EXPERIMENTAL AND NUMERICAL STUDIES ON LEADING EDGE FILM COOLING PERFORMANCE: EFFECTS OF HOLE EXIT SHAPE AND FREESTREAM TURBULENCE

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ABSTRACT
This study deals with the experimental and numerical studies of the effect of hole exit shape and free-stream turbulence on turbine blade leading edge film cooling. The study examines several test cases with two blowing ratios (BR=1.0 and 2.0) and three mainstream turbulence intensities (1.0, 3.3 and 12.0 %) using two types of leading edge models with cylindrical holes and diffuser holes [1]. The leading edge model consists of a semi-circular part of 80mm diameter and a flat after-body. In this study total pressure loss coefficient is measured by total pressure probe. Film effectiveness and heat transfer coefficient on the model surface are measured by the transient method using thermochromatic liquid crystal with video camera. In addition, detailed investigation of the film cooling is carried out using CFD simulations. RANS approach using Shear Stress Transport turbulence model and Detached Eddy Simulation (DES) approach are employed to solve the flow field. In the case of diffuser hole, the effect of mainstream turbulence intensity appears significant, and its spanwise averaged film effectiveness is decreased.

NOMENCLATURE
- BR: Blowing ratio (= \( \rho_c U_c / \rho_m U_m \))
- \( C_p \): Pressure coefficient
- \( C_{p,t} \): Total pressure loss coefficient
- \( D \): Diameter of the leading edge model, mm
- \( d \): Diameter of film cooling hole, mm
- \( h \): Heat transfer coefficient, W/(m\(^2\)K)
- \( \dot{m} \): Mass flow rate, Kg/s
- \( P \): Pressure, Pa
- \( \text{Re}_D \): Reynolds number
- \( T \): Temperature, K
- \( TLC \): Thermochromatic liquid crystal
- \( T_u \): Turbulence intensity

\( t \): Time, s
\( U \): Main stream velocity, m/s
\( x, y, z \): Cartesian coordinates, mm
\( \alpha \): Angle, deg
\( \eta \): Film cooling effectiveness
\( \mu \): Viscosity, Pa s
\( \rho \): Density, kg/m\(^3\)

SUBSCRIPT
- \( aw \): Adiabatic wall
- \( ave \): Average in the spanwise direction of the model
- \( d \): Dynamic quantity
- \( f \): Fluid
- \( L \): Leading edge surface
- \( \text{ref} \): Reference value
- \( t \): Stagnation
- \( w \): Wall
- \( \infty \): Relative to mainstream
- 2: Relative to secondary air

1. INTRODUCTION
In order to raise thermal efficiency of gas turbine, higher turbine inlet temperature (TIT) is needed. However, higher TIT increases thermal load to its hot-section components, reducing their life span. Therefore, very complicated cooling technology such as film cooling and internal cooling is required especially for HP turbine blades. In film cooling, relatively cool air is injected onto the blade surface to form a protective layer between the surface and hot mainstream gas. Since the highest thermal load usually occurs at the leading edge of the blade and damage is likely to happen in this region, film cooling is typically applied to the leading edge through an array of hole rows called showerhead cooling holes. The flow field near the
leading edge is extremely complicated with stagnation, strong pressure gradients and curvature, and interaction between mainstream and coolant jets becomes increasingly complex, eventually making its proper cooling difficult. Therefore, increases around film coverage in the leading edge will lead to significant benefits to the turbine blade life.

Mehendale and Han [2] used a blunt body with a semi-circular leading edge and a flat afterbody to study the effect of high mainstream turbulence on leading edge film cooling and heat transfer. Two turbulence level (Tu = 9.67% and Tu = 12.9%) were generated by passive grid and a jet grid at a leading edge Reynolds number of 100,000. The cooling air was injected through two rows of film holes at ±15 and ±40 degrees from the stagnation with three blowing ratios of 0.4, 0.8, and 1.2. They found that the leading edge film effectiveness for a blowing ratio of 0.4 was significantly reduced by high mainstream turbulence. For blowing ratio of 0.8 and 1.2 the mainstream turbulence effect was diminished in the leading edge but still existed on the flat sidewall region. They also pointed out that the leading edge heat transfer coefficient for blowing ratio of 0.8 increased with mainstream turbulence, but the effect was not consistent for blowing ratio of 0.4 and 1.2.

