Film Cooling from Two Rows of Holes with Opposite Orientation Angles : Heat Transfer*

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Heat transfer coefficient ratio distributions are measured for film cooling from two rows of holes with opposite orientation angles. The inclination angle is fixed at 35° , and orientation angles are set to 45° for the downstream row and -45° for the upstream row. Four film cooling hole arrangements including inline and staggered configurations are investigated. The blowing ratios considered are 0.5, 1.0 and 2.0. Configuration effects on the heat transfer are not significant at the blowing ratio of 0.5. At the higher blowing ratios of 1.0 and 2.0, however, the inline configuration produces higher heat transfer coefficient distributions compared to others. In spite of the high heat transfer level, the inline configuration shows better film cooling performance, represented by the heat flux ratio, at the blowing ratios of 1.0 and 2.0 due to its higher adiabatic film cooling effectiveness.

Key Words: Film Cooling, Heat Transfer Coefficient, Heat Flux Ratio, TLC(Thermochromic Liquid Crystal), Injection Hole Geometry

1. Introduction

The efficiency of gas turbine engine depends highly on the turbine inlet temperature, but the inlet temperature is limited to the allowable temperature of blade material. Film cooling is commonly used to prevent thermal failure of turbine blades that could result from the operation at high temperature. In film cooling, cooling air bled from the compressor is discharged through holes on the turbine blade wall or the end wall. The coolant from holes forms a thin thermal insulation layer on the blade surface to protect the blade from being overheated by the hot gas flow from the combustor. In such a film cooling system its effectiveness is affected by blade geometry, film hole arrangements, free-stream conditions and coolant to free-stream velocity ratio, to name a few.

The compound angle orientation hole has two characteristic angles as illustrated in Fig. 1. The

inclination angle(α) is defined as the angle between the injection vector and its projection on x - z plane whereas the orientation angle(β) is defined as the angle between the streamwise direction and the projection of the injection vector on x - z plane. In the compound angle orientation system, the coolant is injected with a spanwise momentum component. Thus, the compound angle configuration provides more uniform film coverage. There exist complex three-dimensional flows in the film-cooled area in the case of the compound angle injection. As the orientation angle increases the counter-rotating vortices, which are usually observed in the simple angle injection, turn to a single strong vortex, and the strength of downstream secondary flow depends strongly on the velocity ratio⁽¹⁾. Previous studies, in general, show that the compound angle injection provides substantially improved effectiveness compared with the simple angle injection at the same spanwise hole spacing, normalized streamwise location and blowing ratio^{(2),(3)}. It is also noteworthy that the film coolant injected from compound angle holes interacts with the mainstream fluid more vigorously to increase the heat transfer

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Fig.1 Configuration of compound angle orientation.

coefficient⁽⁴⁾.

Past studies on the film cooling from two rows of holes with compound angle orientation adopted staggered hole configuration. They arranged each row of holes to have the same orientation angles. Since the compound angle injection has a lateral momentum component, the injectant trajectory moves in the z -direction as it flows downstream. In the present study, two rows of holes with opposite orientation angles are adopted to investigate the interactions of injectant flow with vortices. Four different hole configurations including inline and staggered ones are investigated. For all configurations effects of three blowing ratios of 0.5, 1.0, and 2.0 are studied. Heat transfer coefficient distributions are measured with thermochromic liquid crystal (TLC). The film cooling performance, represented by the heat flux ratio, is calculated from the heat transfer coefficient and the effectiveness data reported in the companion paper⁽⁵⁾.

Fig.3 Injection hole geometry.

Nomenclature

L	Film hole length
x, y, z	Streamwise, normal and spanwise coordinates
δ	Boundary layer thickness
η	Adiabatic film cooling effectiveness, $\left(= \frac{T_{aw} - T_{\infty}}{T_c - T_{\infty}} \right)$
Ψ	Overall film cooling effectiveness, $\left(= \frac{T_w - T_{\infty}}{T_c - T_{\infty}} \right)$

