Experimental Investigation of Adiabatic Film Cooling Effectiveness and Heat Transfer Coefficients over a Gas Turbine Blade Leading Edge Configuration

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Abstract:
Increasing the rotor inlet temperature is one of the key technologies in raising gas turbine engine performance, for which the turbine blades need to be cooled. Film cooling is one of the efficient cooling techniques to cool the hot section components of a gas turbine engines. In film cooling, a gas which is cooler than the main stream is passed onto the external surface via small slots or rows of holes within the surface. In the present study, the experimental investigation was conducted for an adiabatic film effectiveness and heat transfer coefficients over a gas turbine blade leading edge model at a subsonic cascade tunnel facility of CSIR-National Aerospace Laboratories, Bangalore. This study aims at investigating the effect of blowing ratio on the adiabatic film cooling effectiveness and heat transfer coefficients experimentally for the 20 Degree hole inclination angles gas turbine blade leading edge model. The blade leading edge model was fabricated using the Rapid Proto Typing method using a very low thermal conductivity nylon based alloy material. This study aims at bringing the optimized blowing ratio values for the considered hole diameter of leading edge configuration. The comparative results showed that the blowing ratio beyond 2.0 does not have any improvement in the adiabatic film cooling effectiveness.

1. INTRODUCTION:

Generally, the Gas turbines are operated at higher temperatures in the range of 1200-1800 degree Celsius so the materials used for gas turbine blades should with stand these high temperatures without any melting and thermal stresses. The gas turbine blade leading edges are the vital parts in the turbines as they are directly hit by the hot gases, hence the optimized cooling of gas turbine blade leading edge surfaces is essential. Film cooling is used in many applications to reduce convective heat transfer to a surface. Multi row film cooling with span wise inclined film cooling holes, called showerhead, is extensively used for cooling the leading edge regions of cooled turbine vanes and blades. The cooling performance of the film cooling configuration is influenced by the combined effect of film cooling effectiveness, hole heat transfer and cold side heat transfer. A number of studies have considered leading edge film cooling performance using configurations of round holes and shaped holes [1, 2, 3, 4]. These laboratory investigations are concerned with evaluation of adiabatic film cooling effectiveness and occasionally the heat transfer coefficients. These two parameters are helpful

in characterizing the external heat loads on the blade surfaces. Film cooling is the introduction of a secondary fluid at one or more discrete locations along a surface exposed to a high temperature environment to protect that surface not only in the immediate region of injection but also in the downstream region [5].
In this study a 20 Deg. hole inclination angle showerhead cooling configuration was investigated for the adiabatic film cooling effectiveness and for the heat transfer coefficients. The blade leading edge model was fabricated using the Rapid Proto Typing method using a very low thermal conductivity nylon based alloy material with the outer diameter of 89 mm, inner diameter of 65 mm and with two rows of holes on either side of stagnation row at 30 and 60 deg. hole orientation angles respectively. The film cooling hole rows are arranged in a staggered manner to cover the more flow area on the blade surface and each row is consisting of 5 holes at a pitch of 18 mm with the hole diameter of 5.6 mm. Experiments were carried out at a nominal flow Reynolds number of 1, 00,000 based on the leading edge diameter, varying the blowing ratios of 1.0, 1.5, 2.0 and 2.5, at the density ratio of 1.30. The model was viewed through a Flir make Infrared Camera for the noncontact surface temperature measurements. The low thermal conductivity nylon and hard foam inserted material model was used to find the adiabatic film cooling effectiveness and heat transfer coefficients. The model was wound with the thin 0.15mm S.S sheet to have the smooth flow over the model surface during the film cooling experiments and to maintain the uniform heat flux during the heat transfer coefficient experiments. Thermocouples were fixed to the underside of S.S sheet to obtain local surface temperature for the infra red temperature correction.

