Effects of Turbulence Promoters of Gas Turbine Blades on Film Cooling Performance*

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Abstract
The effects of orientation of turbulence promoting ribs in the internal cooling air passage on the film cooling performance of outer surface of gas turbine blades have been studied experimentally. Both the adiabatic film cooling effectiveness and the heat transfer coefficient downstream of the film-cooling hole of a flat plate were measured for two different orientations of the turbulence promoting ribs. It is found that the adiabatic film cooling effectiveness and heat transfer coefficient on the external surface are considerably affected by the inversion of the rib orientation in the internal passage. The film cooling performance is evaluated in terms of the net heat flux reduction and the net surface temperature reduction, both of which are calculated from the measurement of adiabatic film cooling effectiveness and heat transfer coefficient. The evaluation of these quantities has clarified the effects of the internal cooling structure on the film cooling performance.

Key words: Gas Turbine Blade, Film Cooling, Turbulence Promoting Rib, Heat Transfer Coefficient, Adiabatic Film Cooling Effectiveness, Net Heat Flux Reduction, Net Surface Temperature Reduction

1. Introduction

Improvement of the thermal efficiency of gas-turbine combined-cycle plant reduces power generation costs and environmental impact. Cooling technologies developed for turbine blades contribute to the increase of the gas turbine temperature and thus to the improvement of the thermal efficiency. Further improvement of the thermal efficiency needs substantial reduction of the mass flow rate of cooling air because the bleeding of the cooling air from the compressor is always associated with the loss of power generation. This reduction can be achieved by the improvement of cooling technologies for turbine blades.

Recent turbine blades employ internal convection cooling with ribbed passages in the blades and external film cooling with air jet onto the surface in order to decrease blade surface temperature and heat flux through the blade. The film cooling and the internal convection cooling have been studied extensively since 1960s and many papers can be found on their individual effects. For example, there are reports on the effects of the density ratio (1), the blowing angle (2), and the geometry of injection holes (3)(4) for the film cooling while there are reports on the effects of geometry of turbulence promoters (5)(6), the passage bend (7), and the blade rotation (8) for the internal convection cooling. In contrast, studies on
the combined effects of the film cooling and the internal convection cooling are rather scarce except for a few studies such as those of Gritsch et al. (9), Fawcett et al. (10), Kissel et al. (11), and Sakai et al. (12). The authors showed that the film cooling characteristics are different between the case where the cooling air is supplied from the internal flow and the case where it is supplied from a stagnant plenum. Also, Sakai et al. (12) showed that the adiabatic film cooling effectiveness varies greatly depending on the Reynolds number of the internal flow. These results suggest a possibility that the film cooling performance can be improved by the optimization of the internal cooling structure.

The present study aims at clarifying the effect of the orientation of turbulence promoting ribs installed in the internal cooling air passage on the film cooling performance of outer surface of gas turbine blades. As shown in Fig. 1, the orientation refers to the relation between the direction of the main flow on the outer surface and the direction of turbulence promoting ribs. The rib orientation called ‘Rib1’ is such as to deflect the cooling air favorably toward the main flow while the rib orientation called ‘Rib2’ is such as to deflect the cooling air adversely. These two configurations are widely used for actual turbine blades, but the effect of their orientation on the film cooling performance has not yet been understood well. Their effect is evaluated in this study in terms of the variations of the surface temperature of turbine blade and of the heat flux through the turbine blade. This evaluation is made through the measurement of the adiabatic film cooling effectiveness (AFCE, hereafter) and the heat transfer coefficient (HTC, hereafter) at the outer surface of the test plate that is prepared for the present experiment. Conventionally, the AFCE has attracted much more attention than the HTC. This is because (1) the measurement and the CFD calculation of AFCE is easier than that of HTC and (2) a simple assumption (13) of the HTC without film cooling being equal to the HTC with film cooling is often made for the outer surface except for the area close to the injection hole. It can be shown that this assumption allows the evaluation of the surface temperature of turbine blade to be made without measuring HTC. As shown later in this paper, however, this assumption is not accepted for the present conditions and the direct measurement of HTC is vital for the accurate and reliable evaluation of the effect of rib orientation on the cooling performance.

