Numerical Study of Pin-fin Cooling on Gas Turbine Blades

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Abstract. This paper describes a numerical study of internal pin-fin cooling performance of a trailing-edge cutback configuration for gas turbine blade. The study was performed at two steps: first, to validate simulation results from an existing TE cutback cooling with staggered pin-fin arrays inside the cooling passage against experimental measurements. Three structured meshes were used for grid convergence and to evaluate film-cooling effectiveness and discharge coefficient; second, to investigate the pin-fin cooling performance with various blowing ratios. Simulations were performed by keeping the same initial and boundary conditions as the corresponding experiment. The results show that validation can be considered acceptable by keeping quality grid and its resolution in near wall regions. Both computational data of the adiabatic film-cooling effectiveness and the discharge coefficient are in fairly good agreement with the test data. The pin-fin array has important roles to promote flow turbulence activity inside the cooling passage, in addition to increase surface areas for heat transfer. Hence the turbulence intensity is more pronounced due to the existence of the pin-fin and it is concomitant with the coolant flow inside the wedge-shaped duct.

INTRODUCTION

Modern gas turbines often operate at very high inlet temperatures up to 1,500°C to achieve the highest possible overall engine performance in terms of thermal efficiency and power output. It can be up to 2,000°C for aircraft gas turbines of extreme higher engine power demand. Turbine blade is usually cooled down using various internal and external cooling techniques in order to keep the surface temperature well below the melting point of metal blade for safety and durability of engine operations. In fact, a higher turbine inlet temperature could lead to other adverse effects, including melting, oxidation, corrosion, erosion, degradation of structural strength, cracks, thermal-fatigue, and buckling, thus risking turbine blade failure [1-2].

The internal feature of pin-fin arrays inside the cooling passage has major impact on the performance of the gas turbine blade trailing-edge cooling, as the pin-fin cooling is commonly integrated with the trailing-edge ejection cooling. The pin-fin geometries usually have the height-to-diameter ratio between 0.5 and 4. This component typically serves to enhance the heat transfer from the blade wall around the pin-fin section. As expected, heat can be transferred from both the end-walls and the pin-fin arrays. In addition, the flow separation and wakes are shed at the pin-fin downstream, followed by horseshoe vortex. It creates additional flow mixing, and further enhances the heat transfer process. One industry challenging of this application is mainly the manufacturing constraint in a very tight trailing-edge of gas turbine blade. It has been found that the distribution of heat transfer in the cooling passage is strongly affected by the design of pin-fin array [3], in which the design aspects of internal features, such as the type of array, spacing, size and shape, are highlighted for consideration when investigating the pin-fin cooling.

Over the past decades, the blade trailing-edge cooling performance has been studied experimentally and numerically with various internal configurations inside the cooling passage, including staggered or in-line arrangements with cylindrical [4-5], elliptical pin-fin [6-7], long ribs [8-9], rectangular [10-11], diamond [12] and drop-shaped [13], respectively. Amongst these studies, Tarchi et al. [4] carried out an experimental research by proposing innovative trailing-edge internal cooling designs with a pentagonal arranged circular pin-fin and a staggered elliptical pin-fin. Similar configurations were further investigated by [5], addressing the turning flow effects in front of the cooling passage. Author [14] also evaluated the turning flow effects using different internal cooling designs with enlarged pedestals and square or semi-circular ribs. Their results showed that the combined effect of flow acceleration and pin-fin array could lead to a significant increase of the heat transfer performance. The heat transfer coefficient of pin-fin surface was found always greater than that of the surrounding end walls. In terms of the pressure loss, the streamwise elliptical pin-fin orientation has exhibited a significant contribution towards an overall decrease of the pressure drop.

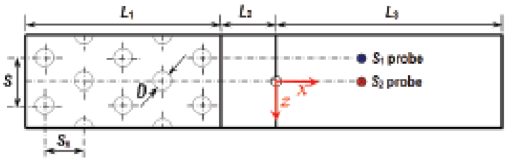
By referencing to the experiments of [15-16] and [17], the present paper will carry out numerical studies of the pin-fin arrays inside the cooling passage of blade trailing-edge cutback model. An experimental configuration investigated by those researchers will be used to demonstrate the effectiveness of the pin-fin cooling and their heat transfer performance.

