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EFFECTS OF GUIDE VANES ON THE TIP HEAT TRANSFER ENHANCEMENT OF A TURBINE BLADE

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ABSTRACT
Gas turbine blade tips encounter large heat load as they are exposed to the high temperature gas. A common way to cool the blade and its tip is to design serpentine passages with 180-deg turns under the blade tip-cap inside the turbine blade. Improved internal convective cooling is therefore required to increase the blade tip life time. This paper presents numerical predictions of turbulent fluid flow and heat transfer through two-pass channels with and without guide vanes placed in the turn regions using RANS turbulence modeling. The effects of adding guide vanes on the tip-wall heat transfer enhancement and the channel pressure loss were analyzed. The guide vanes have a height identical to that of the channel. The inlet Reynolds numbers are ranging from 100,000 to 600,000. The detailed three-dimensional fluid flow and heat transfer over the tip-walls are presented. The overall performances of several two-pass channels are also evaluated and compared. It is found that the tip heat transfer coefficients of the channels with guide vanes are 10~60% higher than that of a channel without guide vanes, while the pressure loss might be reduced when the guide vanes are properly designed and located, otherwise the pressure loss is expected to be increased severely. It is suggested that the usage of proper guide vanes is a suitable way to augment the blade tip heat transfer and improve the flow structure, but is not the most effective way compared to the augmentation by surface modifications imposed on the tip-wall directly. [**Keywords:** Heat transfer enhancement; Tip-wall; Guide vanes; Numerical prediction]

INTRODUCTION
It is well recognized that one way to achieve higher power output of gas turbine engines is to realize higher turbine inlet temperature. However, such realization poses rigorous challenges for the design of advanced cooling methods for turbine blades. Blade tip regions encounter high heat loads due to the high temperature leakage flow. Cooling methods are therefore needed to improve the tip cooling to ensure a long life time of the blade tip. In a typical cooled turbine blade, multiple channels aligned with the blade spanwise direction are connected by 180-deg turns to produce a serpentine passage to fully utilize the cooling potential of the coolant, as shown in Fig. 1. In this way, the blade internal tip can be cooled to a certain extent by means of turn induced impingement. Augmented internal convective cooling for the tip-cap is required to meet the high-performance gas turbine with high inlet temperature. The 180-deg turn is also known to cause a substantial increase in pressure loss due to the complex flow with separation, mixing and impingement involved.

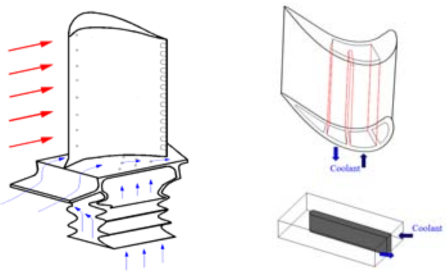


Fig. 1 A typical serpentine passage inside a turbine blade

There have been many studies on multipass (mostly, two-pass) channel flow fields and heat transfer. Both experimental measurements and numerical simulations show that low heat transfer associated with flow separations always occurs at the corners in the turn regions and downstream the turn along the inner walls, as depicted in Fig. 2. For example, from PIV measurements [1], a strong Dean-type secondary flow and a large separation bubble existed on the inner wall downstream of the bend exit. The secondary flow consists of two counter rotating vortices causing a strong impingement of the flow on the outer walls at the bend exit. Schabacker et al. [2] later measured flow field in a similar ribbed channel to show the interaction between the turbulator-induced secondary flow and the bend induced secondary flow. Son et al. [3] also measured the flow field and heat transfer inside a two-pass square channel with a smooth wall and a 90-deg turbulator-roughened wall, indicating that the flow impingement is the primary factor for the heat transfer enhancement rather than the flow turbulence level itself. Therefore, in order to modify or improve the flow structure with low heat transfer involved, properly designed guide vanes/ribs are suggested to be placed in related regions.

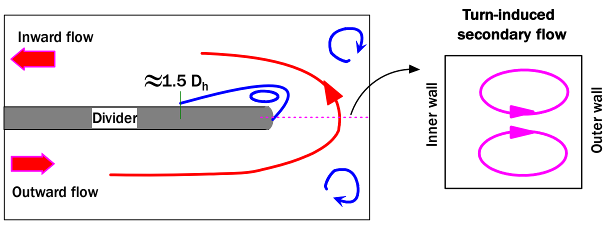


Fig. 2 Conceptual flow pattern around a 180-deg turn

However, there have been only a few studies on effects of guide vanes inside serpentine channels published in the open literature. Rathjen et al. [4] studied mass/heat transfer in a smooth two-pass channel with a 90-deg guide vane, whose intention is to avoid the separation zone. Liou et al. [5, 6] experimentally studied the effects of the number of guide vanes on the 3D flow field in a 60-deg curved combustor inlet. It was found that by adding three guide vanes the flow separation in the curved combustor could be eliminated completely, and in most regions with increasing number of guide vanes the maximum radial mean velocity, difference between radial and spanwise normal stress, and the turbulent kinetic energy were decreased. Rao et al. [7, 8] and Babu et al. [9] investigated the effects of guide vanes on pressure drop in smooth or roughened channels with a 180-deg bend. Experimental results showed that pressure losses were significantly affected by the shape and position of the guide vanes, and the overall pressure drop could be decreased by as much as 14-20% if properly shaped extended guide vanes were located in the center of the bend. Luo and Razinsky [10] performed numerical simulations of the turbulent flows inside several 2D and 3D 180-deg U-ducts with and without guide vanes by a RANS method. Results showed that the combined vane and uniform cross-section area led to much weaker secondary-flow vortices and smaller separation, which accounted for a substantial reduction in pressure loss. Very recently, Xie et al. [11] analyzed the effect of guide ribs on a blade tip heat transfer and found that the tip heat transfer could be increased by 20~50% while the channel pressure loss could be decreased by about 10% if arc-like guide ribs were placed. In their study, the ribs had cross-section blockage ratio of 0.182. Very recently, Schuler et al. [12] performed experimental measurement and numerical simulation of the effect of turning vanes inside a ribbed rectangular two-pass internal cooling channel. They found that the pressure drop was reduced by 25% while the heat transfer was kept the same level if an appropriate turning vane was applied.

