1 Introduction

Application of film cooling technology to gas turbines progresses in combined power generation plants increasing thermal efficiency with rise in turbine inlet gas temperature. In order to assess the durability of the gas turbine blades, it is essential to estimate the temperature distributions of the blades. Accordingly, the prediction of film cooling by CFD is important for the assessment of the durability. It is necessary for the durability assessment to estimate the cooling performance, especially on the leading edge region, where flow impinges and heavy thermal load appears. Hence, the present study focuses on the CFD prediction of the leading edge film cooling performance.

Experimental investigations of the leading edge film cooling have been previously done and clarified a lot of important characteristics of the performances. Mick and Mayle [1] measured film cooling effectiveness and heat transfer on the semicircular leading edge with three rows of cooling holes. They presented that the effectiveness is decreased as the blowing ratio is increased. Mehdendale and Han [2] studied the influences of the blowing ratio and the inlet turbulence intensity by using the same model as that of Mick and Mayle [1]. Furthermore, Ou et al. [3], Eckad et al. [4], and Johnston et al. [5] investigated the influences of inlet turbulence on the cooling performance. Funazaki et al. [6] studied influences of the incoming periodic turbulence by wakes on the film cooling of the semicircular leading edge model. Cruse et al. [7] studied the effect of leading edge shapes on the film cooling effectiveness by using an infrared camera with circular and elliptic leading edge geometries. Reiss and Böcs [8] examined the influence of the cooling hole geometry on film cooling performance. The authors presented fan-shaped holes’ increased film cooling performance due to the reduced tendency of “lift off” at high blowing ratios. Ou and Rivir [9] employed liquid crystal image to measure the film cooling heat transfer. Ahn et al. [10] measured the effectiveness on a rotating blade leading edge by using pressure sensitive paints. More recently, Lu et al. [11] conducted a detailed investigation of the hole angle and shape effects by using transient infrared thermography techniques. Falcoz et al. [12] presented detailed measurements of the film cooling effectiveness with thermochromatic liquid crystal.

With regard to the numerical approaches for film cooling, starting with flat plate configurations, a number of the investigations (e.g., Refs. [13–24] for flat plate configurations and Refs. [25–36] for leading edge configurations) have been conducted in order to predict the cooling performance and to clarify the physics of the
thermal flow field, so far. In terms of the leading edge film cooling, the CFD studies also presented useful knowledge. The United Technology Pratt and Whitney organized the blind test of film cooling on cylindrical leading edge experiments of Cruse et al. [7]. On this blind test, Lin et al. [25], Martin and Thole [26], and Thakur et al. [27] calculated the same thermoflow field by applying different mesh types with discretization schemes and turbulence modeling including a k-ω model and a k-ε model with wall functions. Chernobrakin and Lakshminarayana [28] also ran simulations for the same flow by a low Reynolds number k-ε model, later. All those computations correctly predicted the tendencies of lateral-averaged effectiveness but included discrepancies in local effectiveness. Lin et al. [25] and their extended work [29] with the SST model clarified how hot spots can form on the stagnation region due to the flow induced by coolant jets. On the other hand, they also indicated that the simulations underpredicted the spread of the coolant in the wall-normal direction. York and Leylek [30–32] applied the realizable k-ε turbulence model into the leading edge film cooling in order to investigate the film cooling effectiveness, heat transfer coefficient and the effects of hole shapes, respectively. The authors showed that the predictions are generally in good agreement with the experiments but discrepancies increases with the blowing ratio. All the numerical studies for the leading edge cooling mentioned above had been conducted based on the steady RANS modeling. Those outcomes demonstrate that CFD is able to serve useful information for the estimation of the leading edge film cooling.

However, it is fundamentally difficult in principle for the eddy viscosity models in the steady RANS simulations to correctly resolve anisotropic behavior induced by large scale structures that are probably the primal mechanisms of the mixing of the film coolant with the free stream. A large-eddy simulation (LES) and, of course, DNS are essential to finely resolve turbulence scales. Due to this, the LES has been applied to the calculation of the film cooling. Tyagi and Acharya [23] and Guo et al. [24] for the flat plate configuration and Rozati and Tafti [34,35] and Sreedharan and Tafti [36] for the leading edge configuration. Those studies have clarified the detailed physics of the film cooling flow. On the other hand, it is supposed to be difficult for the near future environment of high-performance computing, at least for industrial simulations, to estimate the convective heat transfer of cooled blades in a hot section (e.g., HPT blades) by the LES. Those convective heat transfer calculations need a considerably fine mesh similar to the DNS to resolve turbulence scales to be close to the near wall region. High speed computing and massive resources required make it difficult to apply the full LES to real configurations and conditions of actual cooled blades.