NUMERICAL TREATMENTS

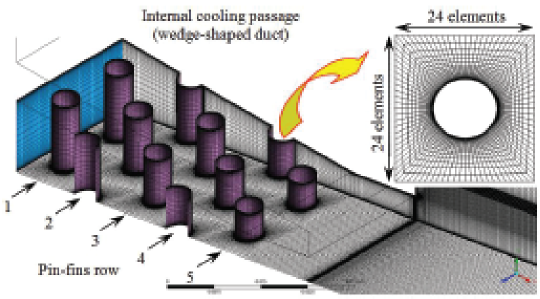
Physical problem, flow conditions and mesh

The present numerical study considers an experimental model with five rows of staggered-array cylindrical pin-fin previously investigated by [16]. The key dimensions are shown in inserted table in Fig. 1(a). The arrays are fitted into a 10-degree wedge-shaped duct (L_1 region), replicating a typical trailing edge (TE) shape of the gas turbine blade. It represents the shape of trailing edge that follows the angle of the coolant ejection slot. Both spanwise pitch (S/D) and streamwise pitch (S_x/D) ratios are kept the same as the experiment. The ratio of lip thickness to coolant passage height (t/H) is set to be 1. The coolant ejection area known as the slot-exit (A_{slot}) is located under the lip position.

Dimensions	L_1	L_2	L_3	S	S_x	t	H	D	α
mm	52	14.4	60	12	10.4	4.8	4.8	4.8	10°



(a) The geometry of domain.



(b) The 3D structured mesh of a wedge-shaped duct, inserted by the local one-block 2D structured grids around the pin-fin geometry.

FIGURE 1. Geometry and meshing

The computational study used the same inflow and boundary conditions as those previous studies carried out by [16]. The mainstream inflow has a fixed velocity (u_{hg}) of 56 m/s and static temperature (T_{hg}) of 500 K. The coolant inflow was a fixed temperature (T_c) of 293 K, while coolant velocity (u_c) was varied from 4 to 15 m/s, subject to the blowing ratio (M) ranging from 0.5 to 1.1.

The computational domain was carefully constructed within high quality grids of $\Delta y^+ < 1$ on all surfaces as shown in Fig. 1(b), while the adiabatic/protected wall surface, as an important area, was constructed by higher quality meshes with average $\Delta y^+ < 0.5$ as suggested by some researchers. The grid size is a key important factor, independent of the near wall grid resolution [18]. The consistency of local grid spacing in the 3-D domain was refined in each of the x, y and z directions, in order to have a sufficiently fine spatial resolution to capture unsteady flow phenomenon in the mixing region.

Algorithm

A finite-volume method was utilized to solve the governing equations for incompressible flow condition. The equations are discredited and computed spatially using second-order accuracy scheme on multi-block structured grids, whilst temporally calculated with the second-order fully implicit algorithm. The SIMPLEC algorithm was chosen with second-order numerical scheme applied for all flow equations (i.e., pressure, momentum, and energy) of URANS and DES calculations. With respect to the stability requirement, a small time-step size up to 1.25×10^{-5} seconds was applied in the computations.

Coefficient of discharge (CD) and Adiabatic film-cooling effectiveness ($\overline{\eta}_{aw}$)

Same formulas as seen in [16] are used to evaluate effectiveness, discharge coefficient and blowing ratios. These equations also have been presented in publication released by [19].