Baldauf et al. (14)(15)(16) measured AFCE by using an infrared (IR) camera and evaluated HTC for a wide range of density ratio and blowing ratio. They proposed a simple one-dimensional heat-transfer model to account for the combined effects of the film cooling and the internal convection cooling. They presented a non-dimensional quantity called “net heat flux reduction” (NHFR, hereafter) that is expressed by AFCE and HTC. A similar approach is taken here and it is extended to propose a new non-dimensional parameter called “net surface temperature reduction” (NSTR, hereafter) that is to evaluate, together with NHFR, the effect of the orientation of turbulence promoting ribs. It is shown in this paper that the behaviors of NHFR and NSTR are significantly dependent on the rib orientation and such behaviors are not correctly captured if the assumption on HTC mentioned above is made.

Nomenclature

\[ a \] : blowing angle, 35°

\[ BR \] : blowing ratio
\( d \) : film-cooling hole diameter, 20mm  
\( h \) : heat transfer coefficient, W/m\(^2\)K  
\( L \) : ejection hole length, 70mm = 3.5\( d \)  
\( p \) : pitch  
\( q \) : heat flux, W/m\(^2\)  
\( Re \) : Reynolds number, \( Ud/\nu \)  
\( t \) : thickness of blade metal or thermal barrier coating, m  
\( T \) : temperature, K  
\( T_u \) : turbulence intensity, %  
\( U \) : mean velocity, m/s  
\( x \) : x-coordinate, streamwise  
\( y \) : y-coordinate, lateral direction  
\( z \) : z-coordinate, height direction  
\( \alpha \) : ratio of heat transfer coefficients, \( h_W / h_0 \)  
\( \beta \) : inclined angle of turbulence promoting rib, degree  
\( \gamma \) : net heat flux reduction (NHFR), \( 1 - q_f / q_0 \)  
\( \eta \) : adiabatic film cooling effectiveness, \( (T_G - T_{AW}) / (T_G - T_C) \)  
\( \lambda \) : thermal conductivity, W/mK  
\( \theta \) : dimensionless wall temperature, \( (T_G - T_C) / (T_G - T_W) \)  
\( \nu \) : kinetic viscosity coefficient, m\(^2\)/s  
\( \rho \) : density, kg/m\(^3\)  
\( \chi \) : net surface temperature reduction (NSTR), \( (T_{W0} - T_W) / (T_G - T_C) \)  

**Subscripts**  
\( Ave \) : average  
\( AW \) : adiabatic wall or blade surface  
\( C \) : coolant air  
\( CW \) : inner surface  
\( f \) : cooling film  
\( G \) : hot gas or main flow  
\( jet \) : exit of film cooling hole  
\( loss \) : loss  
\( metal \) : blade metal  
\( MW \) : blade metal surface (TBC-metal interface)  
\( s \) : spatial  
\( supply \) : supply  
\( TBC \) : thermal barrier coating  
\( W \) : wall  
\( 0 \) : without ejection, reference values  

**Superscripts**  
\( \bar{\cdot} \) : lateral (or area) average  

**Abbreviation**  
AFCE : adiabatic film cooling effectiveness  
HTC : heat transfer coefficient  
NHFR : net heat flux reduction  
NSTR : net surface temperature reduction
2. Formulation of film cooling

The effectiveness of film cooling can be discussed by means of the reductions of the temperature of the blade outer surface and of the heat flux which passes through the blade. These two kinds of reduction are quantified in terms of NHFR and NSTR and their formulation is given here by considering a one-dimensional heat-transfer model proposed by Baldauf et al. (14).