On the above background, this study investigates performances of unsteady CFD calculations by applying RANS modeling to the leading edge film cooling. Taking practicable simulations for industrial problems in the near future into consideration, calculations of unsteady RANS (URANS) and a detached-eddy simulation (DES) are conducted from a view of time-averaged estimation of the cooling performance, which is firstly important for the durability assessment of the cooled blade with a large matrix of parameters. The URANS employing the k-ε-ω-LES turbulence model [37,38] and the Spalart and Allmaras turbulence model [39], and the DES [40], which is a LES with the Spalart and Allmaras turbulence model in the near wall region, are addressed to solve thermal convection. In the unsteady calculations, RANS modeling was possibly expected to work as a filter to extract structures having impacts on the cooling performance. The present authors previously calculated the film cooling mainly for flat plate configurations with steady modeling using the k-ω type turbulence model (chiefly by the SST model) and the DES [33]. The authors presented that the DES could predict secondary vortices counter-rotating against the primary vortex pair in the flat plate cooling, which is supposed to enhance the lateral spreading of the film coolant. The authors also tried to calculate the leading edge film cooling, in addition, but acquired little information for the CFD capability for the leading edge film cooling. The present calculations were conducted by simulating an experiment of the semicircular leading edge model. Distributions of film cooling effectiveness on the model surface and temperatures distributions in the flow field are compared with the measurements with a thermoprobe consisting of thermocouples and thermochromatic liquid crystal. Thermoflow characteristics predicted by the present unsteady calculations were investigated.

2 Leading Edge Model

Figure 1 shows the leading edge model used in the numerical calculations and experimental measurements. The leading edge model consists of a semicircular geometry of 80 mm diameter and a flat after-body. The model has plenum inside to supply the secondary air as film coolant through film cooling holes. On the wall of the model, the film cooling holes of 8 mm diameter are periodically placed with staggered arrangement in the three rows. Hole angle is 30 deg to the model surface in the heightwise direction, and the pitch of the periodic holes is 62 mm (7.72d). Two rows of the film cooling holes, hole 1 (row 1) and hole 2 (row 2), are located at 15 deg on both sides from the stagnation line in the case without film cooling. The other row of the film cooling hole, hole 3 (row 3), is placed at 50 deg from the stagnation. As shown
in Fig. 2(a), the computational domain simulating the experimental setups is composed of the primary domain, the secondary chamber, and the connecting duct between the primary domain and the chamber. The primary domain is a pitch of the periodical arrangement of the film cooling holes in the heightwise direction, consisting of the main stream region, the film cooling holes (hole 1, hole 2, and hole 3), and the plenum, as shown in Fig. 2(b).

### 3 Descriptions of Numerical Calculation

The "FLUENT" code based on the finite volume method was used in the numerical calculations. Unsteady Reynolds-averaged and filtered Navier–Stokes equations were solved with the URANS models and the DES, respectively. The discretized equations are solved by the SIMPLE algorithm with time integration. The second order implicit scheme is utilized for the time integral of both the URANS and the DES with a nondimensional time step of $4.05 \times 10^{-4} D/U_{in}$. Time-averaged statistics were accumulated over the period of about $8D/U_{in}$.

#### 3.1 URANS

Unsteady Reynolds-averaged Navier–Stokes equations were solved with the $k-\varepsilon$ model [37,38] (V2F) and the Spalart and Allmaras model [39] (SA). The V2F was employed to attempt to correctly treat wall asymptotic behaviors of the turbulent properties and to avoid production of anomalous turbulent energy in the flow around the leading edge by a two-equation eddy viscosity model. Regarding this point for the SA, the turbulence production is evaluated based on the antisymmetric components of the velocity gradient tensor (the vorticity tensor). The convective terms of the momentum equations were discretized by the third order MUSCL scheme [42] and the other terms are solved with the second order central difference scheme.