Ekkad et al. [3] studied the effect of free stream turbulence on the detailed distributions of film effectiveness and heat transfer coefficient on a cylindrical leading edge model using transient liquid crystal image method. Their results also show that higher mainstream turbulence reduces the film effectiveness for lower blowing ratios but the effect diminishes at higher blowing ratio. Rozati et al. [4-5] and Takahashi et al. [6] studied leading edge film cooling by numerical method with Large Eddy Simulation (LES) and Detached-Eddy Simulation (DES). Although they compared the numerical aerothermal performance with the experiment results, the effect of free stream turbulence was not taken into account in their simulations.

Saumweber et al. [7] studied free stream turbulence effect on flat plate film cooling with shaped holes. They found that the effect of increased turbulence level was detrimental to film effectiveness of the shaped hole at all blowing ratios. Laterally averaged film effectiveness was reduced up by 30% when the turbulence intensity was increased from 3.6 to 11%. The effect was more pronounced at smaller blowing ratios.

Kim et al.[8] and Reiss et al.[9] studied the influence of shaped injection holes on leading edge film cooling. They found that the holes with laid-back-type widened exits clearly enhanced the overall cooling performance of the showerhead, compared to classical cylindrical hole cases. However, the effect of free stream turbulence was not investigated in these studies.

York et al. [10] used a computational methodology for the analysis of film cooling from diffused holes on the simulated leading edge of a turbine airfoil. Their results show that the advantage in effectiveness was due to the shallower trajectory of the coolant exiting the holes, causing it to stay closer to the surface than in the case of nondiffused holes. But, there were few examples which studied the effect of cooling hole exit shape and mainstream turbulence using any leading edge model.

In this investigation, the influences of hole shape (cylindrical and diffuser) as well as free stream turbulence on leading edge film cooling are studied. The objective of the present study is to clarify their cooling performances as well as aerodynamic behavior in their of total pressure loss under several free stream turbulence conditions. Test cases with three turbulence levels and four blowing ratios were examined.

2. EXPERIMENT
2.1 Experimental Facility

Figure 1 shows the test apparatus. The experiment was conducted using the wind tunnel for heat transfer test at Iwate University. Two air supply systems exist in the experimental facility, and the secondary air is heated. The secondary air was then supplied to the test model installed in the center of the test section duct via the upper piping as shown in figure 1. The mainstream velocity was measured by a Pitot tube installed 210mm upstream of the stagnation point of the test model. The test section duct was built from acrylics plates, with cross-section areas of test section 280mm × 450mm and length 1150mm (360mm from the stagnation point of test model to a test section duct entrance). The static pressure holes was prepared by both side of the test model on the lower plate of the test duct to check the symmetric flow around the model with respect to the duct center line. In this study, the difference of static pressure on the both side was adjusted so as to be less than 1Pa. Two types of turbulence grid (Grid A, Grid B) were employed in this study. Each of them was attached to the inlet of the transition duct. The distance from the stagnation point of the test model to the turbulence grid was 750mm. Grid A consisted of stainless steel pipe of 8mm in diameter. Grid B was composed of square wooden of bars(12mm × 12mm). When measuring the mainstream turbulence intensity, I-type hot wire anemometer was used along with DANTEC Stream Line System. A SONY DCR-VX2000 CCD camcorder used for
recording TLC painted test model surface was installed in the angle of 60° from the stagnation point of the test model.

In this study, the test model for static pressure measurement and two types of test model with film cooling holes were prepared. Figure 2(a) shows the test model for static pressure measurement. Each of the test models having the same dimensions was made from a photocuring resin, semi-circular leading edge of 80mm in diameter (D), and flat plate parts of 100mm in length and the height of the test model 280mm. In the heat transfer measurement, only one side of the test model was monitored, including a half part of the semi-circular part and downstream flat-plate part of 9.36d (=75mm) in length. When installing the test model in test section, flat plate region was extended by attaching 650mm acrylics plates. The holes for measuring static pressure were prepared along the mid-span of the test model, while any cooling holes were not provided on this test model as shown on the top of Figure 2. Each of the static pressure holes was equally spaced by 10 degrees in the semi-circular region. Static pressure holes were also created on the flat plate region 8mm downstream of the junction between the semi-circular part and flat plate part.