The performance of a trailing-edge cutback cooling is commonly expressed by film-cooling effectiveness at the protected walls between the slot-exit and downstream of the trailing edge. If the surfaces of TE cutback have the adiabatic wall condition, a film-cooling effectiveness can be derived from the ratio of temperature difference between the hot gas and the predicted wall surface temperature to the hot gas and the coolant gas temperature difference as given in equation (1).

$$\eta_{aw} = \frac{T_{hg} - T_{aw}}{T_{hg} - T_{c'}}, \tag{1}$$

Where T_{aw} is the temperature at the adiabatic wall surfaces, T_{hg} is the hot gas temperature at the mainstream flow at inflow region and $T_{c'}$ is the coolant gas temperature measured at the centre of the slot-exit between two neighboring pin-fin geometries.

The discharge coefficient, C_D, is a representation of the global pressure losses inside the cooling passage, which is defined by the measured coolant mass flow over the ideal mass flow as formulated below:

$$C_D = \frac{\dot{m}_{c,real}}{\dot{m}_{c,ideal}}, \tag{2}$$

$$C_D = \frac{\dot{m}_{c,real}}{p_{1,t} \cdot \left(\frac{p_2}{p_{1,t}}\right)^{\frac{\kappa+1}{2\kappa}} \cdot A_{slot} \cdot \sqrt{\frac{2\kappa}{(\kappa-1) \cdot R \cdot T_{1,t}} \left[\left(\frac{p_{1,t}}{p_2}\right)^{\frac{\kappa-1}{\kappa}} - 1\right]}}, \tag{3}$$

where $p_{1,t}$ and $T_{1,t}$ are the total pressure and temperature at the coolant inlet, respectively, p_2 is the static pressure at the slot exit, A_{slot} is the area of the slot exit, κ is the specific heat capacity and R is the gas constant.

Blowing ratio (M), Reynolds number (Re), and Heat transfer coefficient (HTC)

The non-dimensional blowing ratio (M) is defined as a factor of slot-averaged mean density and velocity product over the density and velocity product at the mainstream hot gas inlet plane. The equation can be further transformed using the mass flow rate at the slot exit, which is equal to that of the cooling inlet (i.e. mass conservation).

$$M = \frac{\overline{(\rho_c u_c)}_{slot}}{\rho_{hg} u_{hg}} = \frac{\dot{m}_c}{A_{slot} \rho_{hg} u_{hg}}, \tag{4}$$

The Reynolds number (Re) is formulated in two different manners, the first is referred to throat section, representing the minimum area of wedge-shaped duct and the second is based on the hydraulic diameter of inflow section (L_0) as:

$$Re_d = \frac{\dot{m}D}{A_{min}\mu} \text{ and } Re_{L0} = \frac{\dot{m}D_{L0}}{A_{L0}\mu}, \tag{5}$$

Where Re_d and Re_{L0} are the Reynolds number at the throat section and the inflow section respectively, D and D_{L0} are the diameter of pin fin and inflow section, A_{min} is the minimum passage area between two adjacent pin-fin, A_{L0} is the cross section area of inflow section, μ and is the dynamics viscosity and k is the thermal conductivity at the L_0 region.

The heat transfer coefficient (HTC) is calculated by equation (6) below.

$$h = \frac{q_w}{(T_w - T_{nw})}, \tag{6}$$

Where h is the heat transfer coefficient, T_w is the wall temperature, T_{nw} is the near-wall fluid temperature, q_w is the wall heat flux.