2. EXPERIMENTAL SETUP:

The experiments were carried out in a 2-D blow down tunnel. The schematic of the test rig is shown in Fig.1 and Photographic view of experimental facility is shown in the Fig.2. The tunnel consists of flow control valve, settling chamber and test section. The settling chamber was provided with screens to reduce the turbulence level in the free stream. The size of the rectangular test section was 320mm x 230 mm. The test section consists of three parts namely inlet section, model section and outlet section. In the model section provision was made to attach a viewing window to view the leading edge model. The transparent window was so selected that it allows the infrared radiation in the spectral band of the IR camera. A pitot tube and a thermocouple were located upstream of the model leading edge at a distance 150 mm to measure the flow velocity and total temperature. Air was drawn from a central storage system. The velocity upstream of the model leading edge was measured in the test section by means of a pitot tube and the required flow velocity was set by controlling the inlet flow control valve. In the present experiments, the main stream air was at room temperature and the required coolant to mainstream density ratios were obtained by cooling the secondary cooling flow to -42°C. The secondary cooling flow to the model was supplied by a separate line provided with an orifice plate for measuring the mass flow. The metered air was passed through a heat exchanger where liquid nitrogen was used as coolant to cool the air to the required temperature. The temperature of the secondary cooling air was maintained at the required level by controlling the liquid nitrogen supply to the heat exchanger. The coolant temperature was measured inside the model in the leading edge cavity by means of thermocouples located at two spanwise locations. The pitot pressure and the orifice plate upstream pressure and differential pressures were measured by means of a differential pressure scanner. The turbulence intensity in the test section was measured at a location 150mm upstream of the model leading edge by means of hot wire anemometer and was found to be 8%.

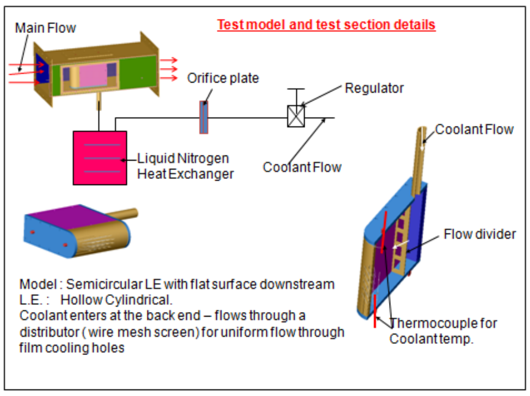

Fig. 1: Schematic of Test Rig and
Test Model

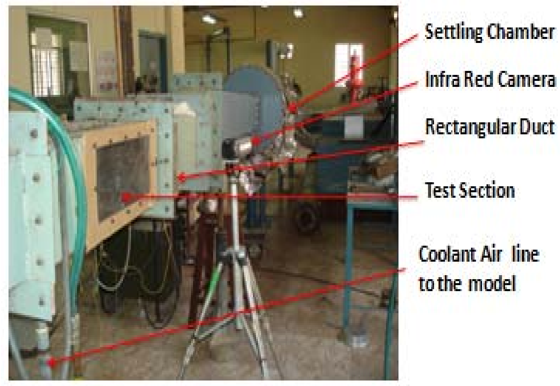

Fig. 2: Photographic View of
Experimental setup

3. DESCRIPTION OF THE TEST MODEL:

The test model has a semicircular leading edge which merges with a flat profile at $\pm$ 90° from the stagnation line. The RPT fabricated and prepared test model is shown in the Fig.3. The length of the flat region was 200mm. The secondary coolant flow was supplied at the end of the flat region as shown in the above figure. The span of the model was 210mm. Two thermocouples were fixed at 25% span wise distance from both ends to measure the coolant temperature in the leading edge cavity. A pressure tapping was provided to measure the coolant pressure in the leading edge cavity. The thermocouples were connected to a data logger for temperature measurement and the coolant pressure was measured by a pressure scanner. The thermal conductivity of the material selected was very low in the order less than 0.1 W /mK in order to have the adiabatic test model. The model was fabricated with the cooling holes arrangement and 5 rectangular slots for mounting thermocouples to measure the local surface temperature. These thermocouples were used as reference thermocouples to obtain the correction to the thermographic system output to take into account the contribution from background radiation and other factors. Subsequently, the grooves were filled with Araldite paste and hand finished to ensure no surface discontinuity. The thermocouple output was connected to a data logger for measurement. Subsequently the leading edge part was connected to the rectangular box assembly which contains the inlet for the secondary flow and the flow straightener. To ensure uniform emissivity for the viewed surface the model was painted using high emissivity black paint. To identify the location of the thermocouple in the thermogram, local markers were fixed to the model as shown in Fig.4.