For blade tip heat transfer, little information is available. In the earlier study of Han et al. [13], mass/heat transfer data of smooth tip surface (in the paper named outer wall in the turn) in smooth and turbulated square two-pass channels was measured by the naphthalene sublimation technique. The typical averaged tip heat transfer of smooth and ribbed channels at Reynolds number of 30,000 were about 1.8 and 2.5 times larger than the fully-developed duct turbulent heat transfer, respectively. Note that for the turbulated two-pass channel, only the inlet and outlet channels were turbulated while the tip wall was kept smooth. The turbulator height-to-hydraulic diameter ratios were 0.063 and 0.094. Wagner et al. [14, 15] measured the heat transfer in non-rotating and rotating multipass channels with and without turbulators placed in the straight sections. It was found that the heat transfer in the turn was approximately two and three times larger than the fully-developed duct turbulent heat transfer for the non-rotating and rotating conditions, respectively. The turbulators produced about 10% higher augmentation for the averaged tip heat transfer. In their studies, the turbulator height-to-hydraulic diameter ratio was 0.1 and the baseline Reynolds number and Rotation number are 25,000 and 0.24, respectively. A review of turbine blade tip heat transfer and issues of axial blade tip function, design and durability were addressed by Bunker [16, 17], and a subsequent review of tip heat transfer and internal/external tip cooling technology was provided by Sunden and Xie [18].

Recently, Bunker [19] measured the heat transfer of an internally cooled blade tip-cap with four kinds of arrays of shaped pins. It was found that the effective heat transfer coefficient could be increased up to a factor of 2.5 while the tip turn pressure drop increase was negligible. Several numerical studies of heat transfer enhancement of an internal blade tip-wall with circular pins, dimples and protrusions have been performed by the present authors [20-24]. Simulation results showed that the augmentation devices forced the vortices towards the tip wall and thereby improved the turbulent mixing of the approaching cold fluid and hot fluid near the internal tip-wall. The results were compared with similar experimental data by Bunker [19]. Decent overall agreement was found but the noted discrepancy was deemed to be related to the differences in the shape and arrangement of augmentation devices being imposed on the internal tip-cap directly.

Even though related work of the heat transfer augmentation in two-pass channels with shaped pin-finned tips can be found in the above-mentioned experimental work by Bunker [19] and the numerical simulations by the authors [20-24], no available details of the effects of guide vanes on the tip heat transfer are available. Furthermore, most previous studies were concerned with the heat transfer on the leading or/and trailing walls, or only with the pressure drop through the channels. Very limited information of tip-wall heat transfer augmentation produced by guide vanes is available. Accordingly, it is desirable to observe the effects of guide vanes on internal tip cooling, and to present more details on the tip-wall heat transfer enhancement. For these reasons, the main objective of the present study is to investigate the heat transfer enhancement over an internal blade tip inside two-pass channels with and without guide vanes placed at the turn region. The Reynolds number is ranging from 100,000 to 600,000. The overall performances of two-pass channels are also compared and evaluated subject to different criteria. Although similar work related to guide vanes was presented by the authors before [11], the vanes were like ribs because their height was small (9% of channel height). In the present work the effects of guide vanes whose height is the same as channel height on the blade tip heat transfer are considered.

NOMENCLATURE

A	wall surface area
D_h	hydraulic diameter
f	friction factor
H	channel height, vane height
h	heat transfer coefficient
k	turbulent kinetic energy
L	length of two-pass channel
Nu	Nusselt number
p	pressure
P	pumping power
Pr	Prandtl number
Pr_t	turbulent Prandtl number
q	wall heat flux
Re	Reynolds number, $Re=\rho\, u_i\, D_h/\mu$
T	temperature
u_i	inlet velocity
W	tip height (z-direction)
Y	tip width (y-direction)

Greek symbols

Δp	pressure drop
ε	Rate of energy dissipation
λ	fluid thermal conductivity
μ	fluid dynamic viscosity
ν	fluid kinematic viscosity
ρ	fluid density

Subscripts

0	fully developed flow channel
i	inlet
o	outlet
s	smooth-tip channel
t	turbulent
w	wall

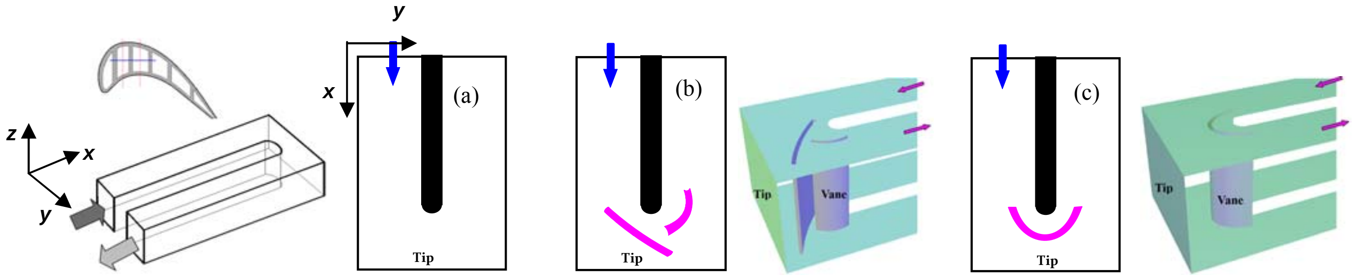
BRIEF DESCRIPTION OF PHYSICAL MODELS

The geometrical models simulated in this study are schematically shown in Fig. 3. The description of the two-pass channels can be found in previous papers [19, 20], and is not repeated here. Two schemes of guide vanes are designed in this study. One has two guide vanes placed near the corners, hereafter called Case I. The other has one arc-like guide vane placed near the divider, called Case II. Similar cases as Case II can be found in [7-9]. The guide vanes placed in the turn regions have a thickness of 5.08 mm and a height of 139.7 mm (identical to channel height). In all simulations, the guide vanes are assumed to be made of aluminum. For the smooth two-pass channel, no guide vanes are placed in the turn region.