Figure 2 illustrates a model for heat transfer from the blade outer surface to the internal cooling passage. The left side shows the state without film cooling, and the right side shows the state with film cooling. The blade outer surface is covered with a thermal barrier coating (thermal conductivity $\lambda_{TBC}$, thickness $t_{TBC}$). Heat fluxes, $q_0$ and $q_f$, passing through the blade in the states without film cooling and with film cooling, respectively, are given by the following equations:

$$q_0 = \left(\frac{1}{h_b} + \frac{1}{h_w}\right)^{-1} (T_G - T_C)$$

(1)

$$q_f = \left(\frac{1}{h_f} + \frac{1}{h_w}\right)^{-1} (T_f - T_C)$$

(2)

Here, it is assumed that $h_w = \left(\frac{t_{TBC}}{\lambda_{TBC}} + \frac{1}{\lambda_{metal}} + \frac{1}{h_c}\right)^{-1}$ does not change due to the presence of film cooling. The boundary layer over the blade outer surface is disturbed by film cooling, and thus in general $h_b < h_c$.

Using the blade outer surface temperature in the state with film cooling, $q_f$ is given by the following equation.

$$q_f = h_f (T_f - T_w)$$

(3)

If the surface temperature of thermally insulated blade (i.e., the adiabatic film temperature) is taken to be $T_{AW}$, then it is evident from the equation above that $T_f = T_{AW}$. Therefore,

$$q_f = h_f (T_{AW} - T_w)$$

(4)

The reduction in heat flux is evaluated with the following NHFR $\gamma$.

$$\gamma = 1 - \frac{q_f}{q_0} = 1 - \left(\frac{1+\alpha}{\alpha + \left(h_f/h_b\right)}\right) (1-\eta)$$

(5)

Here,

$$\alpha = \frac{h_w}{h_b} \quad \text{and} \quad \eta = \frac{T_G - T_f}{T_G - T_C} = \frac{T_0 - T_{AW}}{T_G - T_C}$$

(6), (7)

$\eta$ is the AFCE.

The equation corresponding to Eq. (4) in the case without film cooling is as follows.

$$q_0 = h_b (T_G - T_{wo})$$

(8)
The following NSTR $\chi$ can be obtained from the above equation and Eq. (4).

$$\chi = \frac{T_{W0} - T_{W}}{T_{0} - T_{C}} = \eta - \frac{\alpha}{1+\alpha} \left( \frac{1-\gamma}{h_f/h_0} \right)$$

(9)

From Eqs. (5) and (9), it is evident that the effectiveness of film cooling is quantified by $h_f/h_0$, $\eta$ and $\alpha$. In this study, $\gamma$ and $\chi$ are evaluated by measuring $h_f/h_0$ and $\eta$, while $\alpha$ is estimated using the results for the first stage blade with its turbine inlet temperature of 1300°C. From the previous analysis, the following values are used: $\alpha = 0.48$ for the blade with TBC and $\alpha = 1.05$ for the blade without TBC. Note that if $\alpha = 0$ is substituted into Eq. (9), then $\chi = \eta$ and it is possible to evaluate NSTR from AFCE. The conventional assumption $h_f = h_0$ leads to $\gamma = \eta$ from Eq. (5) and $\chi = \eta / (1+\alpha)$ from Eq. (9), resulting in that both NHFR and NSTR can be evaluated from AFCE ($\alpha$). On the other hand, if it is assumed that $h_f >> h_0$ in Eq. (9), $\chi = \eta - 0.32$ for $\alpha = 0.48$, and $\chi = \eta - 0.51$ for $\alpha = 1.05$. As shown later, the assumptions on $h_f$ are not justified for the conditions considered in the present study.

3. Experiment

3.1 Experimental equipment

An overview of the flow facility is shown in Fig. 3. This apparatus is primarily comprised of a main flow section which has a test section simulating a turbine blade surface, a secondary flow section which simulates the internal cooling passage of the blade, and a film-cooling hole which connects the secondary flow section and the main test section.

The main air flow comes into the test section after passing through expansion, straightening and contraction sections. The resultant turbulence level is less than 0.5%. The test section is an acrylic rectangular duct having 240mm width and 100mm height. Three film-cooling holes are drilled in the bottom plates as shown in the side-view schematic in Fig. 3. Their diameter, $d$, is 20mm and inclination angle to the main flow, $\alpha$, is 35°. Their lateral pitch, $p$, is 60mm.