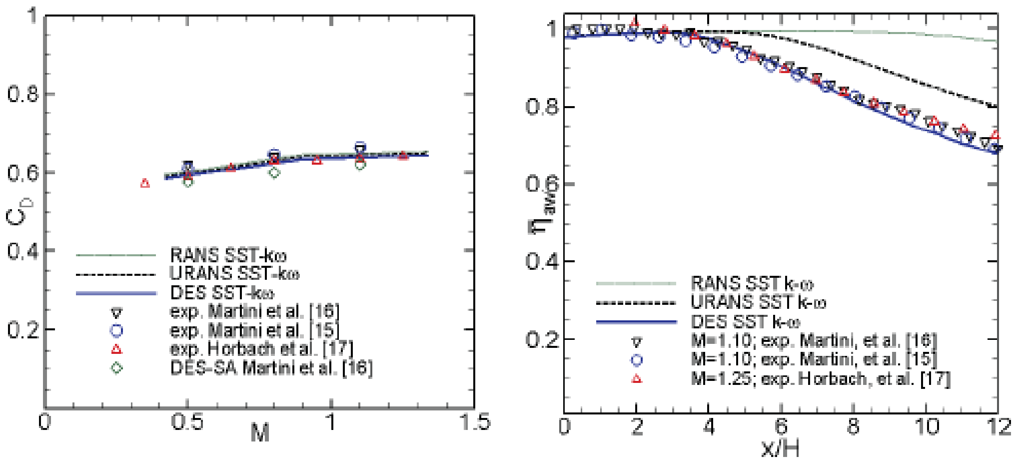
RESULTS AND DISCUSSION

Grid refinement study and Validation

The grid refinement studies consider three successive meshes of the computational domain from coarse to fine at a fixed-wall wall temperature (T_w) of 325 K (see [20]). The aim is to assess the mesh sensitivity on predicting adiabatic film-cooling effectiveness for three types of meshes, considering detached eddy simulation (DES) at low and high blowing ratios (M = 0.5 and 1.1), in comparison with the experimental data of [16] and [17]. It forms the grid independence tests for two different blowing ratios. Previous studies suggested that the DES modeling requires a fine resolution, to resolve the flow and heat transfer in the near end-wall region. The results from the coarse mesh cause largely over-prediction of the film-cooling effectiveness near the downstream region. The use of the fine mesh produces results that are in a good agreement with the test data along the protected/adiabatic wall surfaces from the slot-exit at x/H = 0 to the downstream region of x/H = 12, with the importance cooling effectiveness decay being captured successfully. The effects of mesh resolution have shown clearly that the deviation is large for a low-blowing ratio and relatively low for a higher blowing ratio.

Figure 2(a) gives the predicted discharge coefficients (C_D) for three different blowing ratios (M) and with various turbulence models, in comparison with the experimental measurements. It has been found that simulation results using both steady/unsteady RANS and DES are in good agreement with the experimental data.

Figure 2(b) illustrates the variations of the adiabatic film-cooling effectiveness along the adiabatic/protected wall surface from the exit to downstream at various blowing ratios, in comparison with experimental measurements and previous numerical results using DES-SA model [10]. It has been found that the SST $k-\omega$ turbulence used in present study is capable of predicting correct trend compared to the eddy simulation. CFD data follows a strong decay of the adiabatic film-cooling effectiveness, which is the same with the experimental measurements. The decay is slightly reduced by increasing the blowing ratio.

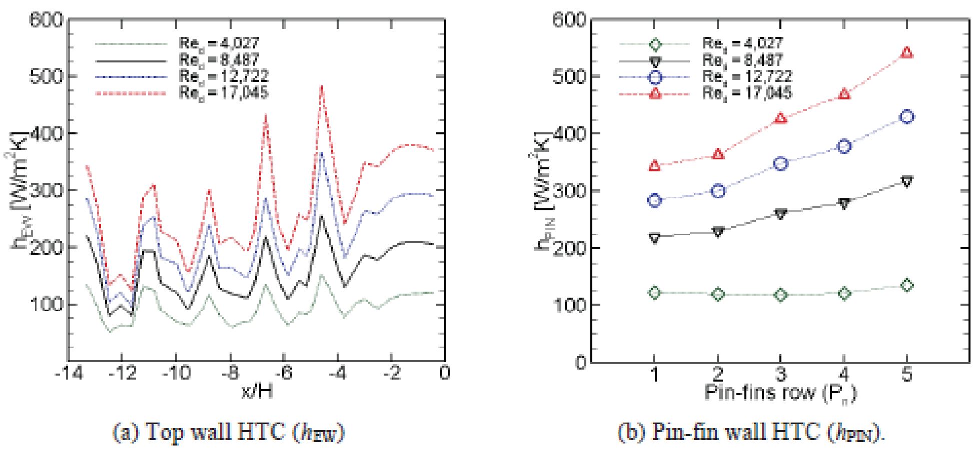


(a) The vaidated data of C_D. (b) The validated data of FCE.