2. Experimental Apparatus and Procedures

An open type subsonic wind tunnel is used to generate the free-stream (Fig. 2(a)). A boundary layer trip wire of 1.8 mm diameter is located at $20D$, where D is the hole diameter, upstream from the film holes to produce a fully turbulent boundary layer flow at the exit of the holes. Experimental conditions are summarized in Table 1. As described in Fig. 2(b), the bottom plate of the duct consists of an upstream plate, one of four injection plates (Fig. 3) and a measurement plate. The measurement plate starts at $x/D = 1.0$. To measure the temperature distribution on the entire surface downstream of the injection holes, a thermochromic liquid crystal (TLC) sheet is used. The TLC sheet covers the ranges $1.0 \leq x/D \leq 31.0$ and $-12.0 \leq z/D \leq 12.0$. Beneath the TLC sheet, a 30 μ m thick stainless steel foil is used to generate a uniform heat flux. The stainless steel foil is heated by an AC electrical power supply. The current and the voltage across the heating element are measured to calculate input power. A CCD camera is used to capture TLC color images. The CCD camera is aligned perpendicular to the TLC sheet 1.2 m away to exclude any optical problems. Two 150W HQI lamps are used

(a) Schematic diagram of wind tunnel and coolant supply system

(b) configuration of measurement plate

Fig.2 Schematic diagram of experimental setup.

to illuminate the TLC sheet.

There are various techniques used with TLC for temperature measurements. Among many techniques, the steady-state, hue capturing method is adopted in this study. The liquid crystal used to measure heat transfer coefficient distributions has a color changing temperature range from 25°C to 45°C. Since the bandwidth of the TLC sheet is wide, the sheets can map the entire isothermal pattern of a surface from a single image. Robust TLC color-temperature response calibration, however, is necessary for high accuracy measurements. The relationship between the hue value of the color image and temperature is calculated using a polynomial curve fit by calibration. The free-stream and the injectant temperatures are measured with T-type thermocouples that are calibrated in a constant temperature bath with a precision platinum resistance thermometer.

The heat transfer coefficient in the film cooling situation can be defined as⁽⁶⁾:

$$q_f = h_f(T_{aw} - T_w) \quad (1)$$

Here, T_{aw} , T_w , h_f and q_f denote the adiabatic wall temperature, wall temperature, heat transfer coefficient and the heat flux, respectively. Another way of defining the heat transfer coefficient is as follows:

$$q_f = h(T_\infty - T_w) \quad (2)$$

Dividing both sides of Eq. (2) by Eq. (1), it follows that

$$\frac{h}{h_f} = 1 - \eta \frac{T_c - T_\infty}{T_w - T_\infty} \quad (3)$$

As can be seen in Eq. (3), when the coolant temperature and the free-stream temperature are the same, h_f is identical with h . In this study, the free-stream temperature and the injectant temperature are both fixed at 20°C, and the wall is heated with a uniform heat flux. The heat transfer coefficient h is then calculated from Eq. (2). The heat transfer coefficient results are presented in terms of the heat transfer coefficient ratio (h_f/h_o) to reduce the effect of mainstream Reynolds number. Here, h_f denotes the heat transfer coefficient with film injection, while h_o denotes the heat transfer coefficient without film cooling at the same location.

Since the objective of film cooling is to reduce the heat load to gas turbine components, it is necessary to compare the heat loads with and without film cooling. The heat transfer coefficient with film cooling is given by Eq. (1). The heat transfer coefficient without film cooling can be expressed as follows:

$$q_o = h_o(T_\infty - T_w) \quad (4)$$

The heat flux ratio is then defined as

$$\frac{q_f}{q_o} = \frac{h_f(T_{aw} - T_w)}{h_o(T_\infty - T_w)} = \frac{h_f}{h_o} \left(1 - \frac{\eta}{\Psi} \right) \quad (5)$$

where Ψ is the overall cooling effectiveness, which ranges from 0.5 to 0.7 for gas turbine blades with film cooling. A value of 0.6, which is typical of a film cooled turbine blade operation and the effectiveness value in the companion paper⁽⁵⁾ are used for the heat flux ratio calculation.

The uncertainty analysis is evaluated on 20 to 1 odds (95% confidence level). All the uncertainty values are evaluated from the method of single-sample experiments proposed by Kline and McClintock⁽⁷⁾. The uncertainty of the heat transfer coefficient ratio is evaluated at a typical value of $h_f/h_o = 1.2$. The analysis yields an uncertainty of 9.5%. The uncertainty of the heat flux ratio is 10.1% with a typical q_f/q_o value of 0.8.