The secondary flow section has a square cross-section of 60mm × 60mm and is placed beneath the main flow section in an orthogonal configuration. The volume flow rates in the secondary flow section are measured at locations upstream and downstream of the film-cooling holes so that the volume flow rate of the air injected through the film-cooling holes is evaluated as a difference between the two measurements. For the measurement of AFCE, the air in the secondary flow section is pre-heated with a heater placed at the exit of the blower. As shown in Fig. 1, three different internal geometries of the secondary flow section are considered, i.e., those denoted as ‘Rib1’, ‘Rib2’ and ‘noRib’. Both Rib1 and Rib2 have turbulence promoting ribs while noRib does not have any ribs and provides a reference case. Individual rib has a square cross section of 6mm × 6mm, and pairs of ribs are installed in an opposing fashion on top and bottom walls of the secondary flow section.
As depicted in Fig. 1, a total of ten pairs of ribs are installed at a regular pitch of 60mm. The resultant pitch-to-height ratio is 10, which is within typical values of 8 to 12\(^{18}\). Rib1 and Rib2 are different in their rib orientation in that Rib1 deflects the cooling air favorably toward the main flow while Rib2 does adversely. The inclination angles of ribs to the main flow are 60° and 120° in Rib1 and Rib2, respectively.

The coordinate system in this experiment defines the main flow direction as \(x\), the lateral direction as \(y\) and the height direction as \(z\) with their origin placed on the center of the middle film-cooling hole.

### 3.2 Experimental conditions

The experimental conditions are summarized in Table 1. The mean velocities in the main test section, \(U_G\), and in the secondary flow section, \(U_C\), defined at locations upstream of the film-cooling holes, are both 20m/s. The resultant Reynolds number is \(2.5 \times 10^4\) in the main test section and they are \(2.1 \times 10^4\) and \(2.5 \times 10^4\) in the secondary test section for the measurements of AFCE and HTC, respectively. These values of the Reynolds number are comparable to those in the operating conditions of actual gas turbines. The temperature of the main flow, \(T_G\), is a room temperature (293K-299K), while that of the secondary flow, \(T_C\), is adjusted either to 333K for AFCE or to \(T_G \pm 0.2K\) for HTC. The mean velocity of injection air through each film-cooling hole \(U_{jet}\) is calculated from the volume flow rate measured as mentioned above. The blowing ratio, \(BR\), is defined as follows:

\[
BR = \frac{\rho_{jet} U_{jet}}{\rho_G U_G}
\]  

The present experiments are conducted for \(BR = 0.5, 0.75, 1.00\) and 1.25, for corresponding to the operation range in actual turbine blade.

### 3.3 Measurement of AFCE

In order to estimate NHFR and NSTR, the measurement results of AFCE carried out by Sakai et al.\(^{19,20}\) are used and analyzed here. Their measurements were carried out in the same flow facility under the same experimental conditions as presently. An adiabatic boundary condition on the heat transfer surface was realized by using a bakelite plate 12mm in thickness and 0.42W/(m*K) in thermal conductivity. Furthermore, a thermally insulating material 40mm in thickness and 0.04W/(m*K) in thermal conductivity was mounted to the back on the bakelite plate. For the measurement of \(T_{wy}(x,y)\), a total of 117 pairs of T-type thermocouple with 0.2mm wire diameter were embedded in the bakelite plate in a grid pattern which has 9 points at 4mm intervals in \(x\) direction and 13 points at 5mm intervals in \(y\) direction. The streamwise position of this plate was changed to a total of 7 locations in order to cover the heat transfer surface of \(1.7 < x/d < 18.2\) and \(-1.3 < y/d < 1.3\) with 69*9 measurement points. After the temperature field is fully stabilized, the temperature distribution was measured. Data sampling was made at 100Hz, for 60s to obtain time averaged temperature.