FIGURE 2. Validated data.

Heat transfer coefficient of the pin-fin array

Figure 3(a) shows the span-wise averaged heat transfer coefficient (h_{EW}) at the top walls of the coolant passage. It has been recognized that the curves of span-wise averaged heat transfer coefficient show large peaks in connection with the pin-fin rows. It is more obvious for a higher Reynolds number. This finding is consistent with previous experiment by Authors. [4], who studied internal trailing edge cooling with staggered pin-fin. Authors [5] also found out the same trend based on the investigations of an innovative pin-fin cooling with considering turning flow effect at the inflow region. The presence of the heat transfer coefficient peaks corresponds to the pin-fin stagnation points.



(a) Top wall HTC (h_{EW}) (b) Pin-fin wall HTC (h_{PIN}).

FIGURE 3. The averaged heat transfer coefficient.

Figure 3(b) indicates the averaged heat-transfer coefficient (h_{PIN}) at the surface of the pin-fin array of various Reynolds numbers at the coolant slot, which is known as the throat section (A_6). The heat transfer coefficient is seen to increase about a factor of two with the Reynolds number. The increase of heat transfer coefficient near the slot exit is more obvious. It is also more noticeable in line with the increase in the Reynolds number. The experiment carried out by [4] yielded a similar increase of heat transfer, compared to the finding in the present study. Despite that they have used seven rows of staggered pin-fin inside the wedge-shape duct, and the internal pin-fin cooling was not coupled with trailing-edge cutback. Their boundary condition was also very different from the present computation.

Figure 4 illustrates the qualitative comparison in an effort to complete the quantitative heat transfer coefficient as previously explained in Fig 3. It found that the distribution of heat-transfer coefficient is broadly symmetric along the staggered arrays. A stagnation region is present in front of each pin-fin array. It is indicated by the red color in the left position for each pin-fin row. The red color is seen to be obvious near the slot exit position ($x/H = 0$). The increase of Reynolds number has a significant effect on the intensity of the red color. It is more pronounced for higher blowing ratios or higher Reynolds numbers of the coolant fluids inside the cooling passage. This finding is seen to be in good agreement with the findings reported by [4]. The blue color within the range of $-3 < x/H < 0$ at the bottom wall indicates the adiabatic boundary condition in this region. It relates to a zero heat transfer coefficient at the bottom wall.