#### 3.2 DES

The DES modeling based on the Spalart–Allmaras model [39] was applied. The DES model replaces the length scale $d_w$ in the SA, which is the distance to the closest wall, with the length scale defined as

$$l = \min(d_w, C_{des}\Delta)$$

where $\Delta$ is based on the largest grid space in the $x$, $y$, or $z$ directions forming the computational cell, and $C_{des} = 0.65$ is the empirical constant. The second order central difference scheme is applied into all the spatial discretizations of the momentum equations.

#### 3.3 Computational Mesh and Conditions

Figure 3 shows the outline of the computational mesh (surface mesh) employed in the calculations. Unstructured grid system, which consists of hexahedral cells and tetra cells, was employed to solve the discretized equations. Except in the plenum and the secondary air chamber, the hexahedral cells are generated in the main stream region and the film cooling holes.

It is supposed to be some challenge to search grid convergence for the DES switching the modeling from the LES to the SA close to the wall, as indicated by Eq. (1). The present study, however, just focused on the leading edge portion with a very thin boundary layer in the whole domain, near wall mesh is fined by using hang-
ing nodes in order to secure resolution. The total number of the cells is 3,995,322, 1,898,260 cells in the main stream region around the leading edge model, 1,783,376 cells in the film cooling holes, 271,830 cells in the plenum, and 41,856 cells in the secondary air chamber. The cells on the model surface are formed layered-structure in the near wall regions. The value of $y^+$ for the computational point of the first cell above the wall is less than unity so that the wall function approach is not applied on the wall. For a reference of mesh dependency in the solutions, Fig. 4 shows comparisons of spanwise-averaged film cooling effectiveness on the model between the DES calculations with the primary mesh mentioned above and a coarse mesh. The computational conditions will be shown later. The total cell number of the coarse mesh is $1,422,738$, $587,748$ cells in the main stream region, $521,304$ cells in the film cooling holes, and $313,686$ cells in the other regions. The coarse mesh results show only a small difference from the primary mesh ones. In the present study, the primary mesh had been employed for all the calculations of the DES and URANS.

Table 1 shows inlet conditions for the numerical calculations. In this study, the temperature of the second air was set at higher than the main flow temperature, according to experimental conditions, as presented below. All the calculations were performed for Reynolds number $Re_D$ of $8.55 \times 10^4$, based on the model diameter $D$ and inlet flow conditions. Two cases for the mean blowing ratio ($BR \sim 1.2$) of the three holes, hole 1, hole 2, and hole 3, are calculated for each of the turbulence model. Adiabatic conditions were imposed on the wall of the model. Mass flow rate, total temperature and turbulent quantities were imposed at the inlets of the main stream region and the secondary air chamber, and the static pressure is set at both the outlets of the main stream region and the plenum. The static pressure at the plenum outlet was specified to balance a mean mass flow in film cooling holes, which is determined by the prescribed mean blowing ratio $BR$. With regard to the inlet turbulence quantities in each turbulence model, steady component is set on the basis of the turbulence intensity of $0.1\%$, referred to the wind tunnel experiment. Unsteady components of the flow primitives are not given at the inlets in the calculations.

All the computations were executed with a PC cluster of eight CPUs. The process time per CPU for the DES and the SA is approximately $5$ s per iteration in the SIMPLE algorithm, and the process for the V2F took about $20\%$ longer than for the SA because of the number of the equations.

4 Experimental Measurements

Local temperatures on and around the leading edge model, which is installed in a test section of a low speed wind tunnel, were measured with thermocouples and thermochromatic liquid crystal applied on the model surface. Turbulence intensity of the main stream into the test section of the wind tunnel is less than $0.5\%$.

Locations of the temperature measurement by the thermocouples are shown with dotted lines in Fig. 5. The temperature distributions on four traverse planes normal to the model surface, shown in Fig. 5(a), were measured by a probe of a K-type thermocouple. On each of the plane of $5d \times 7.72d$ (a pitch of the periodic hole arrangement), temperatures are measured on $17 \times 9$ crossing points of the grid lines shown in Fig. 5(b) by traversing the probe. Local temperature profile of very near wall fields is also measured on the half surface of the model at the same side as the above flow field measurements, as a surface temperature by using the thermocouple probe. Wall temperature profile on the whole surface of the model was also measured by the transient method using thermochromatic liquid crystal with two video cameras set at each side of the model, which is referred to in Ref. [33]. Those temperatures measured were converted to film cooling effectiveness. The experimental details were described in Refs. [43–45] including information for uncertainties in the measurement of the thermochromatic liquid crystal. Table 2 presents the experimental condition for each of the measurement.