It should be noted that the chosen geometry and dimension of the two-pass channels are similar to those in Bunker's experiments [19] except for the outlet. Also, heat transfer results were presented in the papers by Bunker [19] and the authors [20-24], but those heat transfer augmentations studied arose from the tip itself where pins, dimples and protrusions were imposed directly. This study concerns the tip heat transfer augmentation due to the presence of guide vanes in the turn region rather than in tip-wall. In this case, all the tip-walls are smooth without any augmented devices. The tip-wall heat transfer performance will be compared with those in previous works [20-24].

COMPUTATIONAL DETAILS

Turbulence Model: The key features of the flow in two-pass channels are separation, recirculation, and impingement. The prediction of heat transfer and cooling is most important for gas turbine design. Therefore, the agreement between the calculated Nusselt number and experimental data is the main criterion for selecting a turbulence model. Six different turbulence models, i.e., the standard k-ε model, the RNG k-ε model, the realizable k-ε model, the SST k-ω model, the v^2-f model and an RSM model have been evaluated for simulation of the turbulent heat transfer of the smooth two-pass channel [22-24]. Based on the previous evaluation, the realizable k-ε turbulence model has been selected for all computations.



(a) w/o ribs (b) Case I, two guide vanes (c) Case II, one guide vane

Fig. 3 Schematic of numerical models

Governing Equations: The governing equations of fluid flow and heat transfer for different variables can be expressed as follows [25].

Continuity equation

$$\frac{\partial u_{\mathrm{j}}}{\partial x_{\mathrm{j}}}=0 \tag{1}$$

Momentum equations

$$\rho\frac{\partial u_{\mathrm{i}}u_{\mathrm{j}}}{\partial x_{\mathrm{j}}}=-\frac{\partial p}{\partial x_{\mathrm{i}}}+\frac{\partial}{\partial x_{\mathrm{j}}}((\mu+\mu_{\mathrm{t}})(\frac{\partial u_{\mathrm{j}}}{\partial x_{\mathrm{i}}}+\frac{\partial u_{\mathrm{i}}}{\partial x_{\mathrm{j}}})) \tag{2}$$

Energy equations

$$\frac{\partial u_{\mathrm{i}}T}{\partial x_{\mathrm{i}}}=\frac{\partial}{\partial x_{\mathrm{i}}}((\frac{\mu}{Pr}+\frac{\mu_{\mathrm{t}}}{Pr_{\mathrm{t}}})\frac{\partial T}{\partial x_{\mathrm{i}}}) \tag{3}$$

Turbulent kinetic energy k equation

$$\frac{\partial}{\partial x_{\mathrm{j}}}(\rho u_{\mathrm{j}}k)=\frac{\partial}{\partial x_{\mathrm{j}}}\left[(\mu+\frac{\mu_{\mathrm{t}}}{\sigma_{\mathrm{k}}})\frac{\partial k}{\partial x_{\mathrm{j}}}\right]+\Gamma-\rho\varepsilon \tag{4}$$

Rate of energy dissipation ε equation

$$\frac{\partial}{\partial x_{\mathrm{j}}}(\rho u_{\mathrm{j}}\varepsilon)=\frac{\partial}{\partial x_{\mathrm{j}}}\left[(\mu+\frac{\mu_{\mathrm{t}}}{\sigma_{\varepsilon}})\frac{\partial \varepsilon}{\partial x_{\mathrm{j}}}\right]+C_1\Gamma\varepsilon-C_2\frac{\varepsilon^2}{k+\sqrt{\nu\varepsilon}} \tag{5}$$

where Γ represents the production rate of k and is calculated by

$$\Gamma=-\overline{u_{\mathrm{i}}u_{\mathrm{j}}}\frac{\partial u_{\mathrm{i}}}{\partial x_{\mathrm{j}}}=\nu_{\mathrm{t}}(\frac{\partial u_{\mathrm{i}}}{\partial x_{\mathrm{j}}}+\frac{\partial u_{\mathrm{j}}}{\partial x_{\mathrm{i}}})\frac{\partial u_{\mathrm{i}}}{\partial x_{\mathrm{j}}} \tag{6}$$

$$\mu_{\mathrm{t}}=\rho C_{\mu}\frac{k^2}{\varepsilon} \tag{7}$$

The coefficients in the turbulence model are

$$C_1=\max\left[0.43,\frac{\mu_{\mathrm{t}}}{\mu_{\mathrm{t}}+5}\right],C_2=1.0,\ \sigma_{\mathrm{k}}=1.0,\sigma_{\varepsilon}=1.2\,.$$

Boundary Conditions: This study concerns the effect of guide vanes on tip-wall heat transfer enhancement. Therefore, except for the smooth tip-wall, the remaining walls including vane walls are assumed to be adiabatic. In order to approach the experimental condition by Bunker [19] where a uniform heat flux was created by a heater, a constant heat flux is prescribed on the smooth tip-walls. No-slip velocity conditions are applied at all walls. At the inlet, uniform values of the inlet velocity and temperature (fixed at 300 K) are set and a turbulence intensity level of 5% and the hydraulic diameter are used to compute the inlet turbulence characteristics. An outflow condition is chosen at the outlet boundary.

The fluid is assumed to be incompressible with constant thermal physical properties and the flow is assumed to be three dimensional, turbulent, steady and non-rotating. The working fluid is dry air. In this study, because of the high Reynolds numbers and complicated computational model, the standard wall functions of the realizable k-ε model are applied on the walls for the near wall treatment. The minimum convergence criterion for continuity, momentum quantities error, and k and ε equations are 10^{-4} and 10^{-7} for the energy equation.