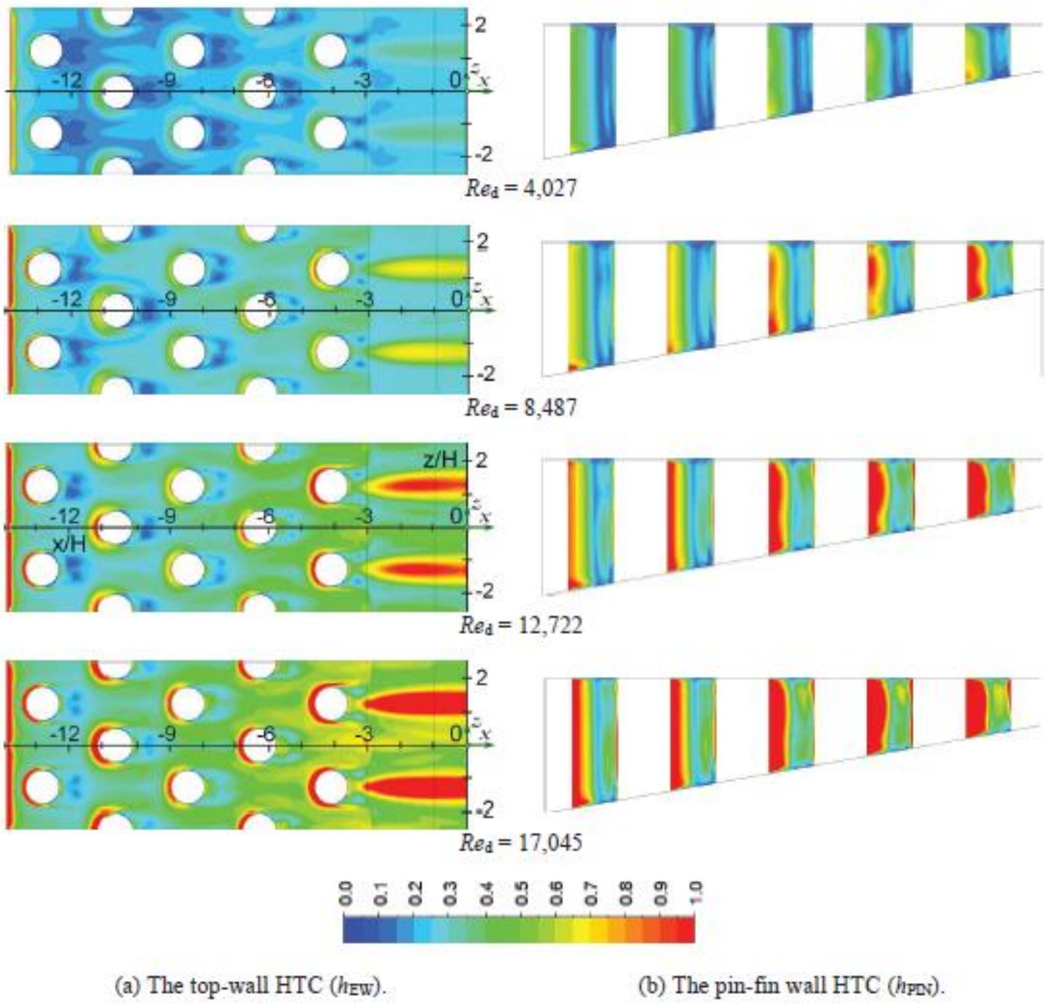
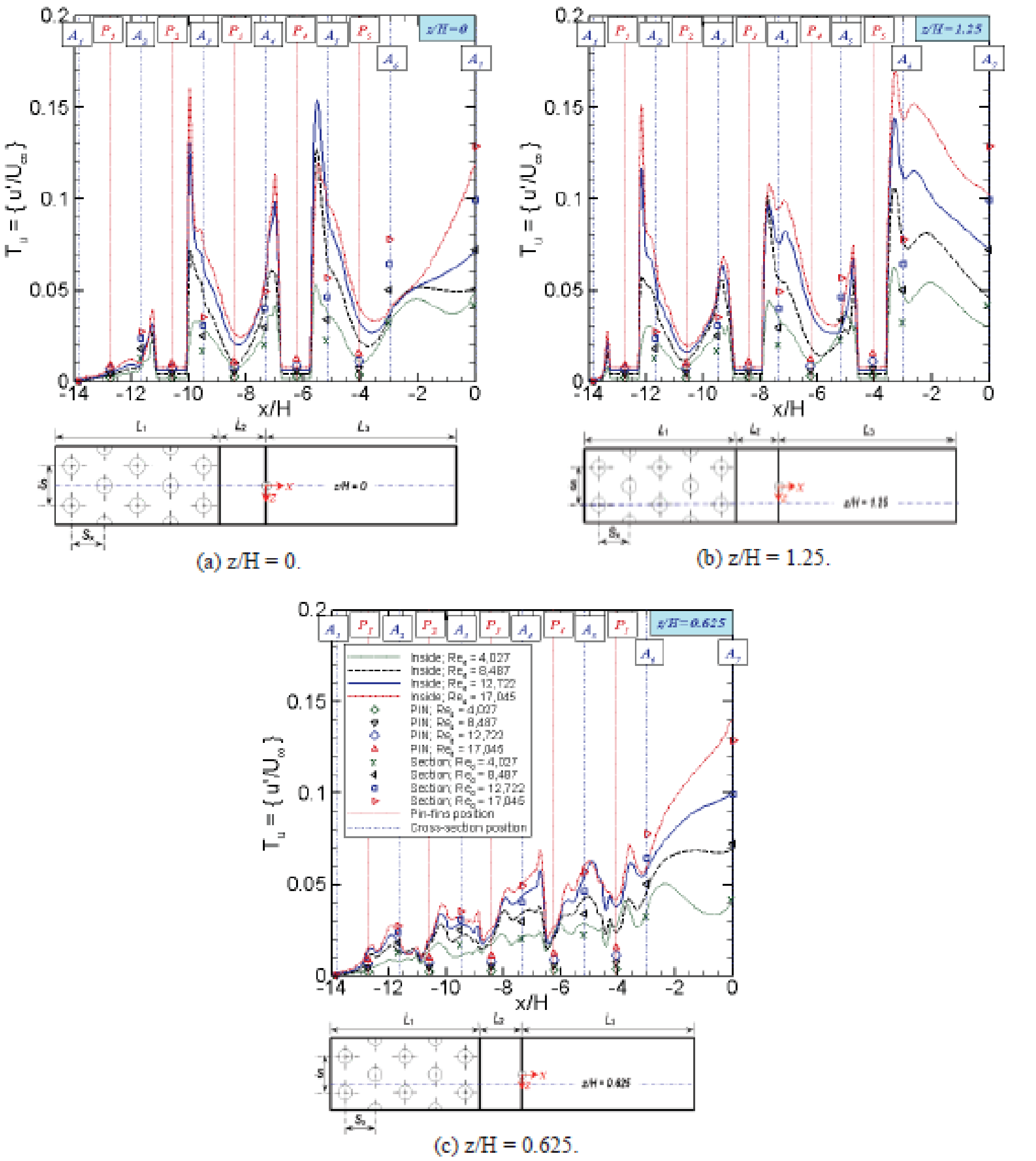