The local AFCE, \(\eta(x,y)\), and the laterally averaged AFCE, \(\overline{\eta}(x)\), are evaluated from the measured \(T_{wy}(x,y)\) as follows:

\[
\eta(x,y) = \frac{(T_G - T_{wy}(x,y))}{(T_G - T_C)}
\]  

\[
\overline{\eta}(x) = \frac{(T_G - \overline{T_{wy}}(x))}{(T_G - T_C)}
\]
3.4 Measurement of HTC

The temperature measurement of the heat-transfer surface under the condition of constant heat flux is made for \( h_0 \) and \( h_f \). The evaluation of \( h_0 \) is based on Eq. (8), while that of \( h_f \) is made from Eq. (4) by using a convenient relation, i.e., \( T_{AW} = T_C = T_G \), that holds in the case where \( T_C = T_G \) (i.e., isothermal injection case). Consequently, \( h_f \) is evaluated from the following equation:

\[
q_f = h_f \left( T_G - T_w \right)
\]

Figure 4 shows the wall-temperature measurement system. A stainless steel foil 400mm long, 240mm wide and 0.05mm thick is glued to the surface of the main test section to cover the area downstream of the film-cooling hole. The condition of constant heat flux is generated by Joule heating with DC current via copper-plate electrodes. The surface of the stainless steel foil was coated with black paint with known emissivity (0.94). Its surface temperature distribution is measured using an IR camera through an observation window that is installed in the top panel of the duct. A Germanium plate, 132mm diameter and 3mm thickness, with anti-IR-reflection coating on both sides of the plate is used. The streamwise location of this observation window is changed according to the location of surface temperature measurement. To minimize conduction heat loss through the bakelite plate, it is covered with thermally insulating material as mentioned above. The present IR-camera measurement is checked and corrected by referring to the readings of T-type thermocouples embedded in the bakelite plate.

The net heat flux, \( q \), transferred by the fluid is evaluated as \( q = q_{supply} - q_{loss} \) where \( q_{supply} \) is calculated from the electrically supplied power and \( q_{loss} \) is estimated from radiation and conduction heat losses. These heat losses are estimated to be 8% and 3% of \( q_{supply} \), respectively.

All data acquisitions are made for 1 minute at a sampling rate of 100Hz, except for IR images that are acquired for 30s at an interval of 3s. As a result, the local HTC, \( h(x,y) \), is obtained as follows:

\[
h(x,y) = \frac{q(T_w(x,y) - T_G)}{q_{supply} - q_{loss}} = \frac{(T_w(x,y) - T_G)}{(T_w(x,y) - T_G)}
\]

The ratio of local HTCs and that of laterally-averaged HTCs are defined as follows:

Ratio of local HTCs

\[
h_f(x,y)/h_0(x,y)
\]

Ratio of laterally averaged HTCs

\[
\overline{h}_f(x)/\overline{h}_0(x)
\]

where \( \overline{h}(x) \) is calculated from averaging of \( h(x,y) \) over \(-30 \leq y \leq 30\)mm. Although not shown here, the present \( \overline{h}_0(x) \) is consistent with the empirical one\(^{(21)}\).

3.5 Measurement Uncertainty

Possible error sources in the measurement of HTC are as follows:

(a) measurement of electrically supplied power,
(b) estimation of the conduction and radiation heat loss, \( q_{loss} \),
(c) lateral conduction heat loss due to planar temperature variation in the stainless steel foil,
(d) temperature measurement using the present IR camera,
(e) light reflection at the observation window for IR camera measurement, and
(f) temperature measurements using thermocouples for \( T_C \) and for IR camera correction.
The error source (c) is found to be negligibly small, in accordance with Funazaki et al.\textsuperscript{(22)}. The total uncertainty associated with HTC measurement is evaluated to be approximately 10%. This value is slightly higher than the total uncertainty associated with AFCE which is reported to be approximately 7%\textsuperscript{(19)}.

4. Results and discussion about AFCE

4.1 Laterally averaged AFCE

Figure 5 shows the streamwise distribution of the laterally averaged AFCE \( \overline{\eta} \) at the blowing ratios \( BR = 0.5, 0.75, 1.00 \) and \( 1.25 \), where noRib is given only for \( BR = 0.5 \) and \( 0.75 \). For both the rib orientations, \( \overline{\eta} \) has a high value near the film-cooling hole and decreases gradually in the downstream. At \( BR = 1.25 \), the decrease of \( \overline{\eta} \) in the streamwise distribution is small, except for the region near the film-cooling hole. Such behaviors are in agreement with the data reported by Kohli et al.\textsuperscript{(23)} for a similar flow geometry with a plenum chamber, as plotted in Fig. 5 for \( BR = 0.5 \).