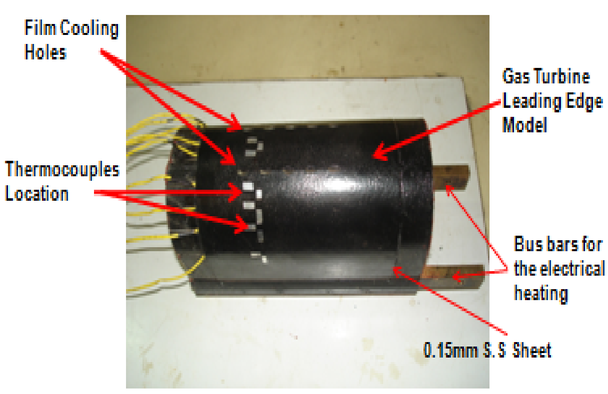

Fig.3: Fabricated Gas Turbine
Leading Edge Model

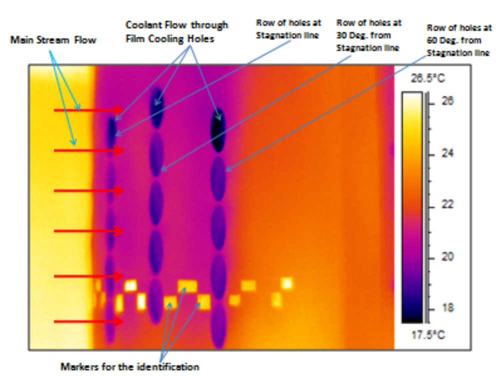

Fig. 4: Typical thermo graphic image
of the Leading Edge Model

4. EXPERIMENTAL PROCEDURE:

Initially the main stream valve was opened and the required mainstream flow velocity was set. The secondary flow was opened and liquid nitrogen was supplied to the heat exchanger. The coolant temperature was continuously monitored. After the secondary air attains the required temperature, the coolant flow was adjusted to ensure the required blowing ratio. The blowing ratio was estimated using the relation:

$$BR = (\rho_c V_c) / (\rho_\infty V_\infty). \qquad \text{(Eq.1)}$$

The main stream flux $(\rho_\infty V_\infty)$ was estimated from the measured mainstream velocity and the estimated mainstream density based on the static pressure and total temperature measured in the test section. The coolant mass flux $(\rho_c V_c)$ was estimated by dividing the measured mass flow by the total cooling hole area based on the inlet diameter of the film cooling hole. After establishing the required blowing ratio, the coolant temperatures and the model surface temperatures measured by the thermocouples were continuously monitored. The steady state condition was assumed to be attained when the temperature variation over a period of one minute was less than $\pm 2°C$. The model was maintained in steady state condition for a period of 5 minutes and the data were recorded. The thermocouples output were recorded over a period of 20 seconds in 10 scans. For the analysis, the average of the 10 scans value was used. A total of 5 thermo grams were recorded in sequence. For analysis the average of the 5 thermo grams was evaluated and used.

An uncertainty analysis performed indicates that the uncertainty in the calculated blowing ratio was $\pm$ 0.05 based on mainstream velocity uncertainty of +0.07 m/s and coolant velocity uncertainty of $\pm$0.5m/s.

4.1 Thermo graphic system calibration:

A long wavelength M/s Flir make Infrared thermo graphic system was used for the surface temperature measurement. In the present study, the test model was viewed through a window and the model surface has curvature. One of the important parameters which influence radiation measured by the system depends on the emissivity of the surface. To ensure that the surface has a uniform emissivity, the surface was painted with black paint. The in-situ calibration of camera has done with the similar test model and these calibration equations are used in the thermal image processing.