FIGURE 4. The averaged heat transfer coefficient visualization.

Turbulent intensity along the cooling passage

Figure 5 represents the characteristics of turbulence levels ($T_u = u'/U_\infty$) for various Reynolds numbers at the coolant slot. The u' velocity is based on the root-mean-square (RMS) streamwise fluctuation flow velocity. It was found that the turbulence levels are gradually increased along the cooling passage as clearly indicated by the averaged turbulence levels at the cross-section areas (A_n). The growth of turbulence level is stronger than that at the ejection slot (A_7), mainly for higher blowing ratio. The peak levels are achieved at A_7 and A_6 locations for the case without and with lands, respectively. These levels are greater than that at the pin-fin surfaces (P_n).



(a) z/H = 0.

(b) z/H = 1.25.

(c) z/H = 0.625.

FIGURE 5. The characteristics of turbulence levels.

By analyzing data at three different positions of z/H = 0; 0.625; and 1.25, the turbulence level fluctuates are found largely dependent on relative position to the pin-fin array. The turbulence level behind the pin-fin is greater than that in front of the pin-fin. It is more pronounced concomitant with the coolant fluid inside the wedge-shaped duct. This higher level is likely related to the flow unsteadiness behind the pin-fin. A relative lower fluctuation at z/H = 0.625 compared to other positions implies that the pin-fin array has played important role in the turbulence flow motions inside the cooling passage, and also the flow mixing region thereafter.

CONCLUSION

The heat transfer performance in an internal cooling passage of a gas turbine blade has been studied numerically. CFD validated discharge coefficient (C_D) and heat transfer coefficient (HTC) are in good agreement with the experimental data. The averaged heat transfer coefficient at the surface of the pin-fin array is more noticeable in the region near the slot exit in line with the increase in Reynolds number. The pin-fin array has played an important role in the turbulence flow motions inside the cooling passage. The turbulence intensity is more pronounced due to the existence of the pin-fin, in which it is concomitant with the coolant fluid inside the wedge-shaped duct. It is conjectured to be related to the flow unsteadiness around the pin-fin.

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