CHAPTER 3

Heat Transfer Enhancement of a Gas Turbine Blade-Tip Wall

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Abstract

The blade-tip region encounters high thermal loads because of the hot gas leakage flows. A common way to cool blade tips is to design serpentine passages with 180° turn with the blade-tip walls inside the turbine blades. Improved internal convective cooling is therefore required to increase blade-tip lifetime. This chapter summarizes the heat transfer enhancement of an internal blade-tip wall having pins, dimples and protrusions, and the influence of guide ribs and pins material. It is found that the pinned tip exhibits best performance to improve the blade-tip cooling. However, when considering the added mechanical stress, making dimples is a more suitable way especially at low Reynolds numbers. Properly designed guide ribs might reduce the pressure loss.

Keywords: Blade-tip wall; heat transfer enhancement; numerical simulation

1 Introduction

The heat load transferred to a gas turbine blade is continuously increased with the increase of the turbine inlet temperature. The blades can only survive if effective cooling methods are used to remove the heat load from the turbine blades. Cooling of the blade should include the cooling of all regions exposed to high temperature gas and thermal load. The blade-tip area is such a region particularly for high-pressure turbines. The hot leakage flow increases the thermal loads on the blade tip, leading to high local temperature. It is therefore very essential to cool
the turbine blade tip and the region near the tip. A common way to cool the blade tip is to adopt internal cooling by designing serpentine (two-pass, three-pass, or multi-pass) channels with a $180^\circ$ turn/bend inside the blade (as shown in Fig. 1). Taking advantage of impinging and turning effects, the tip can be cooled to a certain extent. To augment forced convection, advanced methods of enhancing the internal convective cooling for the tip region are required. The blade lifetime and reliability will then be improved. Several publications reviewing the gas turbine heat transfer and cooling technology research can be found [1–7].

It is well recognized that many heat transfer augmentation devices (e.g. fins, ribs, pins, dimples, and protrusions) can be used to improve the intensity of heat transfer. Dimples and protrusions are arrays of extended surfaces that are indented or protruded on primary surfaces or smooth walls in a channel, as shown in Figs 2c and 3d. They are often spherical in shape. Like ribs and pins, it is expected that dimples and protrusions can generate strong vortices when they are placed in a flow and then higher turbulence is produced around them. Many previous research works have shown that pins, dimples, and protrusions can provide substantial heat transfer enhancement in confined low aspect ratio channels. Although it is well known that ribs and pins can significantly augment heat transfer over primary or smooth surfaces, the augmentation is always penalized by the large additional pressure loss. For example, Chyu [8] experimentally measured mass/heat transfer enhancement of up to 4.0 for circular staggered pins at high Reynolds numbers but with a friction coefficient increase of around 70 compared with that of a fully developed smooth flow channel. Bailey and Bunker [9] obtained heat transfer enhancement of up to 3.6 for ribbed channels with very high blockage ratios.

![Figure 1: A typical serpentine passage inside a turbine blade.](image1)

Figure 1: A typical serpentine passage inside a turbine blade.

![Figure 2: Heat transfer augmentation devices for a smooth surface.](image2)

(a) smooth-wall (b) pin-wall (c) Dimple-wall (d) Protrusion-wall

Figure 2: Heat transfer augmentation devices for a smooth surface.
HEAT TRANSFER ENHANCEMENT OF A GAS TURBINE BLADE-TIP WALL

(rib height to channel height) of approximately 27.5–47.5\% over a smooth channel without ribs. However, this enhancement was accompanied by a friction coefficient increase of as much as 65. By comparing several types of heat transfer augmentation techniques for internal cooling of turbine airfoils, Ligrani et al. [10] concluded that except for swirl chambers, dimples exhibited better overall performance than pins, rib turbulators, and other surface roughnesses.