5 Results and Discussion

5.1 Blowing Ratio of Each Film Cooling Hole. Figure 6 shows the time-averaged fraction of blowing ratio for each film cooling hole to mean blowing ratio $BR$ for the three holes. All the turbulence models estimate nearly the same blowing ratio fractions for each of the cooling hole. The blowing ratio of hole 1 and hole 2 located close to the stagnation region is smaller than hole 3. In the case of the small mean blowing ratio ($BR \sim 1$), the blowing

<table>
<thead>
<tr>
<th>$BR$</th>
<th>$\rho_<em>U_</em>$ $(\text{kg} / (\text{m}^2 \cdot \text{s}))$</th>
<th>$T_*, K$</th>
<th>$\rho_*U_2$ $(\text{kg} / (\text{m}^2 \cdot \text{s}))$</th>
<th>$T_2, K$</th>
<th>Second air chamber supply $(\text{kg} / \text{s})$</th>
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</thead>
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<tr>
<td>1</td>
<td>19.29</td>
<td>291.17</td>
<td>19.82 (DES)</td>
<td>321.15</td>
<td>0.030</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>19.87 (V2F)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>20.06 (SA)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>41.06 (DES)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>41.60 (V2F)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>41.85 (SA)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>19.28</td>
<td>290.57</td>
<td>20.06 (SA)</td>
<td>329.15</td>
<td>0.030</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>41.85 (SA)</td>
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</tbody>
</table>

Fig. 4 Variations of predicted effectiveness with mesh

Fig. 5 Locations of temperature measurement by TC: (a) measured planes and (b) measurement points in each plane

Table 1 Computational conditions
ratio fraction of hole 2, as which hole 3 is located in the same side of the semicircular model, is larger than hole 1 located on the opposite side at the same angle from the stagnation line without film holes as hole 2. Those differences of the blowing ratio fractions are due to static pressure distribution around the leading edge model, and decreases with the increase of the mean blowing ratio because of the increase of the dynamic pressure of the secondary flow ejecting. Hereinafter, the mean blowing ratio is designated as a “blowing ratio.”

5.2 Film Cooling Effectiveness on the Model Surface. Figure 7 shows predicted distributions of time-averaged local film cooling effectiveness on the model surface, and Figs. 8(a) and 8(b) show measured distributions of the film cooling effectiveness with (a) the thermocouple probe and (b) the thermochromatic liquid crystal, respectively. The black area in Fig. 8(b) is owing to none reaction of the thermochromatic liquid crystal in the transient method. The measured distribution agrees with each other for the angle α < 0 deg, and the numerical distributions and their variations with the blowing ratio BR are generally similar as the experimental measurements. Both of the measurements and the predictions indicate that local effectiveness decreases and film coverage get worse with the increase of the blowing ratio BR, owing to the increase of penetration of the ejected secondary air into the main flow. All the results also show that the film coverage region leans downward of the model as the blowing ratio BR increases.

The CFD calculations, however, predict local higher effectiveness than the measurements in regions downstream of the trailing edges of the film cooling holes, as shown in Fig. 7. This local high effectiveness in the prediction is more remarkable and concentrated spatially in the predictions by the SA than in the other predictions, especially in the region downstream of hole 3. Furthermore, the distribution by the SA in the case of the smaller blowing ratio BR=1.04 shows a peak area (a plateau) of high effectiveness down hole 1 (α > 30 deg), which is not found in the experimental measurement by the thermochromatic liquid crystal of Fig. 8(b) (BR=1.06) and in other numerical predictions.

Figure 9 shows time-averaged distributions of the spanwise-averaged film cooling effectiveness. In this figure, previous results by steady calculations using the SST model [33] are also included. The figure indicates that all the CFD calculations have reasonable capabilities to predict the spatial-averaged profiles of the leading edge film cooling. The DES and the V2F results are comparatively in agreement with the measurements. However, the SA of BR = 1.04, as shown in Fig. 7, and also the previous steady calculation by the SST model of BR = 2.05 predict high effectiveness in the downstream region of hole 1 (α > 30 deg), which appear to be qualitatively different from the measurement.