Figure 6 shows the \( BR \) dependency of \( \overline{\eta} \) at \( x/d = 2.9, 7.1, 11.1 \) and \( 15.1 \). Whereas \( \overline{\eta} \) for Rib1 decreases monotonically with \( BR \), \( \overline{\eta} \) for Rib2 increases with \( BR \) (except for 0.75 < \( BR < 1.25 \) at \( x/d = 2.9 \)). The results show that for \( BR < 1.00 \), \( \overline{\eta} \) for Rib1 is higher than that for Rib2, and this relationship is reversed for \( BR = 1.25 \). Note that \( \overline{\eta} \) for noRib takes intermediate values between Rib1 and Rib2. It is evident that the \( BR \) dependency of \( \overline{\eta} \) is influenced considerably by the rib orientation. Possible reasons are: the difference in the way the blown cooling air penetrates into the main flow near the film-cooling hole and the difference in its lift-off from the wall surface in the region downstream of the film-cooling hole. This conjecture should be clarified by the detailed understanding of the structure of the velocity field and temperature field downstream from the film-cooling hole.

Fig. 5 Effects of rib orientation on the streamwise distribution of laterally averaged AFCE (-1.3 < \( y/d < 1.3 \) lateral average)
4.2 Local Distributions of AFCE

Figure 7 shows the lateral distributions of local AFCE $\eta$ at $x/d = 2.9$ for each $BR$. There is a marked effect of the rib orientation in the sense that $\eta$ for Rib1 has a symmetric distribution about $y/d = 0$, whereas $\eta$ for Rib2 and noRib is asymmetrical. More specifically, the lateral maximum for Rib1 is located at $y/d = 0$ and it becomes the largest for $BR = 0.5$. In contrast, the lateral maximum for Rib2 is located in the region of $y/d < 0$ and it becomes the largest at $BR = 0.75$.

5. Results and Discussion about HTCs ratio

5.1 Ratio of laterally averaged HTCs

Figure 8 shows the streamwise distributions of the ratio of laterally averaged HTCs $\overline{h}_{j}/\overline{h}_{\infty}$ at $BR = 0.5, 0.75, 1.0$ and $1.25$. For comparison, the data for $BR = 0.5$ taken by Goldstein et al.[24] in a similar flow geometry with a plenum chamber are included. The agreement between their data and the present results supports the validity for the present measurement of HTC.
For lower BR (i.e., 0.50 and 0.75), $\overline{h_f}/\overline{h_0}$ exhibits almost no differences between Rib1 and Rib2 both showing a relatively high value near the film-cooling hole. On the other hand, $\overline{h_f}/\overline{h_0}$ for BR = 1.25 has a lower value near the film-cooling hole and exhibits a broad maximum. At this BR, $\overline{h_f}/\overline{h_0}$ for Rib2 is consistently higher than that for Rib1, as is evident from the result that the broad maxima for Rib1 and Rib2 are about 1.2 and 1.3, respectively.

Figure 8 shows the BR dependency of $\overline{h_f}/\overline{h_0}$ at $x/d = 2.9, 7.0, 11.1$ and 15.1. The results show that for all rib orientations, $\overline{h_f}/\overline{h_0}$ increases monotonically with BR (except for BR = 1.00-1.25 at $x/d = 2.9$). Especially near the film-cooling hole, $\overline{h_f}/\overline{h_0}$ for Rib2 higher than those for Rib1 and noRib at almost whole BR range of the measurements. And the difference due to the rib orientation is found to be more significant for higher BR.

Fig. 8 Effect of rib orientation on the ratio of laterally averaged HTCs in the streamwise distribution (-1.5 < y/d < 1.5 lateral average)

Figure 9 shows the BR dependency of $\overline{h_f}/\overline{h_0}$ at $x/d = 2.9, 7.0, 11.1$ and 15.1. The results show that for all rib orientations, $\overline{h_f}/\overline{h_0}$ increases monotonically with BR (except for BR = 1.00-1.25 at $x/d = 2.9$). Especially near the film-cooling hole, $\overline{h_f}/\overline{h_0}$ for Rib2 higher than those for Rib1 and noRib at almost whole BR range of the measurements. And the difference due to the rib orientation is found to be more significant for higher BR.

