flow configurations have been examined and the following conclusions obtained:

1. Effectiveness is greater at high swirl ratios since at these high ratios the coolant is doing more work on the rotor (due to viscous drag) resulting in a drop in its total temperature which gives a high effectiveness.
2. The effect of different seal flow configurations on the effectiveness display very different trends at different speeds, but configuration IV always produced the highest effectiveness compared to the other three tested configurations due to the pre-swirl coolant not suffering from contamination by seal inflows.
3. For seal flow condition II, the disc cooling flow is used to purge the low radius area inside the inner seal. The only exit path for this flow is through the pre-swirl chamber. Some of the purge flow, which has a higher temperature than the pre-swirl flow, could be mixing with the pre-swirl coolant, reducing its effectiveness.
4. In configuration III, the effectiveness decreases rapidly at low swirl ratio. This is because it allows air to enter the pre-swirl chamber reducing the pressure ratio across the pre-swirl nozzle giving a lower coolant velocity and correspondingly reducing the swirl ratio.
5. For the effect of the inflow flow rate through inner seal on the effectiveness, the test used an outflow through the outer seal. The results show that a higher inflow rate always results in a lower effectiveness at various speeds and the trend becomes significant with increase of inflow rate.
6. The effect of swirl ratio on the coefficient of discharge presents a substantially linear relationship for the higher speed conditions, while only for the lowest speed of 5000 rpm does the coefficient of discharge display a nonlinear variation with swirl ratio.

ACKNOWLEDGEMENTS

The authors would like to express their acknowledgements to Rolls-Royce plc for their supports.

REFERENCES

- [1] Lewis P, Wilson M, Lock G and Owen J M, Physical Interpretation of Flow and Heat Transfer in Pre-Swirl Systems. Proceedings of GT2006, GT2006-90132, ASME Turbo Expo 2006: Power for Land, Sea and Air, Barcelona, Spain, May 8-11, 2006.
- [2] Wittig S, Kim S, Jakoby R and Weibert I, Experimental and Numerical Study of Orifice Discharge Coefficients in

High-Speed Rotating Disks. *Transactions of the ASME*, Vol 118, p400-407, 1996.

[3] Meierhofer B, Franklin C J, An Investigation of a Preswirled Cooling Airflow to a Turbine Disc by Measuring the Air Temperature on the Rotating Channels. The Gas Turbine Conference and Products Show, ASME, March 9-12, 1981.

[4] El-Oun Z B and Owen J M, Preswirl Blade-Cooling Effectiveness in an Adiabatic Rotor-Stator System, *ASME J. Turbomachinery*, 111, pp. 522-529, 1989.

[5] Wilson M, Pilbrow R, Owen J M, Flow and Heat Transfer in a Pre-Swirl Rotor-Stator System, *ASME J. Turbomachinery*, 119, pp. 364-373, 1997.

[6] Geis T, Dittmann M and Dullenkopf K, Cooling Air Temperature Reduction in a Direct Transfer Preswirl System, *ASME Paper GT2003-38231*, 2003.

[7] Chew J W, Ciampoli F, Hills N J and Scanlon T, Pre-Swirled Cooling Air Delivery System Performance, *ASME Paper GT2005-68323*, 2005.

[8] Laurello V, Masada J, Araki M, Ishizaka K and Nakamura T, Correlation of Pre-Swirl Effectiveness with the Turbulent Flow Parameter and Application to the Mitsubishi MF111 Up-Grade. *Proceedings of GT2006*, GT2006-90182, ASME

Turbo Expo 2006: Power for Land, Sea and Air, Barcelona, Spain, May 8-11, 2006.

[9] Ciampoli F, Chew J W, Shahpar S and Willocq E, Automatic Optimization of Pre-Swirl Nozzle Design. *Proceedings of GT2006*, GT2006-90249, ASME Turbo Expo 2006: Power for Land, Sea and Air, Barcelona, Spain, May 8-11, 2006.

[10] Snowsill G D, Young C, The Application of CFD to Underpin the Design of Gas Turbine Pre-Swirl System. *Proceedings of GT2006*, GT2006-90443, ASME Turbo Expo 2006: Power for Land, Sea and Air, Barcelona, Spain, May 8-11, 2006.

[11] Karabay H, Chen J, Pilbrow R and Owen J M, Flow in a Cover-Plate Preswirl Rotor Stator System. The international Gas Turbine and Aero Engine Congress and Exhibition, Orlando, 1997

[12] Rohde J E, Hadley T, Richards T and Metzger G W, Discharge Coefficients for Thick Plate Orifices with Approach Flow Perpendicular and Inclined to the Orifice Axis, *NASA TN D-5467*, 1969.

[13] McGreeham W F and Schotsch M J, Flow Characteristics of Long Orifice with Rotation and Corner Radius, *ASME paper 87-GT-162*, The 32nd International Gas Turbine Conference and Exhibition, Anaheim, California, April 1988.