Figure 2(b) shows two types of test models that have film cooling holes for heat transfer measurement. The shapes of film cooling hole exits were cylindrical and diffuser-shaped. The equivalent diameter of the cooling hole (d) was 8mm, and the diffuser cooling holes were expanded toward hole exit. The cooling hole was created in the position of $\alpha = \pm 25^\circ$ and $\alpha = \pm 55^\circ$ at each test model. However, only the region of $\alpha \geq 0^\circ$ was captured by the camera in heat transfer measurement. The angle of inclination of cooling hole to the surface was $40^\circ$. A partition plate for preventing secondary air from leaking out was installed in the back section of the plenum. Four holes were drilled in the partition plate so that it was possible to measure the temperature in the plenum in the height direction at four places. Furthermore, the temperature in the cooling hole of mid-span was measured by thermocouples (see the middle of Figure 2).

2.2 Test Conditions

All tests were conducted in the wind tunnel at Reynolds number of 43,000 based on leading edge diameter (D) and density ratio is 0.92. Blowing ratio $BR = \frac{\rho_m U_c}{\rho_c u}$ was examined $BR = 1.0$. This Blowing ratio is averaged value of all cooling holes. Figure 3 shows streamwise distributions of turbulence intensity measured at the mid span with and without turbulence grid. Each of the minimum turbulence intensities is referred to as the corresponding reference turbulence intensity in this study. Therefore, the reference turbulence intensity for the No Grid, Grid A and Grid B are 1.0, 3.3 and 12.0 %, respectively.

In this study, since only the half of the test model was observed by the CCD camera in the heat transfer measurement, it was necessary to check the symmetry of the flow in the test section. Therefore, the test model for static pressure measurement where, total pressure $P_t$ and dynamic pressure $P_d$ shown in Fig. 2(a) model for pressure measurement and (b)model for heat transfer measurement were obtained a by Pitot tube positioned 210mm upstream of the stagnation point. Since wakes generated by the grid was expected to hit the Pitot tube, it position was changed to three different places to obtain averaged values. The values of $C_p$ at the test model in each test condition are shown in Figure 4. $C_p$ distribution is almost symmetrical with respect to the stagnation line ($\alpha = 0^\circ$), indicating that the stagnation point did not shift even under the influence of free-stream turbulence. Although the separation appeared at $\alpha = 90^\circ$ in No Grid and Grid A case, it was no longer observed in Grid B case.
2.2 Aerodynamic investigations

In this study, aerodynamic loss is estimated by calculating the total pressure loss coefficient defined by Eq.(2)

\[ C_{p,t} = \frac{P_{t,ref} - P_{t,\infty}}{\frac{1}{2} \rho U_{\infty}^2} \]  

\( P_{t,\text{ref}} \) is mass averaged Eq.(3) where

\[ P_{t,\text{ref}} = \frac{\dot{m}_2}{\dot{m}_{\infty}} P_{t,\infty} + \frac{\dot{m}_{\infty}}{\dot{m}_{\infty}} P_{t,\infty} \]  

Figure 5(a) shows a total pressure probe made for near-wall pressure measurement. Location of measurement plane is \( x/d=7D(=560\text{mm}) \). The probe head geometry is \( 1.5\text{mm} \times 0.5\text{mm} \). Figure 6(b) shows the measurement grid. 50 points total pressure measurement was performed in the direction of normal to model surface and span at 1mm intervals and 5mm intervals, respectively.

2.3 Thermal Investigations

2.3.1 Theory of transient TLC technique

In this study the test model was coated with TLC (Nihon microcapsule). The nominal color bands of TLC was from \( 22.5^\circ \text{C} \) to \( 30.5^\circ \text{C} \).

The present study used two different reference temperatures to determine film effectiveness and heat transfer coefficients both from a single test in a way proposed by Kim et al. [11], which will be briefly described in the following.