During recent years, various pins, dimples, or protrusions have received much attention for enhancing heat transfer in internal cooling passages of turbomachinery. Numerous experimental and numerical studies of heat transfer and pressure drop for shaped pins, dimples, or protrusions are available [11–22]. Metzger et al. [11] studied the developing heat transfer in rectangular ducts having short pin-fin arrays. They found that the heat transfer coefficient gradually increased to the first three or four rows followed by a gradual decrease through the remaining rows. Lau et al. [12] experimentally studied the effect of the pin configuration on the local endwall heat/mass transfer in pin-fin channels. They observed that the decrease of the streamwise pin space led to an increase of the endwall heat/mass transfer. Chyu et al. [13] reported that the pin-fin heat transfer was approximately 10–20\% higher than the endwall heat transfer. Goldstein et al. [14] found that the stepped-diameter circular pin-fin array provided a higher mass transfer coefficient and a smaller pressure loss than a uniform-diameter circular pin-fin array. Chyu et al. [15] experimentally studied the heat transfer on staggered dimpled surfaces using the transient liquid crystal method. They reported that the heat transfer ratios everywhere on the dimpled surface were higher than those of the smooth channels, and the largest factor of heat transfer enhancement was more than 2.5 located downstream the area of the dimples. Moon et al. [16] used a similar experimental method to investigate rectangular channels with dimples imprinted on one wall. Results showed that a heat transfer enhancement of 2.1 times could be achieved with approximately 1.6–2.0 times pressure drop penalty compared with the smooth surfaces. Moon and Lau [17] experimentally reported average heat transfer coefficients on a wall with concave and cylindrical dimples in a square channel. They found that the cylindrical dimples offered higher heat transfer enhancement with lower pressure drop than the concave dimples with identical diameter and depth. Griffith et al. [18] measured averaged Nusselt numbers in a rotating rectangular channel with dimples placed on both sides. They found that the rotation could augment the heat transfer on dimpled surfaces, especially on the trailing dimpled surface toward which the colder fluid core migrates due to the Coriolis forces. Ligrani et al. [19] performed flow visualization, pressure, and velocity measurements on the flow structure of a dimpled wall. They observed a primary vortex pair shed periodically from the center of each dimple, and two additional secondary vortex pair formed near the spanwise edges of each dimple. Mahmood et al. [20] and Ligrani et al. [21] measured local and spatially Nusselt number for the dimpled surface with and without protrusions on the opposite wall. Results showed that for the dimple-protrusion channel, Nusselt numbers were augmented considerably because of the added vortical, secondary flow structures and flow unsteadiness produced by the protrusions. Hwang et al. [22] used transient Thermochromic liquid crystal (TLC) technique to measure the
local heat transfer coefficients on the dimple or protrusion walls at low Reynolds numbers varying from 1,000 to 10,000. They found that a double protrusion channel wall provided the highest average heat transfer rate associated with the highest pressure drop. At low Reynolds number of 1,000 the heat transfer enhancement factor was up to 14. However, in the high Reynolds number range the double protrusion channel wall offered lower performance factor than other cases: double dimple wall, simple protrusion wall, simple dimple wall. Park et al. [23] predicted turbulent structures above a dimpled surface using FLUENT with the realizable $k$–$\varepsilon$ model without wall functions. They claimed that due to the advection of reattaching and recirculating flow within the dimple cavities as well as the strong instantaneous secondary flow and mixing, the vortex pair contains augmented magnitudes of eddy diffusivity for momentum and heat.

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 occur at the corners in the turn region and downstream the turn along the inner wall, as depicted in Fig. 3. Particle image velocimetry (PIV) measurements in [24] show a strong Dean-type secondary flow and a large separation bubble on the inner wall downstream of the bend exit. The secondary flow consisting of two counter rotating vortices caused a strong impingement of the flow on the outer walls at the bend exit. Schabacker et al. [25] 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. [26] also measured the flow field and heat transfer inside a two-pass square channel with a smooth wall and a 90° 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, to modify or improve the flow structure with low heat transfer involved, properly designed guide vanes/ribs are suggested to be placed in appropriate regions.

However, there have been only a few studies on effects of guide vanes (turning ribs) inside serpentine channels published in the open literature. Rathjen et al. [27] studied mass/heat transfer in a smooth two-pass channel with a 90° guide vane. The intention was to avoid the flow separation zone. Liou et al. [28,29] experimentally studied the effects of the number of guide vanes on the 3D flow field in a 60° curved combustor inlet. It was found that by adding three guide vanes, the flow

![Conceptual flow pattern around a 180° turn.](image)
separation in the curved combustor could be eliminated completely. 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. [30, 31] and Babu et al. [32] investigated the effects of guide vanes on pressure drop in smooth or roughened channels with a 180° bend. Experimental results showed that pressure losses were significantly affected by the shape and position of the guide vanes. 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 [33] performed numerical simulation of the turbulent flows inside several 2D and 3D 180° U ducts with and without guide vanes by a Reynolds-Average-Navier-Stokes (RANS) method. Results showed that the combined vane and uniform cross-sectional area led to much weaker secondary-flow vortices and smaller separation, which accounted for a substantial reduction in pressure loss.

For blade-tip heat transfer, little information is available. In the earlier study by Han et al. [34], 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 were measured by the naphthalene sublimation technique. The typical averaged tip heat transfer of smooth and ribbed channels at Reynolds number of 30,000 is 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. [35,36] measured the heat transfer in nonrotating and rotating multipass channels with and without turbulators placed in 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 nonrotating 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 were 25,000 and 0.24, respectively.