Grid Dependence: Typical surface grids are shown in Fig. 4. Because of curved guide vanes located in the turn region, an unstructured grid (T-grid) has to be filled into the domain of the turn region, which is occupying about 95% of the total number of cells. The grids around the guide vanes are fine while coarser grids are distributed near the side walls. For the inlet and outlet parts, structured grids are generated.

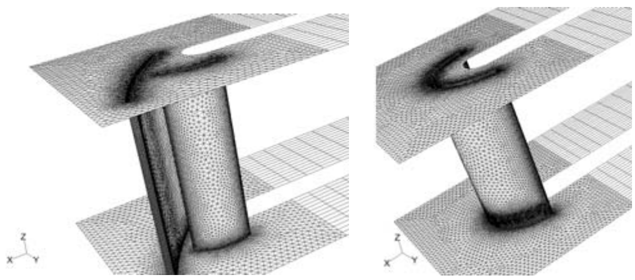


Fig. 4 Typical surface grids for computations

To ensure the accuracy and validity of the numerical results, careful checks of the grid independence of the numerical solutions have been carried out by considering four grid systems with large number of grid points (around 2.0 million mixed cells) at a Reynolds number of 200,000. It is found that the relative deviation of the Nusselt numbers for 2.0M cells and 3.0M cells are 3% and 1.9% for Case I and Case II, respectively. Thus, to save computer resources and keeping a balance between computational economy and prediction accuracy, the grids of mixed 2.0M cells have been chosen.

RESULTS AND DISCUSSION

Parameter Definitions: Before analyzing the effect of guide vanes on the heat transfer enhancement and channel pressure drop, the Nusselt number and friction factor definitions should be presented. First, the Fanning friction factor f is defined as

$$f = \frac{\Delta p}{2\rho u_i^2} \cdot \frac{D_h}{L} \tag{8}$$

where u_i is the inlet velocity, L is the two-pass channel total length, and D_h is hydraulic diameter based on the inlet channel.

The tip-wall averaged Nusselt number can be calculated in the following way. The local Nusselt number of every cell vertices is first calculated as

$$Nu(i) = \frac{q_w}{T(i) - T_f} \cdot \frac{D_h}{\lambda} \tag{9}$$

where T_f is the mass-weighted average temperature between the inlet and outlet fluid temperatures, and $T(i)$ is the local surface temperature. The overall Nusselt number is determined by averaging of all local Nusselt numbers on the tip-wall surface.

Model Validation: Because no related experimental data of two-pass channels with guide vanes is available, the turbulent flow and heat transfer in the smooth two-pass channel have been computed so that the averaged Nusselt number and pressure drop have to be compared with those in the experiments [19]. It was found that the Nusselt number deviation between the simulations by the realizable k-ε turbulence model and the experimental data was less than 15% and the smallest deviation was about 7%. Thus, the overall agreement between predicted and experimental results ensures the reliability of the physical model and computational method, see also [20-24].

Velocity and Temperature Fields: Figure 5 shows the cross-sectional velocity vectors in the turn region of the three channels. As the fluid flows through the 180-deg turn, the centrifugal forces arising from the streamline curvature and the pressure gradient (low pressure at inner wall while high pressure at outer wall) produces a pair of counter-rotating vortices in the turn. These vortices are significantly strong and are responsible for the transport of cold fluid from the core towards the outer wall. For the two-pass channels with guide vanes, the pair of counter-rotating vortices in the turn is compressed and moves downwards close to the tip-wall. This leads to improved turbulent mixing of the approaching cold fluid with the hot fluid near the tip-wall. Accordingly, the heat transfer over the internal tip-wall is increased due to the continuous mixing. Another feature for Case I is that a large recirculating flow region is produced behind the long-length guide vane. Such recirculation will result in an additional large pressure loss as expected.

Figure 6 shows the tip-wall temperature contours of the three channels. It can be observed that for the channel without guide vanes and with one guide vane (Case II), due to the cold fluid impinging towards the tip-wall, the temperature of the first tip part is lower than that of the second tip part, while for the channel with two guide vanes (Case I), due to the long-length guide vane most impinging flow is guided towards the second tip part and hence lower temperature is resulted. High temperature occurs at the side corners of the second-pass due to the recirculating flow. Besides, the low temperature region is enlarged to certain extent because of the presence of guide vanes. Especially, the presence of two guide vanes (Case I), provides better performance with lower temperature region, as clearly shown in the figure. At this point, the tip-wall heat transfer of Case I is expected to be higher than that of Case II.