When a semi-infinite substance of initial temperature \( T_i \) is exposed to a flow whose temperature \( T(t) \) starts to increase at a certain instant, its surface temperature \( T_w(t) \) accordingly rises. Suppose that heat transfer coefficient of the flow \( h \) is constant, \( T_w(t) \) can be expressed by Eqs. (4) and (5) using Duhamel’s theorem,

\[ T_w(t) - T_i = \sum_{j=1}^{N} U(t - \tau_j)(T_j - T_{j-1}), \]  

\[ U(t - \tau_j) = 1 - \exp(\beta^2)\text{erfc}(\beta), \quad \beta = \frac{h_{j}(t - \tau_j)}{\sqrt{\rho c \kappa}} \]  

where the increase in the flow temperature is approximated by a summation of small temperature steps \((T_j - T_{j-1})\) with the time lag from the initiation \( \tau_j \), and \( U(t - \tau_j) \) in Eq. (5) is an exact solution of the equation for the one-dimensional unsteady heat conduction under the abrupt increase in the flow temperature. Eq.(4) can yield the heat transfer coefficient \( h \) using the information on the temporal variation of the surface temperature as well as the temperature rise of the flow over the surface.

When a film cooling exists, its effect upon the flow temperature should be taken into account through the film cooling effectiveness \( \eta \), which is defined as follows:

\[ \eta = \frac{T_{w,\infty} - T_{w}}{T_2 - T_{\infty}} \]  

Figure 3 Streamwise distribution of turbulence intensity

Figure 4 Static pressure distributions under several turbulence intensities

Figure 5 Measurement probe and measurement points in each plane
where $T_{aw}$, $T_\infty$, and $T_2$ are adiabatic wall temperature, primary flow temperature and secondary flow temperature, respectively. Using this relationship, along with the assumption that $\eta$ is constant even when the secondary flow temperature varies with time, the temperature $T_j$ in Eq. (4) can be replaced by the corresponding adiabatic wall temperature $T_{aw,j}$ given by

$$T_{aw,j} = \eta T_{2,j} + (1-\eta)T_\infty.$$

(7)

From this expression the following expression is obtained.

$$T_{aw,j} - T_{aw,j-1} = \eta(T_{2,j} - T_{2,j-1})$$

(8)

Therefore, replacing $(T_j-T_{aw,j})$ in Eq. (4) by $(T_{aw,j} - T_{aw,j-1})$, one can obtain the expression for the surface temperature,

$$T_a(t) - T_i = \eta \sum_{j=1}^N U(t-\tau_j)(T_{2,j} - T_{2,j-1}).$$

(9)

Use of the above expressions for different two instants $t=t_b$ and $t=t_b$ to eliminate $\eta$ yields the following equation,

$$\frac{T_a(t_a) - T_i}{T_a(t_b) - T_i} = \frac{\sum_{j=1}^N U(t_a-\tau_j)(T_{2,j} - T_{2,j-1})}{\sum_{j=1}^N U(t_b-\tau_j)(T_{2,j} - T_{2,j-1})}.$$ 

(10)

Then the heat transfer coefficient $h$ can be determined from Eq. (10), using a proper method for solving non-linear equations. Substituting the resultant heat transfer coefficient into Eq. (9), film effectiveness is then calculated as follows,

$$\eta = \frac{T_a(t_a) - T_i}{\sum_{j=1}^N U(t_a-\tau_j)(T_{2,j} - T_{2,j-1})}.$$ 

(11)

2.3.2 Temperature measurement around the model

In this study, a thermocouple rake was used to perform temperature measurement on each of the measurement plane, shown in Figure 6(a). The thermocouple rake consists of 13 K-type thermocouples. These thermocouples were installed at 5mm pitch. Planes are $\alpha = 40^\circ$, $70^\circ$, $90^\circ$ and $x/d=9.36d$ (Flat plate position). Figure 6(b) shows the measurement grid. 50 points temperature measurement was performed at 1mm intervals in the direction of normal to the model surface. Measurement planes were located in mid-span of the test model as shown Figure 6(a). The non-dimension temperature was similarly defined as film effectiveness, and it was calculated by following Eq. (12).

$$\eta = \frac{T_{j} - T_{\infty}}{T_{2} - T_{\infty}},$$

(12)

where $T_2$ was the mean temperature of two thermocouples in the inside of cooling holes.

2.4 Uncertainty Analysis

Major factors that could contribute to the uncertainty of the heat transfer characteristics the transient TLC method were errors in wall temperature in using TLC, primary and secondary temperature measurements using thermocouples. The calibration revealed that the uncertainty of wall temperature given by TLC was about ±0.5 °C for the film-cooled surface. According to the conventional uncertainty analysis [11], it follows that the uncertainty in film cooling heat transfer coefficient was about ±10.5%. Due to simultaneous calculation of film effectiveness from Eq.(11), the film effectiveness uncertainty is estimated to be almost the same as the film cooling heat transfer coefficient. In aerodynamic investigation, the accuracy of the pressure transducer was ±0.25%. The uncertainty of the total pressure loss coefficient defined by Eq.(2) was about ±1.5%.