5.3 Local Temperature Distribution Around the Model. Figure 10 shows time-averaged distributions of measured and predicted local temperatures on transverse planes normal to the model surface, at each angle of α = –30 deg, –40 deg, –70 deg,
and −90 deg, which is downstream of hole 2 and hole 3. In Fig. 10, time-averaged spanwise profiles of the local film cooling effectiveness on the model surface are also presented at the same angles α as those normal planes in the flow field. The CFD results shown in these figures are time-averaged, and the measurement data are acquired by traversing the thermocouples. The local temperature on the normal plane is nondimensionalized by the same definition as the film cooling effectiveness η. The previous steady calculations using the SST model [33] are also included in the figures of the spanwise profiles on the model surface.

Every turbulence model shows relatively good performance for predicting the location of bulk of the secondary air ejected into the main flow on the normal planes, and simulates departures of the secondary air from the surface with the increase of the blowing ratio BR. The V2F, especially, as shown in Fig. 10(b), in the case of BR = 1, appears to have advantage in the prediction of the location of the bulk secondary air on the normal plane. Distributed patterns of the local temperature are supposed to be affected by a so-called kidney-shape vortex pairs, which are found clearly in the results by the SA at the angles α = −30 deg of BR = 1.04 and α = −30 deg and −40 deg of BR = 2.17. Those predicted patterns of the local temperature indicate that the vortices, such as the kidney-shape, do not have a symmetric structure with respect to the center of the cooling holes as found in the flat plate cooling. In this leading edge film cooling, the entrainment of the main flow by the vortex appears to be larger on the upper side (the leading edge side of the hole) than on the lower side (the trailing edge side of the hole), so that the film cooling effectiveness is higher on the trailing edge side of the hole, as shown in Fig. 7.

Regarding those structures on the local temperature in the flow field, all the CFD calculations predict similar aspects but the SA estimates higher nondimensional temperature in their core regions than the V2F and the DES. These local distributions predicted by the V2F and the DES appear to be more diffusive and are consistent with aspects in the measurements, and as shown in Fig. 10(b) of the higher blowing ratio (BR ~ 2), the V2F and the DES estimate the same extent of the normal spreading of the secondary air as the measurements but the SA underestimates that. In Fig. 10(a) of the lower blowing ratio (BR ~ 1), the normal spreading predicted by the V2F is a little larger than the DES but the spreading of the secondary air in every calculation shows similar extent to each other.

Furthermore, in Fig. 11, presenting the predicted time-averaged local temperature distributions on normal planes to the model surface downstream of hole 1 (at angles α = 30 deg and 60 deg), the SA evaluates the bulk of the secondary air to be remarkably closer to the wall, shown in Fig. 11(a) of BR = 1.04 and the normal spreading of the secondary air in the SA is also smaller than the other models. As a result, as shown previously in Figs. 7(c) and 9(a), the plateau of the high film cooling effectiveness downstream of hole 1 is estimated in the SA than in the others. This local high wall effectiveness is also presented in the right hand and lower figure of the spanwise profile of Fig. 11(b). In terms of prediction of peak in the local film cooling effectiveness on the surface shown in the right hand figures of the spanwise wall profiles of Fig. 10, the DES and the V2F show better performance and the DES appears to be the best coincident with the measurements in the present. However, there are discrepancies between the predictions and the measurements. The present unsteady SA even shows equal performance to the steady calculations by the SST model [33] and has larger divergences in the location and the magnitude of the peak.
Figure 10 shows predicted instantaneous distributions of the local nondimensional temperature on the cross-sectional planes normal to the model surface at the hole 2 center (at the angle \( \alpha = -15 \) deg) of \( BR \sim 1 \). In the present turbulence models, the DES predicts the smallest scales around the interface between the main flow and the secondary air. The V2F also shows the wave pattern around there but in the SA, the interface did not find any fluctuation. Here, the DES could only predict the instantaneous penetration of the main flow into the film hole. That is useful for the assessment of the blade durability on the safe side.

5.4 Thermoflow Structure. Figure 13 presents predicted instantaneous views of vortex structures, 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 \( \eta \).