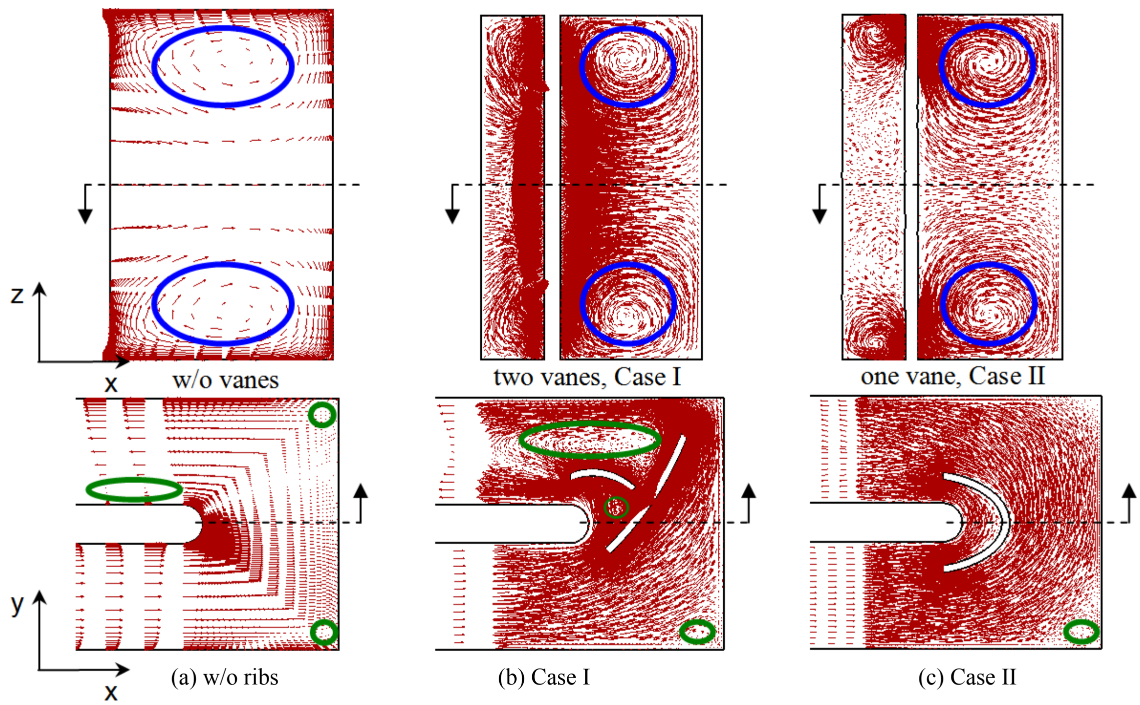
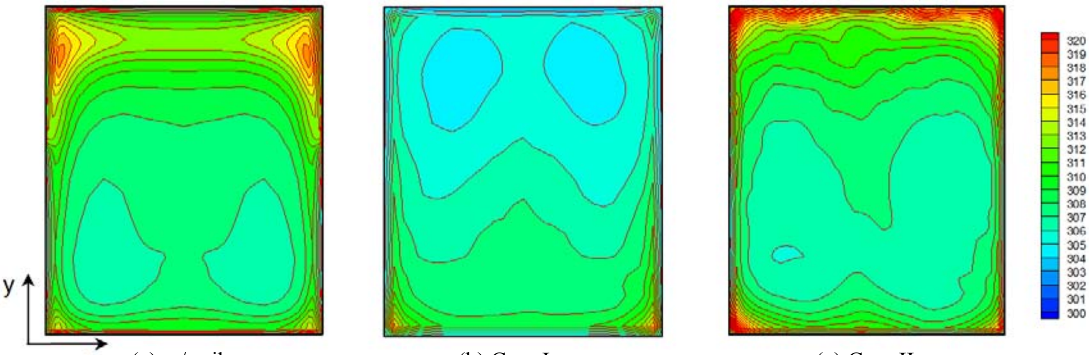


Fig. 5 Flow fields, Re=200,000

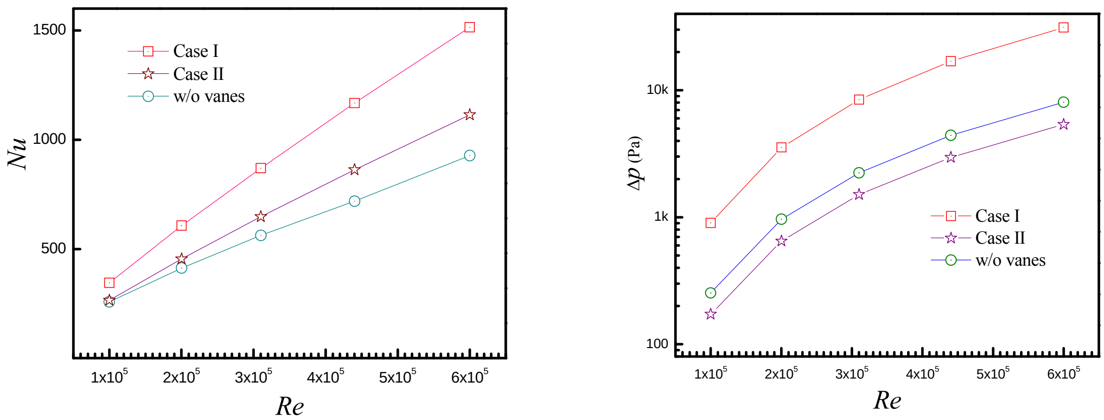


(a) w/o ribs (b) Case I (c) Case II

Fig. 6 Temperature contours on the tip, Re=200,000

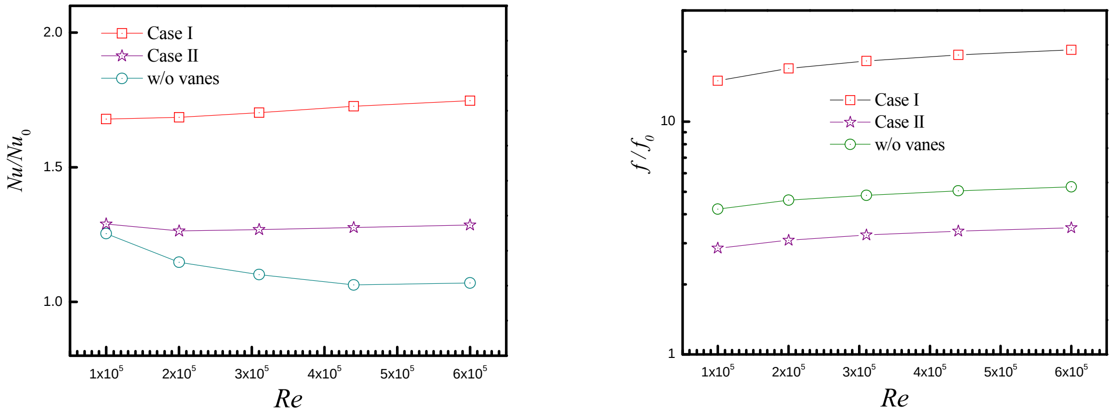
Friction Factor and Heat Transfer: Figure 7 presents the surface averaged Nusselt number and inlet-to-outlet pressure drop for all Reynolds numbers. For the heat transfer in Fig. 7(a), the overall tip-wall Nusselt numbers of the channels with guide vanes, Case I and Case II, are about 10%-60% higher than those of the channel without guide vanes. At high Reynolds numbers, Case I offers the larger heat transfer enhancement. For the pressure loss in Fig. 7(b), the pressure drop of Case I is around 3.5 times higher than that of the channel without guide vanes, while Case II shows around 35% smaller pressure drop compared to the channel without guide vanes. (A similar pressure drop reduction was found in [12] if properly designed turning vanes were used.) Figure 8 provides the averaged Nusselt number and friction factor as normalized by the corresponding values for fully developed turbulent flow and heat transfer based on inlet channel conditions (identical Reynolds numbers). The values for Nu_0 and f_0 of fully developed channel flow are obtained from $Nu_0 = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.3}$, $f_0 = (0.79\ln\mathrm{Re} - 1.64)^{-2}/4$.