3. NUMERICAL SIMULATION

A commercial software, ANSYS CFX 12 was used in this study. Time-averaged Reynolds-Averaged Navier-Stokes (RANS) approach using Shear-Stress Transport (SST) two-equation model and Detached Eddy Simulation (DES) approach [12] were employed. Figure 7 shows the computational domain simulating the experimental setups and the mesh. Assuming the flow symmetry, the domain was restricted to the left half of the flow field around the test model. The height of the domain was 14d, which was wide enough to cover the test model by 2 pitches. Although tetra meshes were mainly used for the computational grid, prism meshes were used to resolve boundary layer at near wall region. As a result of mesh dependency test, total number of the cells was about 11 million in this domain, where 9 million were used for the test section duct region, and 2 million were used for plenum and film holes region. The value of $y+$ for the computational point of the first cell above the wall was less than unity so that the wall function approach was not applied on the wall. The mainstream flow velocity measured in the experiment, temperature and turbulence intensity were specified at the mainstream entrance. The mass flow rate and temperature measured in the experiment were imposed at the secondary air entrance. The boundary condition of symmetry was used for the center of the computational domain. As for DES case, the non-dimensional time step was $4.05 \times 10^{-4} D/U_\infty$. Time-averaged statistics were calculated using accumulated data over the period of about $6D/U_\infty$.
4. RESULT AND DISCUSSION

4.1 Total pressure loss coefficient

Figure 8 shows distributions of the total pressure loss coefficient obtained in the experiment. As for cylindrical hole with no grid case, even if injection of secondary air existed, the total pressure loss coefficient distribution hardly changed in BR=1.0 case compared with the diffuser hole case. In Grid C case, the total pressure loss coefficient became higher in the whole region by the influence of the turbulence grid. In this case, the high region of the total pressure loss near the test section became high. Note that, since the heat transfer coefficient, the heat transfer coefficient near the model surface relatively became thinner.

In order to get averaged quantities of performance, mass averaged total pressure loss coefficient $\bar{\zeta}$ was calculated by Eq. (13).

$$\bar{\zeta} = \frac{\iiint C_{P,t} V dA}{\iiint V dA}$$  \hspace{1cm} (13)

Figure 9 shows the mass averaged total pressure loss coefficient. For two hole exit geometries, the loss coefficients with and without freestream turbulence almost matched with each other within the limit of the error bar.

4.2 Film cooling effectiveness and heat transfer coefficient

Detailed experimental results of film cooling effectiveness and heat transfer coefficient distributions for cylindrical hole case are presented in Figure 10(a). Since the liquid crystal coating without the influence of the injected air did not change its color, the temperature data was not obtained in the black region in this figure. In cylindrical hole case, the film effectiveness distribution found from $\alpha = 55^\circ$ cooling hole expanded in the spanwise direction compared with film effectiveness distribution from $\alpha = 25^\circ$ cooling hole. The difference was due to static pressure distribution around the leading edge model and different local blowing ratio of each film cooling hole. At $\alpha = 90^\circ$, the film effectiveness distribution expanded in the spanwise direction because the mainstream separated. When the mainstream turbulence intensity became high, any film effectiveness distribution expansion was not clearly observed in the downstream region from $\alpha = 90^\circ$. Especially this tendency was remarkably observed in the film effectiveness distribution coming from the $\alpha = 25^\circ$ cooling hole. On the other hand, in semi-circular region, the film effectiveness reduction was hardly seen. In the case of Tu=12.0%, reduction of film effectiveness was observed by mainstream turbulence, but the secondary air which ejected from $\alpha = 25^\circ$ cooling holes interfered with the secondary air ejecting from $\alpha = 55^\circ$ cooling holes, and the film effectiveness distribution expanded in the spanwise direction so that the film coverage increased. As for heat transfer coefficient, the heat transfer coefficient near the cooling holes became high. Note that, since the heat transfer coefficient was measured only in the region where TLC color
changes, the experimental data cannot be completely compared with numerical results. Detailed experimental results of film cooling effectiveness and heat transfer coefficient distributions for diffuser hole case are presented in Figure 10(b). Compared with cylindrical hole case, the film effectiveness distribution expansion was larger in diffuser hole case. This can be explained by the effect of reduced momentum of the secondary air from the diffuser hole, which made the air attached to the test model surface. When the mainstream turbulence intensity increased, the expansion of the film effectiveness was no longer observed cylindrical hole case. In addition reduction of the film effectiveness in distribution from $\alpha = 55^\circ$ cooling holes was seen remarkably. Although the heat transfer coefficient was high near the cooling hole as in cylindrical hole case, the heat transfer coefficient near the diffuser cooling holes was lower than cylindrical hole case.