3. Results and Discussions

3.1 Heat transfer coefficient ratio distributions

The coolant injection can affect the heat transfer characteristics over the surface in several ways. It can reduce heat transfer by thickening the thermal boundary layer. On the other hand, it can raise heat transfer by interaction with the free-stream in such a way that a vortex is formed to produce a high heat transfer region at the downwash location and the injectant acts as a blockage to the free-stream flow, which produces a region of high turbulence immediate downstream of injection holes.

At the blowing ratio of 0.5, Fig. 4 shows two peaks of the heat transfer coefficient ratio near the holes irrespective of configurations. The upper one is resulted from the turbulent motion and the lower one is from the downwash motion of the vortex. The effect of the turbulent motion tends to disappear at about $x/D = 10.0$ though slight differences can be seen according to the configurations. Downwash regions, however, show high heat transfer level throughout the measured surface.

Table 1. Experimental conditions

U_∞	10 m/s
D	10 mm
δ/D	1.50
$H (= \delta^*/\theta)$	1.36
L/D	4.0
$M (= \rho_c U_c / \rho_\infty U_\infty)$	0.5, 1.0, 2.0
$Re_D (= U_\infty D / \nu)$	6,570

Fig.4 Heat transfer coefficient ratio for $M = 0.5$.

When the blowing ratio increases to 1.0, the configuration effects become distinct as in Fig. 5. For configurations 3 and 4, the injectant trajectory separates from the wall with the increase of the blowing ratio. This is why the high heat transfer region corresponding to the downwash location can be observed where x/D exceeds 15.0. For configurations 1 and 2, however, the downwash region shows high heat transfer level from the immediate downstream of the holes due to the existence of the downwash motion of the upstream injectant at the hole exit⁽⁵⁾.

At the higher blowing ratio of 2.0, the variation according to configurations becomes more conspicuous as shown in Fig. 6. Configurations 3 and 4 give similar heat transfer coefficient distributions to those at the blowing ratio of 1.0. For configuration 1, however, the injectant attaches to the wall and produces high heat transfer regions at the downwash location from the near downstream of the holes.

3.2 Heat flux ratio

At the high blowing ratios of 1.0 and 2.0, the inline configuration shows the highest heat transfer coefficient distributions, which is expected to deteriorate the film cooling performance. From the adiabatic film cooling effectiveness data, however, the

configuration is expected to present the best film cooling coverage⁽⁵⁾. To illustrate the combined effects of the adiabatic film cooling effectiveness and the heat transfer coefficient, the heat flux ratio is evaluated using Eq. (5).

To compare the heat flux ratio of different configurations over a full range of blowing ratios, a spatially averaged heat flux ratio is presented in Fig. 7. From the definitions of the heat flux ratio, the heat flux ratio less than unity indicates that the surface thermal load is reduced due to film cooling.

At the blowing ratio of 0.5, the contribution of film coverage is larger than that of heat transfer increase. Configuration 2 shows more uniform adiabatic film cooling effectiveness⁽⁵⁾ to yield the best film cooling performance. With the downwash motion at the hole exits, the inline configuration shows much higher heat transfer coefficient and film cooling effectiveness compared to other configurations at the blowing ratio of 1.0 and 2.0. The heat flux ratio shows that effects of the film cooling effectiveness suppress those of heat transfer. This is why the inline configuration presents much better film cooling performance compared to other configurations at higher blowing ratios.

Fig.5 Heat transfer coefficient ratio for $M = 1.0$.

Fig.6 Heat transfer coefficient ratio for $M = 2.0$.

Fig.7 Space averaged heat flux ratio.

4. Conclusions

The effects of film hole configurations and blowing ratios on heat transfer and heat flux ratio in film cooling from two rows of holes with opposite orientation angles are investigated. Four configurations including inline and staggered ones and blowing ratios of 0.5, 1.0 and 2.0 are considered. The heat transfer coefficient distributions are measured in detail with a thermochromic liquid crystal technique. The film cooling performance is explained by the heat transfer coefficient data combined with the adiabatic film cooling effectiveness. Some important observations are drawn and summarized below.

1. Configuration effect on heat transfer is not significant at the lower blowing ratio such as 0.5.
2. The inline configuration shows the highest heat transfer coefficient distributions at the blowing ratios of 1.0 and 2.0 because the downwash motion plays a dominant role from the immediate downstream of the holes.

3. In spite of the high heat transfer, the inline configuration yields better film cooling performance due to its high adiabatic film cooling effectiveness.

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