From the figure, the heat transfer enhancement of the smooth two-pass channel over the fully developed channel lies between 1.07 and 1.25, while the heat transfer enhancement of the two-pass channels with guide vanes over the fully developed channel lies between 1.26 and 1.75. The increased factors of the pressure drop of the two-pass channel without and with guide vanes over the fully developed channel lie between 4.2 and 5.2 and between 2.8 and 20, respectively. Examples of typical distributions of the local heat transfer enhancement of Case I and Case II at Re=200,000 are plotted in Fig. 9. It is distinctly observed that at the first-pass tip region similar heat transfer enhancement is produced by Case I and Case II, while at the second-pass tip region much higher local heat transfer enhancement is achieved near the edge (see the line for z/W=0.05). Compared to Case II, Case I presents higher local heat transfer enhancement at most regions, resulting in higher averaged heat transfer enhancement.



(a) Nusselt number (b) Pressure drop, Pa

Fig. 7 Heat transfer and pressure drop



(a) Nusselt number ratio (b) Friction factor ratio

Fig. 8 Normalized Nusselt number and friction factor

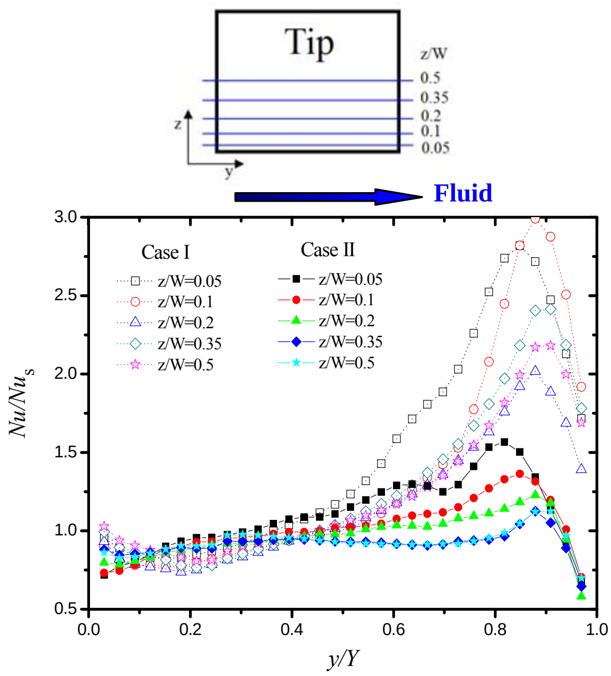
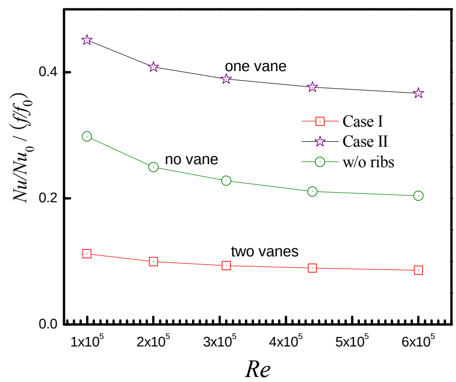


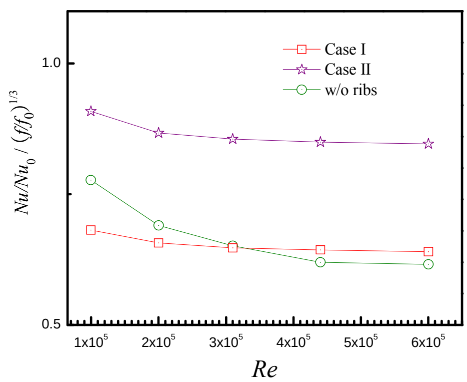
Fig. 9 Typical profiles of local heat transfer enhancement Re=200,000

OVERALL COMPARISON

From the above simulated results, it is found that the two-pass channel with two guide vanes provides higher Nusselt number associated with higher pressure drop than the channel with one guide vane. Accordingly, it is essential to compare the heat transfer enhancement performance of the two channels with guide vanes. Figure 10 presents a comparison of the averaged Nusselt number ratio divided by the normalized friction factor ratio for the three channels. $Nu/Nu_0/(f/f_0)$ parameter is referred to as the Reynolds analogy performance parameter, and $Nu/Nu_0/(f/f_0)^{1/3}$ parameter is to provide a heat transfer augmentation quantity. It relates the heat transfer augmentation Nu/Nu_0 and the friction factor increase f/f_0 under the same ratio of mass flux in a channel with augmentation devices to a channel with smooth surfaces. From Fig. 10, it is found that higher values of the parameter $Nu/Nu_0/(f/f_0)$ and $Nu/Nu_0/(f/f_0)^{1/3}$ are produced by case II at all Reynolds numbers from 100,000 to 600,000, while Case I provides much smaller values of $Nu/Nu_0/(f/f_0)$ than the smooth channel without guide vanes. Beyond Reynolds number of 300,000, Case I provides slightly higher values of $Nu/Nu_0/(f/f_0)^{1/3}$ than the smooth channel. This indicates that below Reynolds number of 300,000. no benefit can be achieved when such kinds of guide vanes are designed and applied.