Fig. 9 Effect of BR on the ratio of laterally averaged HTCs at four positions in the streamwise distribution
5.2 Ratio of local HTCs

Figure 10 shows the ratio of local HTCs $h_f / h_0$ in the region downstream of the film-cooling hole. It is seen that the distributions are notably dependent on the rib orientations. Remarkable difference between Rib1 and Rib2 is the presence of a region of high $h_f / h_0$ immediately downstream of the film-cooling hole for Rib2. This is the reason why $h_f / h_0$ for Rib2 has a higher value than that for Rib1 near the film-cooling hole, particularly at high $BR$, in Fig. 9. This suggests that the air injected from the hole in Rib2 should have a structure that is responsible for heat-transfer enhancement in this region. Compared to noRib and Rib2, the distribution for Rib1 is rather symmetrical about $y/d = 0$ and exhibits streaky pattern that are elongated in the streamwise direction for $x/d > 5$, where an elongated region of low $h_f / h_0$ is sandwiched by a pair of elongated regions of high $h_f / h_0$. Such pattern should be the evidence for the presence of a pair of counter-rotating vortices that are often found in the film cooling.$^{(25)}$

Figure 11 shows the lateral distribution of $h_f(x,y)/h_0(x)$ at $x/d = 2.9$ for each $BR$. There is again a marked effect of the rib orientation in the sense that $h_f(x,y)/h_0(x)$ for Rib1 has double peaks while that for Rib2 has a single peak. The difference becomes more significant at higher $BR$. The peak for Rib2 is higher by about 0.3 than that for Rib1 at $BR = 1.25$.

Figure 12 shows the lateral distribution of $h_f(x,y)/h_0(x)$ at $x/d = 7.0$. It is seen that the effect of film cooling on HTC becomes much less significant at this downstream location.

![Fig. 10 The ratio of local HTCs in the region downstream of the film-cooling hole (1.26 < x/d < 21.2, -1.5 < y/d < 1.5)](image-url)
6. Evaluation of the cooling performance

6.1 Evaluation of area averaged quantities

Film cooling performance is evaluated by calculating $\gamma$ in Eq. (5) and $\chi$ in Eq. (9), as mentioned in §2. In this evaluation, $\gamma$ and $\chi$ are considered in terms of their area average, i.e., $\gamma_s$ and $\chi_s$, because (1) it simplifies the comparison of the effect of rib orientation on the overall film-cooling performance and (2) $h_W$ in Eqs. (5) and (9) is not measured directly but estimated for the conditions of commercially operated turbine blades. The values of $\gamma_s$ and $\chi_s$ are respectively calculated from $\gamma(x, y)$ and $\chi(x, y)$, which are obtained from the measured $\eta(x, y)$ and $h_f(x, y)/h_0(x, y)$, as follows:

$$\gamma_s = \frac{1}{819} \sum_{i=1}^{61} \sum_{j=1}^{13} \gamma(i, j)$$  \hspace{0.5cm} (17)

$$\chi_s = \frac{1}{819} \sum_{i=1}^{61} \sum_{j=1}^{13} \chi(i, j)$$  \hspace{0.5cm} (18)

Fig. 11 Effect of rib orientation on the lateral distribution of the HTCs ratio ($x/d = 2.9$)

Fig. 12 Effect of rib orientation on the lateral distribution of the HTCs ratio ($x/d = 7.0$)
where the averaging area is \(1.7 < x/d < 18.2\) and \(-1.3 < y/d < 1.3\), which covers a total of 819 measuring points. Note that \(\alpha = 0.48\) for the case with using TBC and \(\alpha = 1.05\) for the case without using TBC are assumed here\(^{(17)}\). Baldauf et al.\(^{(16)}\) used \(\alpha = 1.0\) as a representative value corresponding to the condition of actual turbine blade. Their value is very close to the present value for the case without using TBC.