The predictable flow scales depend on the turbulence model, and there are large differences between the SA and the others. Those result from differences in estimations of the eddy viscosity among the turbulence models. The DES and the V2F predict vortex structures representing anisotropic motions, and the DES presents smaller scales resolved than the V2F. The extracted vortex structures from the DES results are found in slightly longer areas downstream of the holes than the V2F results. On the contrary, the SA evaluates the markedly short area where the vortices are found downstream of the holes. Those aspects indicate that the predicted flows by the DES and the V2F promote mixing by the vortex structures more than the flow predicted by the SA.

What the “Reynolds averaging” in the unsteady problem actually represents is not supposed to be well known. In this study, however, both the V2F and the SA based Reynolds averaging are conducted with the same mesh resolution and the same temporal integration as the DES. It is important at least that those dis-
cretized equations appropriately evaluate the eddy viscosity at each cell and each time step. The present numerical results show that the SA of the one-equation model, which is not able to correctly take the “turbulence length scale,” yields a stronger spatial filter than the others. On the other hand, the V2F, which takes into account the wall asymptotic behaviors of the turbulence properties with four constitutive equations, appears to reasonably estimate the eddy viscosity. However, in the RANS, especially the V2F, there must also be influences of the MUSCL (up-wind) scheme on the predictions.

The DES and V2F similarly predict distinctive vortex structures at around envelopes of the bulk structures, which are shear-layers between the main flow and the ejected secondary air. Inside these shear-layer vortices in the DES and the V2F, other structures with higher nondimensional temperature are found. However, in the SA calculations, the shear-layer vortex structures are not found. As shown in predicted instantaneous iso-temperature surfaces (\( \eta = 0.4 \)) of Fig. 14 (at the same angle of depression as Fig. 13), the DES and also the V2F predict fluctuations of the local temperature. These pictures for the DES and the V2F in Fig. 14 exhibit chiefly that the structures in the shear-layers influence the fluctuations of the local temperature. On the other hand, the SA estimates the smooth iso-surfaces, indicating a small fluctuation of the local temperature field. These aspects are consistent with the local temperature distributions on the cross-sectional planes in the flow field, presented in Fig. 12.

Figure 15 views predicted (a) time-averaged and (b) instantaneous iso-temperature surfaces (\( \eta = 0.4 \)) from the bottom of the model for the blowing ratio \( BR \sim 1 \). In every instantaneous picture in Fig. 15(b), the volumes wrapped by the temperature iso-surfaces are seen to remain downstream of the film cooling holes along the surface curvature of the model. The SA only predicts the continuous volumes along the model surface in the instantaneous picture. As shown in Fig. 15(a) for the time-averaged views, however, the DES and the V2F present the volumes shorter downstream of the holes (clearly downstream of hole 1) than the SA. These are supposed to be due to the fact that the DES and also the V2F evaluate the mixing by the unsteady vortex structures. On the contrary, the SA distinctively estimates the long time-averaged volumes of the temperature that are similar to the instantaneous iso-surfaces.
one. Those characteristics of the predicted flow structures differ from each turbulence modeling give information for the diffusion behavior of the local temperature field.

The high film cooling effectiveness downstream of hole 1 (a > 30 deg) at BR=1.04 in the SA, presented before in Figs. 7(c) and 9(a), is relevant to the predicted characteristics of the local temperature field mentioned above. It seems that the stable and continuous volume of higher temperature above the surface than the main flow temperature could be brought closer to the surface in the flow accelerated along the curvature of the leading edge, as shown in Fig. 11(a). This disadvantage found in the present unsteady simulation of the SA model would be also true with the steady RANS simulations, as presented in Fig. 9(c) for the spanwise-averaged effectiveness by the SST model. The present results illustrate a risk, wherein the steady simulation that is not able to evaluate unsteady vortices also fails to predict even a spatially-averaged performance of film cooling on a curved surface.

In this regard, however, in the present study, the temperature of the second air was set higher than the main flow temperature to be brought closer to the surface in the flow accelerated along the curvature of the leading edge, as shown in Fig. 11(a). This disadvantage found in the present unsteady simulation of the SA model would be also true with the steady RANS simulations, as presented in Fig. 9(c) for the spanwise-averaged effectiveness by the SST model. The present results illustrate a risk, wherein the steady simulation that is not able to evaluate unsteady vortices also fails to predict even a spatially-averaged performance of film cooling on a curved surface.