(a) $Nu/Nu_0/(f/f_0)$



(b) $Nu/Nu_0/(f/f_0)^{1/3}$

Fig. 10 Nusselt number ratio normalized by friction factor ratio

Figure 11 presents another comparison of the heat transfer coefficients subject to the required pumping powers for the three channels. From this figure, it is found that at identical required pumping power Case II offers the highest heat transfer coefficient. This indicates that Case II provides good performance for modifying the flow structure and improving gas turbine blade tip cooling at the same pumping power.

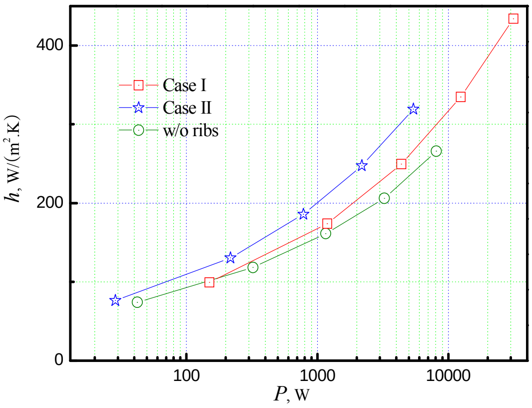
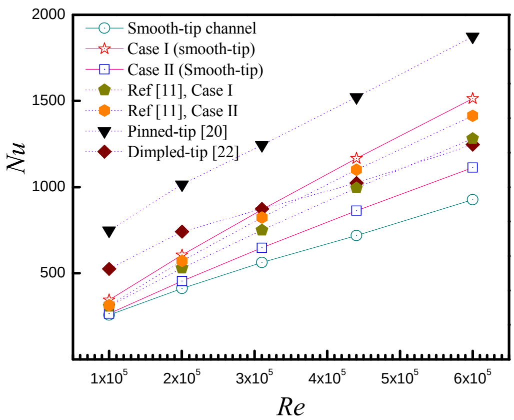
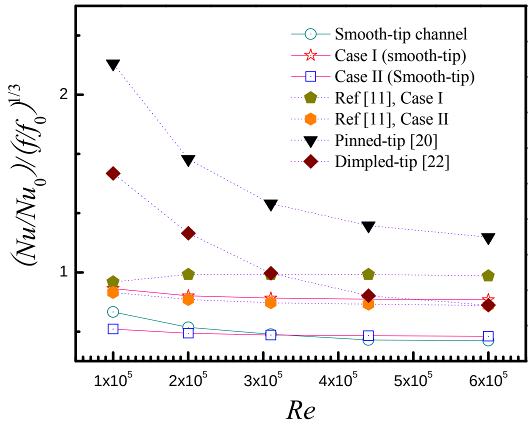


Fig. 11 Heat transfer coefficient versus pumping power

Figure 12 presents comparisons of the tip-wall heat transfer by different kinds of heat transfer augmentation devices. In the figure, the data of pinned-tip and dimpled-tip is adopted from the previous works [19-23], where physical models are such that pins or dimples are directly placed on the internal tip-wall. That is, in those cases the tip-wall itself is augmented. Also, data from cases having guide ribs placed in the turn region is adopted [11]. From the figure, it is found that at identical inlet flow conditions (identical Reynolds number), the tip-wall heat transfer of the presented cases is lower than those of pinned-tip or dimpled-tip two-pass channels. This suggests that the heat transfer augmentation by the guide vanes is not significant compared to that by pins or dimples imposed on tip-wall directly, especially at low Reynolds numbers. However, the main motivation of designing guide vanes is to modify and improve the flow structure locally so as to reduce the overall pressure loss, as shown in Case II of the present study.



(a) Heat transfer



(b) Thermal performance

Fig. 12 Comparison of tip-wall heat transfer

CONCLUSIONS

Simulations of tip-wall heat transfer and pressure drop in two-pass channels with and without guide vanes have been performed numerically. The Reynolds number based on the inlet velocity and inlet hydraulic diameter is ranging from 100,000 to 600,000. The overall performance of the simulated channels is also evaluated and compared. The main findings from this study are summarized as follows:

(1) The guide vanes compress and force the vortices towards the tip wall and thereby improve the turbulent mixing of the approaching cold fluid and hot fluid near the tip.

(2) The presence of guide vanes provides 10%~60% higher tip-wall heat transfer over that of the channel without guide vanes. Guide vanes can reduce the pressure loss when they are designed and placed properly. In the present study, the arc-like guide vane provides good overall performance.

(3) The augmentation of tip-wall heat transfer due to the presence of guide vanes is not significant compared to those by augmentation devices imposed tip-wall directly.

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REFERENCES

[1] Schabacker, J., Boelcs, A. and Johnson, B. V., 1998, “PIV investigation of the flow characteristics in an internal coolant passage with two ducts connected by a sharp 180º bend,” *ASME Paper No. 98-GT-544*.

[2] Schabacker, J., Boelcs, A. and Johnson, B. V., 1999, “PIV investigation of the flow characteristics in an internal coolant passage with 45deg rib arrangement”, *ASME Paper No. 99-GT-120*.

[3] Son, S. Y., Kihm, K. D., and Han, J. C., 2002, “PIV flow measurements for heat transfer characterization in two-

pass square channels with smooth and 90° ribbed walls," *Int. J. Heat Mass Transfer*, vol. 45, pp. 4809-4822.

[4] Rathjen, L., Hennecke, D. K., Bock, S., and Kleinstueck, R., 2001, "Detailed heat/mass transfer distributions in a rotating two pass coolant channel with engine-near cross section and smooth walls," *Heat Transfer in Gas Turbine Systems*, R. Goldstein, ed., New York Academy of Sciences, New York; 2001, Ann. N.Y. Acad. Sci., 934, pp. 432–439.