Figure 11(a) shows the film effectiveness distribution in cylindrical hole case obtained by RANS CFD. In the CFD result, the film effectiveness near the cooling holes was higher than the experimental result. In the case of BR=1.0 No Grid, the CFD result was qualitatively similar to the experimental result in the semi-circular region, however in the flat plate region the film effectiveness distribution expansion was overpredicted in comparison with the experimental result. Even when the turbulence intensity became high, the CFD result hardly changed. Figure 11(b) shows the film effectiveness distribution of diffuser hole case obtained by RANS CFD. In diffuser hole case, the film effectiveness near the cooling holes was also higher than the experimental result. However, as compared with cylindrical hole case, the region with high film effectiveness became smaller. The reduction of film effectiveness by the increase in turbulence intensity was not seen by the diffuser hole case.

Figure 12 shows the time-averaged film effectiveness distribution of cylindrical and diffuser hole cases obtained by DES. If follows from the comparison with figure 11 that in DES case, the portion with high film cooling effectiveness near the cooling holes decreased. Furthermore, the film effectiveness in the plate region also decreased. In theory DES can resolve small vortex structure than RANS CFD, which could be a one reason of the reduction in film effectiveness. On the other hand, as compared with the experimental result (Figure 10), the film effectiveness near the cooling holes is still high. Moreover, expansion of the film effectiveness in the flat plate region was larger than the experimental result.
Figure 13 shows the instantaneous vortex structure, colored by the local nondimensional temperature. These vortex structures are represented by iso-surfaces of second invariant of gradient of velocity tensor $Q$. The nondimensional temperature is evaluated by the same definition as the film cooling effectiveness $\eta$. In cylindrical hole case, vortex structures resolved by DES clearly differ from the RANS CFD result. The DES predicts vortex structures representing anisotropic motions. Similarly, in diffuser hole case, vortex structures with anisotropic motion of horseshoe-like form are predicted by DES. In contrast to the cylindrical case, the DES prediction reveals that the cooling air from the diffuser holes appear to branch into two parts.

Figure 14 shows instantaneous iso-surface of nondimensional temperature $\eta = 0.4$. In the case of cylindrical hole, the DES indicates the fluctuations of the nondimensional temperature. However, the RANS CFD exhibited rather smooth iso-surfaces, failing to reproduce temperature fluctuation between the cooling air and the main flow. These are supposed to be due to the fact that the DES can evaluates more accurately the mixing process by the unsteady vortex structures. In diffuser hole case, the DES predicts the fluctuation of the local temperature field like cylindrical hole case. The iso-surface of nondimensional temperature from $\alpha = 55^\circ$ cooling holes is clearly branched.

Figure 15(a) shows a comparison of spanwise averaged film effectiveness. In the case of cylindrical hole, when mainstream turbulence intensity was low (No Grid and Grid A), the spanwise averaged film effectiveness hardly changed. On the other hand, since film effectiveness distribution expanded in the spanwise direction, the spanwise averaged film effectiveness was slightly higher under Grid B case. The reduction of film effectiveness appeared faster than in the case of the low mainstream turbulence intensity. Although the RANS CFD result was qualitatively in agreement with the experimental result, there was a slight difference due to the effect of mainstream turbulence and RANS CFD tended to overpredict. Since unsteady vortex structures was not predicted by RANS CFD, it can be mentioned that the reduction of spanwise averaged film effectiveness was hardly seen. As for the DES result, the film effectiveness decreased and it became closer to the experimental result rather than the RANS CFD. This is because the diffusion behaviour of the temperature is adequately predicted in DES case. However, in the flat plate region, the reduction of film effectiveness become slow and the difference with the experimental result is still large. From cylindrical hole case result, it is thought that DES CFD reproduced the experimental findings much better than RANS CFD.