4.2 Adiabatic Film Cooling Effectiveness Measurements

In adiabatic film cooling experiments, the main stream air is allowed to flow over the test surface and the coolant flow is passed through the coolant chamber, which ejects through the film cooling holes. The required coolant flow is maintained to have the blowing ratios i.e. the coolant to main stream mass flux ratio of 1.0, 1.50, and 2.0. The coolant temperature is maintained at 231K to have the required density ratio of 1.30. The main stream air at ambient temperature coming from the centralized compressor facility is maintained at 15.76 mm of H_2O to have the Reynolds number of 100000 based on the leading edge diameter. The wall temperature is measured using non contact type infra red camera after the steady state main stream and coolant flows have been achieved. The film effectiveness with the wall temperature measured as adiabatic wall temperature, as the surface is unheated and well insulated. The local wall temperature is mixture temperature of the coolant and mainstream. Thus, the film effectiveness is found using the following relation.

$$\text{Film Cooling Effectiveness}, \eta = \frac{(T_m - T_w)}{(T_m - T_c)} \qquad \text{(Eq.2)}$$

4.3 Heat Transfer Coefficient Measurements

The heat transfer coefficient is measured with mainstream and coolant air at the same temperature, which is ambient temperature and the test surface is heated. The test plate is typically made of low conducting material to avoid heat losses from the gas path noncontact side of the model. The test surface is connected in series by brass bus bars to supply the required current at low voltage for uniform heating the test surface. The test surface when heated, serves as a constant heat flux surface for the heat transfer tests. The conduction loss data is estimated by the conduction loss experiments. The total input electrical power is calculated by measuring the voltage and current values. The net heat input to the test surface is calculated by subtracting the conduction losses from the input electrical power. Here, the input electrical power and the conduction losses are calculated per unit surface area of test plate. Hence, the calculated net heat flux, q is the net heat input per unit surface area of the test plate. The heated surface exposed to the main stream and the required coolant with the suitable blowing ratio. After achieving the steady state, the surface temperature data is captured by the infra red camera thermo graphic image. The wall temperature is extracted from the thermo graphic image data, the main stream temperature is measured by thermocouples. Heat transfer coefficient is calculated by using the following relation. Here, the heat transfer coefficient (h) is calculated from the net heat flux, q = Q/A i.e the net heat input per unit surface area of the plate.

$$\text{Net Heat Input, } Q = hA\Delta T = \text{Qgen} - \text{Qloss} \qquad \text{(Eq. 3)}$$

$$\text{Net Heat Flux, } q = \frac{Q}{A} = \frac{Q_{gen}}{A} - \frac{Q_{loss}}{A} \qquad \text{(Eq. 4)}$$

$$\text{Heat Transfer Coefficient, } h = \frac{q}{(T_w - T_m)} = \frac{q_{gen} - q_{loss}}{(T_w - T_m)} \qquad \text{(Eq. 5)}$$

To establish repeatability, experiments are conducted at least three times on the test model at each blowing ratio. The error analysis based on an absolute temperature error of ±1° C and the pressure variation with ± 0.025 %, yields an average relative error of approximately 6 % on heat transfer coefficient and 5 % on adiabatic film cooling effectiveness. All the instruments used in the experiments are calibrated with the standard calibrating primary and secondary sources.

The experimental test conditions for the adiabatic film cooling effectiveness of test model at the density ratio of 1.30 are shown in Table 1. The experimental test conditions for the heat transfer coefficients of test model are shown in Table 2.

Adiabatic Film Cooling Effectiveness										
D.R 1.30										
Main stream conditions (27 deg)						Coolant condition (-42 deg)				
BR	Temp	Density	velocity	Main Stream (Del P)		Temp	Density	Velocity	Chamber Pressure	
const	kelvin	Kg/m3	m/sec	N/m2	mm of H2O	Kelvin	Kg/m3	m/sec	N/m2	mm of H2O
0.50	300	1.051	17.15	154.5614	15.76	231	1.364	6.61	29.77	3.04
0.75	300	1.051	17.15	154.5614	15.76	231	1.364	9.91	66.99	6.83
1.00	300	1.051	17.15	154.5614	15.76	231	1.364	13.21	119.09	12.14
1.25	300	1.051	17.15	154.5614	15.76	231	1.364	16.52	186.08	18.98
1.50	300	1.051	17.15	154.5614	15.76	231	1.364	19.82	267.96	27.32
2.00	300	1.051	17.15	154.5614	15.76	231	1.364	26.43	476.38	48.58
Note: Density of Air at 27Deg C and at 920m seal level of Bangalore = 1.051 Kg/m^3 and at -42 Deg C is 1.364 Kg/m^3										