[5] Liou, T. M., Lee, H. L., and Liao, C. C., 2001, "Effects of guide-vane number in a three-dimensional 60-deg curved side-dump combustor inlet," *ASME J. Fluids Engineering*, vol. 123, pp. 211-218.

[6] Liou, T. M., Lee, H. L., and Liao, C. C., 2001, "Effects of inlet guide-vane number on flowfields in a side-dump combustor," *Experimental Thermal and Fluid Science*, vol. 24, pp. 11-23.

[7] Ratna Rao, D. V., Babu, C. S., and Prabhu, S. V., 2004, "Effect of Turn Region Treatments on the Pressure Loss Distribution in a Smooth Square Channelwith Sharp 180◦ Bend," *Int J Rotating Machinery*, vol. 10, no.6, pp. 459–468.

[8] Ratna Rao, D. V., and Prabhu, S. V., 2003, "Effect of turn region treatments on the pressure loss through sharp 180 deg bends," *AIAA 1st International Energy Conversion Engineering Conference*, 17-21 August 2003, Portsmouth, Virginia. Paper no. AIAA 2003-5961

[9] Babu, C. S., Ratna Rao, D. V., and Prabhu, S. V., 2003, "Effect of guide vanes on pressure drop in a rib roughened square channel with a sharp cornered 180^0 bend," *AIAA 41st Aerospace Sciences Meeting and Exhibit,* 6-9 January 2003, Reno, Nevada. Paper no. AIAA 2003-1145

[10] Luo, J., and Razinsky, E. H., 2009, "Analysis of turbulent flow in 180 deg turning ducts with and without guide vanes," *ASME J. Turbomachinery,* vol. 131, paper no. 021011.

[11] Xie, G.N., Sundén, B., Zhang, W.H., Wang, L., and Utriainen E., 2010, "Analysis of enhanced heat transfer on a turbine blade tip-wall with and without guide ribs," *Proceedings of 2010 ASME International Mechanical Engineering Congress and Exposition*, IMECE2010, November 12-18, 2010, Vancouver, British Columbia.. Paper no IMECE2010-37302.

[12] Schuler, M., Zehnder, F., Weigand, B., von Wolfersdorf, J., and Neumann, S. O., 2011, "The Effect of Turning Vanes on Pressure Loss and Heat Transfer of a Ribbed Rectangular Two-Pass Internal Cooling Channel," *ASME J. Turbomachinery,* vol. 133, paper no. 021017.

[13] Han, J. C., Chandra, P. R., and Lau, S. C., 1988, "Local heat/mass transfer distributions around sharp 180-deg turns in two-pass smooth and rib- roughened channels," *ASME J. Heat Transfer*, 110, pp. 91–98.

[14] Wagner, J. H., Johnson, B. V., and Kopper, F. C., 1991, "Heat transfer in rotating serpentine passages with smooth walls," *ASME J. Turbomachinery*, vol. 113, pp. 321–330.

[15] Wagner, J. H., Johnson, B. V., Graziani, R. A., and Yeh, F. C., 1992, "Heat Transfer in rotating serpentine passages with trips normal to the flow," *ASME J. Turbomachinery*, vol. 114, pp. 847–857.

[16] Bunker, R. S., 2001, "A review of turbine blade tip heat transfer," *Heat Transfer in Gas Turbine Systems*, R. Goldstein, ed., New York Academy of Sciences, New York; 2001, Ann. N.Y. Acad. Sci., 934, pp. 64-79.

[17] Bunker, R. S., 2006, "Axial turbine blade tips: function, design and durability," *AIAA J. Propulsion and Power*, vol. 22, no. 2, pp. 271-285.

[18] Sundén, B., and Xie, G. N., 2010, "Gas turbine blade tip heat transfer and cooling: a literature survey," *Heat Transfer Engineering*, vol. 31, no. 7, pp. 527-554.

[19] Bunker, R. S., "The augmentation of internal blade tip-cap cooling by arrays of shaped pins," *ASME J Turbomachinery*, 2008, vol. 123, paper no. 041007.

[20] Xie, G. N., Sundén, B., Wang, L., and Utriainen, E., 2009, "Enhanced heat transfer on the tip-wall in a rectangular two-pass channel by pin-fin arrays," *Numerical Heat Transfer-part* A , vol. 55, pp.739-761.

[21] Xie, G. N., Sundén, B., Utriainen, E., and Wang, L., 2010, "Computational analysis of pin-fin arrays effects of internal heat transfer enhancement of a blade tip-wall," *ASME J. Heat Transfer*, 132, paper no. 031901.

[22] Xie, G. N., Sundén, B., Wang, L., and Utriainen, E., 2009, "Augmented heat transfer of an internal blade tip by full or partial arrays of pin-fins," *Int. Symp. on Heat Transfer in Gas Turbine Systems*, 9-14 August 2009, Antalya, Turkey. Paper no THT-52.

[23] Xie, G. N., Sundén, B., and Wang, Q.W., 2010, "Predictions of enhanced heat transfer of an internal blade tip-wall with hemispherical dimples or protrusions," *Proceedings of ASME Turbo Expo 2010: Power for Land, Sea and Air, GT2010*, June 14-28, 2010, Glasgow, UK. Paper no GT2010-22265.

[24] Xie, G. N., Sundén, B., and Zhang, W. H., 2010, "Comparisons of pins/dimples/protrusions cooling concepts for an internal blade tip-wall at high Reynolds numbers," *International Heat Transfer Conference, IHTC-14, August 8-13, 2010, Washington D.C., USA*. Paper no IHTC14-22284.

[25] Shih, T.-H., Liou, W. W., Shabbir A., Yang Z., and Zhu J., 1995, "A new k-ε eddy-viscosity model for high Reynolds number turbulent flows-model development and validation," *Computers Fluids*, vol. 24, no. 3, pp. 227-238.