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6 Conclusions

In order to horizontally assess unsteady CFD calculations by using RANS modeling for the prediction of the film cooling on the semicircular leading edge model, the URANS simulations employing the k-ω-ε-σ-f model and the Spalart and Allmaras turbulence model and the DES based on the Spalart and Allmaras turbulence model were performed and those results were compared with the temperature measurements with the thermocouples and thermochromatic liquid crystal. As a result, some aspects for the performances of those CFD simulations were brought out.

The DES and the k-ω-ε-σ-f model presented good performances in estimations of time-averaged, spanwise-averaged film and local cooling effectiveness on the leading edge surface. Overall, it was concluded that the predicted temperature fields by these simulations were more diffusive and comparatively in agreement with the measurements than the Spalart and Allmaras model and the steady calculation by the SST model done previously. However, there were some quantitative discrepancies in the time-averaged local peak of the effectiveness on the surface in the predictions by the DES and the k-ω-ε-σ-f model. The Spalart and Allmaras model predicted the high film cooling effectiveness on the partial surface at angle α > 30 deg in the case of the blowing ratio BR ~ 1, which was not found in the measurement. That was also similar with the steady calculations by the SST model.

The DES and also the k-ω-ε-σ-f model evaluated explicitly the unsteady fluctuation of local temperature induced by the vortex structures (anisotropic motions). However, the Spalart and Allmaras model could not clearly predict the unsteadiness and the anisotropic motions despite of the unsteady simulation. In the present turbulence modeling, the DES resolved the smallest scale, and only predicted the penetration of main flow into the film cooling hole. The k-ω-ε-σ-f model predicted similar structures to the DES but did not show that penetration of the main flow. Those were due to the differences in the evaluations of the eddy viscosity among the models.

The partial overestimation of the local film cooling effectiveness by the Spalart and Allmaras model mentioned above is supposed to be due to the lack of the unsteady vortex structures in the accelerated flow along the leading edge curvature. This disadvantage found in the present unsteady simulation of the Spalart and Allmaras model would be also true with ordinary steady RANS simulations.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tr>
<td>BR</td>
<td>mean blowing ratio for all film cooling holes, ( \frac{p_U U_z}{p U_m} )</td>
</tr>
<tr>
<td>D</td>
<td>diameter of the leading edge model, m</td>
</tr>
<tr>
<td>d</td>
<td>diameter of film cooling holes, m</td>
</tr>
<tr>
<td>h</td>
<td>heat transfer coefficient, W/(m² K)</td>
</tr>
<tr>
<td>LQ</td>
<td>thermochromatic liquid crystal</td>
</tr>
<tr>
<td>n</td>
<td>wall-normal distance, m</td>
</tr>
<tr>
<td>Q</td>
<td>second invariant of gradient of velocity</td>
</tr>
<tr>
<td>q</td>
<td>heat flux, W/m²</td>
</tr>
<tr>
<td>Re_p</td>
<td>Reynolds number, ( \frac{p U_m D}{\mu} )</td>
</tr>
<tr>
<td>T</td>
<td>temperature, K</td>
</tr>
<tr>
<td>TC</td>
<td>thermocouples</td>
</tr>
<tr>
<td>t</td>
<td>time, s</td>
</tr>
<tr>
<td>U</td>
<td>mean velocity, m/s</td>
</tr>
<tr>
<td>x, y, z</td>
<td>Cartesian coordinates, m</td>
</tr>
<tr>
<td>α</td>
<td>angle, deg</td>
</tr>
<tr>
<td>η</td>
<td>film cooling effectiveness, ( \frac{T_m - T_{wall}}{T_m - T_2} )</td>
</tr>
<tr>
<td>μ</td>
<td>viscosity, Pa s</td>
</tr>
<tr>
<td>ρ</td>
<td>density, kg/m³</td>
</tr>
</tbody>
</table>

Subscripts

- aw     | adiabatic wall |
- ave    | average in the pitchwise direction of the model |
- hole   | relative to each cooling hole |
- M      | relative to main flow |
- W      | wall |
- 2      | relative to film flow |

References