Table 1: Adiabatic Film Cooling effectiveness test conditions

Heat Transfer Coefficient										
Main Stream Conditions (27 deg)						Coolant conditions (27 deg)				
BR	Temp	Density	Velocity	M S (Del P)		Temp	Density	Velocity	Chamber Pre	
Const	Kelvin	Kg/m3	m/sec	N/m2	mm of H2O	Kelvin	Kg/m3	m/sec	N/m2	mm of H2O
0.50	300.00	1.051	17.15	154.6	15.76	300	1.051	8.58	38.64	3.94
0.75	300.00	1.051	17.15	154.6	15.76	300	1.051	12.86	86.94	8.87
1.00	300.00	1.051	17.15	154.6	15.76	300	1.051	17.15	154.56	15.76
1.25	300.00	1.051	17.15	154.6	15.76	300	1.051	21.44	241.50	24.63
1.50	300.00	1.051	17.15	154.6	15.76	300	1.051	25.73	347.76	35.46
2.00	300.00	1.051	17.15	154.6	15.76	300	1.051	34.30	618.25	63.04
Note: Density of Air at 27Deg C and at 920m seal level of Bangalore = 1.051 Kg/m^3										

Table 2: Heat Transfer Coefficient data test conditions.

5. RESULTS AND DISCUSSION:

The first series of experiments were carried out to obtain the coefficient of discharge (Cd) for the holes. The experiments were carried out with the film cooling hole exit pressure equal to atmospheric pressure. The coolant plenum pressure was varied controlling the mass flow. The mass flow through the model was measured using an orifice plate. By varying the mass flow with the Reynolds number, the C_d value of these holes was found as 0.64

The typical experimental film cooling effectiveness results along the stream wise direction at the blowing ratios of 1.0, 1.5, 2.0 and 2.5 are shown in the Fig.5. The peak values in the figures indicate the row of holes location where the coolant is exiting. The repeat runs also conducted for the consistency of readings and the compared run data has shown the consistency in the readings like as shown at the blowing ratio of 2.5 in the Fig.6. The experimental data showed the increase in adiabatic film cooling effectiveness increases with the increase in the blowing ratio however beyond the blowing ratio of 2.0 there is not a much improvement. At the blowing ratio of 1.50 the cooling effectiveness has shown higher but at the stagnation region it has shown the very low cooling effectiveness. Hence the blowing ratio of 2.0 can be considered as the optimised blowing ratio for this test model. The higher cooling effectiveness is observed at the blowing ratio of 2.0 for this 20 deg. hole angle model with 18mm pitch due to the higher flow attachment with the surface.

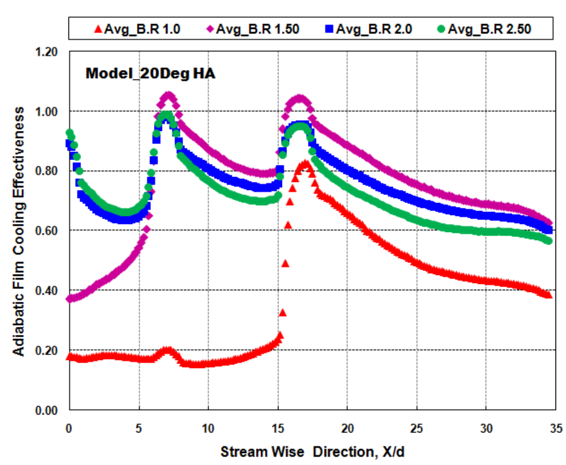

Fig.5: Experimentally evaluated adiabatic film cooling effectiveness in the flow direction.

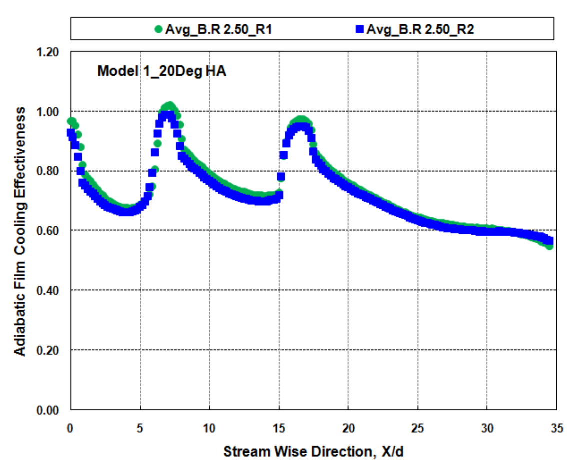

Fig.6: Experimentally evaluated adiabatic film cooling effectiveness of the two runs.