### 6.2 NHFR and NSTR estimate

Figure 13 shows the dependency of \(\bar{Y}_s\) and \(\bar{X}_s\) on \(BR\) for Rib1 and Rib2 for the case with using TBC. In order to highlight the importance of direct measurement of HTC, plots of \(Y_s\) and \(X_s\) that are calculated with frequently referred assumption, i.e., \(h_f = h_0\), are included in this figure. The characteristics of \(Y_s\) and \(X_s\) may be summarized as follows.

1. Both \(\bar{Y}_s\) and \(\bar{X}_s\) for Rib1 show a monotonic and comparatively steep decrease with \(BR\), suggesting that the overall film-cooling performance simply deteriorates with increasing \(BR\) for the range considered here (i.e., \(0.5 < BR < 1.25\)). In fact, \(Y_s\) decreases from 0.17 at \(BR = 0.5\) to only 0.05 at \(BR = 1.25\) while \(X_s\) does from 0.11 at \(BR = 0.5\) to only 0.04 at \(BR = 1.25\). Such an enormous decrease is undesirable in the operation of actual turbine blades.

2. Similar behavior is seen for Rib1 with assuming \(h_f = h_0\) but the decrease of \(\bar{Y}_s\) and \(\bar{X}_s\) with \(BR\) is considerably smaller, meaning that the aforementioned undesired effect due to the increase of \(BR\) on the overall film-cooling performance could be overlooked if \(h_f = h_0\) is assumed.

3. Dependency of \(\bar{Y}_s\) and \(\bar{X}_s\) for Rib2 on \(BR\) is different from that for Rib1 in that those for Rib2 exhibit a broad peak at around \(BR = 0.75\) and they are rather insensitive to \(BR\) in the range considered here.

4. But the values of \(\bar{Y}_s\) and \(\bar{X}_s\) for Rib2 at \(BR = 0.75\) are much smaller than those for Rib1 at the same \(BR\), and those for Rib2 start to exceed those for Rib1 for \(BR > 1.00\).

5. Noteworthy behavior of \(\bar{Y}_s\) and \(\bar{X}_s\) for Rib2 with assuming \(h_f = h_0\) is their increasing trend with \(BR\) in the range considered here, possibly leading to an over expectation for the effect of \(BR\).

The results for the case without using TBC are presented in Fig. 14. General behaviors of \(Y_s\) and \(X_s\) are similar to those for the case with using TBC, but the overall film-cooling performance are deteriorated. For this, it can be confirmed that the film cooling combined with TBC is quite effective for the reductions of heat flux and blade surface temperature.
7. Conclusion

This paper reports the effects of orientation of turbulence promoting ribs in the internal cooling air passage on the film cooling performance of outer surface of gas turbine blades. The AFCE and the HTC on a flat plate for two different rib orientations, Rib1 and Rib2 (Fig. 1), have been measured to evaluate the NHFR and the NSTR. The findings of this study can be summarized as follows;

(1) The measured AFCE shows that its dependency on $BR$ is influenced by the rib orientation. More specifically, the laterally averaged AFCE for Rib1 is higher than that for Rib2 for $BR < 1.00$ but this relationship is reversed for $BR = 1.25$.

(2) The local AFCE for Rib1 has a laterally symmetrical distribution while that for Rib2 has a skewed distribution.

(3) The laterally averaged HTC increases with $BR$ for Rib1 and Rib2 similarly except for the region near the film cooling hole where that for Rib2 increases more than that for Rib1.

(4) The rib orientations affect the local HTC distributions, particularly in the region near the cooling hole where Rib2 shows a single peak of comparatively large value whereas Rib1 shows rather uniform, laterally symmetrical distributions with double peaks.

(5) The evaluation of NHFR and NSTR from the measured AFCE and HTC has clarified that the film cooling performance is appreciably dependent on the rib orientation and that the film cooling performance for Rib1 decreases monotonically and significantly with $BR$ whereas that for Rib2 is rather insensitive to $BR$.

(6) The conventional assumption of $h_f = h_0$ would lead to an optimistic estimation of NHFR and NSTR, thus indicating that the direct measurement of HTC is needed for more reliable and safer assessment of the film cooling performance.

(7) The application of TBC improves both NHFR and NSTR while it does not change their dependency on $BR$.

References


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