Figure 15(b) shows a comparison of spanwise averaged film effectiveness in diffuser hole case. Unlike the case of cylindrical hole, the experimental result of diffuser hole case was decreased as the turbulence intensity became high especially in the downstream region from $\alpha = 90^\circ$. However the change in RANS CFD result due to the increasing of
mainstream turbulence intensity was hardly seen in contrast to the cylindrical hole case. The RANS CFD result of No Grid case was quantitatively close to the experimental result, but when the mainstream turbulence intensity became high, the effect of mainstream turbulence intensity was clearly identified compared with the experimental result. As for DES result, although the anisotropic vortex structure was predicted, the spanwise-averaged film effectiveness was similar to the RANS CFD result.

4.3 Local temperature on normal planes to model surface

Figure 16 shows the distributions of measured and predicted local temperature on the transverse planes normal to the model surface in cylindrical hole case. In this figure, the range of $\eta$ is from 0 to 0.6, which is to emphasize the variation of the non-dimensional coefficient. Since the secondary air had a downward component of velocity, it gradually moved downward. In this case, the experimental results indicate that the mainstream and secondary air were mixed and the non-dimensional temperature of Grid B case decreased compared with the case of No Grid. However, the
influence of mainstream turbulence intensity was hardly seen in the process of the secondary air attachment to the surface. The non-dimensional temperature of the secondary air from $\alpha =25^\circ$ cooling holes was decreased, while the diffusion to the spanwise direction was slightly large. In No Grid case, the CFD result was qualitatively in agreement with the experimental result. However, the CFD result became higher than the experimental result. Furthermore, the reduction of the non-dimensional temperature according to the increase of mainstream turbulence intensity was not observed in the CFD result. As for DES result, the shape of secondary air bulk was different from RANS result. As mention above DES predicted the fluctuation of secondary air bulk, resulting in the reduction of film effectiveness. However the reduction of non-dimensional temperature from $\alpha =25^\circ$ cooling holes was not observed.

Figure 16(b) shows the distributions of measured and predicted local temperature on the transverse planes normal to the model surface in diffuser hole case. As for the experimental result, compared with the cylindrical hole case, the secondary air was attached to the model surface. When the mainstream turbulence intensity became high, the reduction of non-dimensional temperature was observed as in the result of cylindrical hole case. As for CFD result in BR=1.0 case, especially the bulk of secondary air from $\alpha =25^\circ$ cooling hole differed from the experimental result. Reduction of the non-dimensional temperature by the increase of mainstream turbulence intensity was hardly identified as in the CFD result of cylindrical hole case. In the DES case, although the complicated vortex structure was predicted, the secondary air bulk was hardly changed compared with the RANS result. Therefore, it is expected that the vortex structure generated from the diffuser hole did not affect the secondary air bulk drastically, and the spanwise-averaged film cooling effectiveness remained almost unchanged.

5. CONCLUSIONS

The influence of mainstream turbulence on leading edge film cooling was experimentally and numerically studied with two type of cooling holes. The main findings of the study are as follows.

In cylindrical hole case, under low BR, mainstream turbulence acted on the effective mixture of mainstream and secondary air and the diffusivity of the secondary air from $\alpha =25^\circ$ cooling holes was improved. However, it promoted the reduction of secondary air temperature. As a result, the spanwise averaged film effectiveness was decreased as the turbulence intensity became high.

In diffuse hole case, since the momentum in the cooling hole exit was low compared with that of cylindrical hole, the secondary air attached to the model surface and its diffusion over the surface became prominent. Also in the case of diffuser hole, secondary air temperature on normal planes to model surfaces was decreased as the turbulence intensity became high. In this case, the mainstream turbulence enhanced mixing of mainstream and secondary air in the streamwise and spanwise direction, reducing the spanwise averaged film effectiveness significantly.

For both hole geometry cases the DES result were reasonably in agreement with the experimental results in semi-circular region, which was probably because DES is able to predict the complicated vortex structure. Furthermore, the prediction accuracy of the film cooling was better for cylindrical hole case. However the discrepancy between the experimental result and DES result was clearly identified in flat plate region. As for RANS CFD result, the effect of mainstream turbulence was hardly seen in RANS CFD result. This is because it was not possible to predict instability-originated unsteady flow structure by RANS CFD.

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