The heat transfer coefficient values showed the increase with the increase in blowing ratio up to the considered blowing ratio of 2.0 like as shown in the Fig.7. The heat transfer coefficient data also has shown the consistent repeatability between the two runs like as shown in the Fig.8. The peak values in the graphs indicate the row of film cooling holes locations. The heat transfer coefficient value is observed as approximately 400 W/m^2k at the downstream of 30 deg. row hole orientation angle.

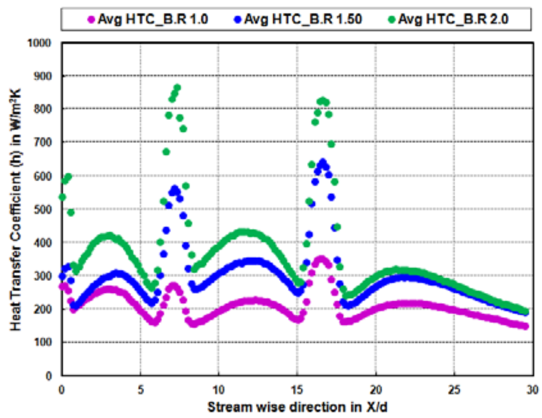

Fig.7: Experimentally evaluated heat transfer coefficient data in the flow direction.

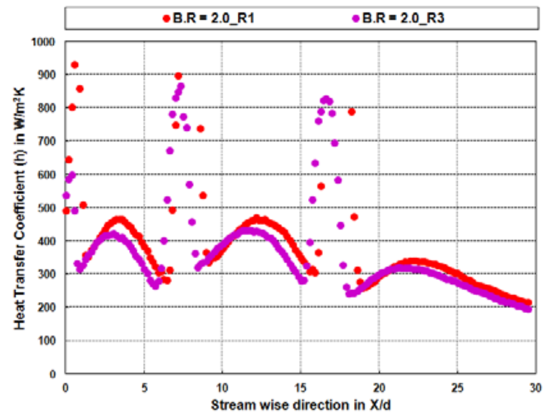

Fig.8: Experimentally evaluated heat transfer coefficient data of the two runs.

These results showed the increase in cooling effectiveness with the increase in blowing ratio up to the value of 2.0 and beyond this not shown the improvement in cooling effectiveness. The heat transfer coefficient values showed the increase with the increase in blowing ratio up to the considered range for this considered leading edge model.

6. CONCLUSIONS:

The present study deals with experimental investigation of adiabatic film cooling effectiveness and heat transfer coefficient data for gas turbine blade leading edge model. The test model has the five rows of cooling holes with cooling holes at an angle of 20 degrees in span wise direction at 18mm pitch spacing. From the experimental evaluation, the cooling effectiveness shown the increase with the increase in blowing ratio up to the blowing ratio of 2.0 and beyond this the cooling effectiveness not showed the improvement. Whereas the heat transfer coefficient data shown the increase with increase in blowing ratio up to the considered range of blowing ratio 2.0. From this experimental analysis, it is found that blowing ratio is the major parameter affecting both the film cooling effectiveness and heat transfer coefficients.

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Nomenclature

T	-	Temperature
P	-	Pressure
V	-	Velocity
A	-	Area
D.R	-	Density Ratio
B.R	-	Blowing Ratio
R_e	-	Reynolds Number
C_d	-	Coefficient of Discharge
X	-	Distance from the Stagnation point of the Leading Edge
D	-	Blade Leading Edge Diameter
d	-	Film Hole diameter
h	-	Convective heat transfer coefficient
p	-	Hole to hole pitch within a row of cooling holes
k	-	Thermal conductivity

Greek Symbols

η	-	Cooling Effectiveness
ρ	-	Density

Subscripts

i	-	Ideal
g	-	Gas
c	-	Coolant
aw	-	Adiabatic wall
ms	-	Mainstream

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