For serpentine channels with 180° turn, turbulent mixing can be strengthened effectively, and therefore, the heat transfer can be augmented on the concerned surfaces. Bunker [37] presented a method to increase convective heat flux on an internal cooled blade-tip cap, where arrays of shaped pins were placed. 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 was negligible compared with that of a smooth surface. From the aforementioned articles, it was clear that most previous studies were concerned with the heat transfer on the leading or/and trailing walls of the two-pass channels, and very limited information is available for tip walls. Thus, it is desirable to explore effective cooling techniques for the internal tip, and to present more details on the heat transfer enhancement. For these reasons, the main objective of the present chapter is to provide a comprehensive summary of the heat transfer enhancement over the tip-wall in a rectangular two-pass channel at high Reynolds numbers, up to 600,000. Typical results are provided to show the tip cooling performance for a variety of cases. More details can be found in recently published papers [38–46].
2 Physical Models of Eight Kinds of Tip-Cooling Concepts

The geometrical models considered in this chapter are schematically shown in Fig. 4. The two-pass channels have rectangular cross section with an aspect ratio of 1:2 with a hydraulic diameter of 93.13 mm. The full tip-wall cap section is 139.7 mm × 165.1 mm, and the tip clearance (the distance between divider to tip cap) is 88.9 mm. The lengths of the inlet channels (first pass) and the outlet channels (second pass) are about ten hydraulic diameters. It is worthwhile to note that the considered models of the rectangular two-pass channels are similar to those in the experiments by Bunker [37], while the internal tip cap is augmented by circular pins, hemispherical dimples, or protrusions.

Case 0: As shown in Fig. 4a, no augmentation devices are placed on the tip wall, first pass, or second pass. This means that all the walls are smooth. A smooth-tip two-pass channel is used as a baseline for performance comparison.

Case 1F: As shown in Figs 4b and g, the internal tip cap is almost completely occupied by a full array of circular pin-fins. The pin arrangement is shown in Fig. 4g. The pins have a height of 8.128 mm and a diameter of 4.064 mm. The streamwise pitch is 21.117 mm (y direction) and the spanwise pitch is 12.192 mm (z direction). The total number of pin fins is 165. The tip (including endwall and pin-fins) surface wetted area enhancement is 1.74 compared with the smooth-tip surface area. In order to provide a similar model to that used in Bunker’s experiments [37], in which a uniform heat flux was prescribed on the external smooth tip-wall, a solid plane tip with a certain thickness is placed below the internal pin-finned tip cap and a uniform heat flux is supplied there. The turbulent convection between the fluid and solid, and heat conduction within the solid parts are
conjugated in the computations [43]. In this case, the pin fins are made of aluminum with the density of 2,719 kg/m³ and the thermal conductivity of 202.4 W/(m K).

*Case 1P:* As shown in Figs 4b and h, a partial array of circular pin fins is symmetrically arranged close to both sides of leading/trailing walls. Except for the distributions, other parameters of the channels and pins are completely identical to that of Case 1F. The area enhancement factor is 1.21.

*Case 1FM:* This case is similar to the Case 1F except that the pin fins are made of insulating material (Renshape) with a density of 1,800 kg/m³ and thermal conductivity of 1 W/(m K). Thus, the area enhancement factor is 1.0, that is, no effective heat transfer area increase is provided. Other materials of the pin fins have also been considered, see [41].

*Case 2:* As shown in Fig. 4c, an array of dimples is pressed on the internal tip cap. The dimples have a height (depth) of 2.032 mm and a print diameter of 4.064 mm. The arrangement is the same as the pin arrangement of Case 1F and hence the number of dimples is identical. The area enhancement factor is 1.092.

*Case 3:* As shown in Fig. 4d, an array of protrusions is located on the internal tip cap. The layout and dimensions of the spherical protrusions are identical to those of the dimples except that the protrusions are located above the internal tip-cap surface like flow obstacles. Accordingly, the protruded tip has the same area enhancement factor (i.e. 1.092).

*Case 4:* As shown in Fig. 4e, four guide ribs are placed in the turn region (due to the down view of the 3D geometry only two guide ribs are shown in the figure, for the side view of geometry, the ribs are symmetrically placed at the bottom and on the upper walls as shown in Fig. 4i). The ribs have a height of 12.7 mm and a thickness of 5.08 mm. Thus, the ribs have a cross-sectional blockage ratio (2e/H) of 0.182. The tip wall is still kept small.

*Case 5:* As shown in Fig. 4f, two arc-like guide ribs are placed in the turn region. The ribs have the same height and thickness as Case 4.

In reality, it might be hard to manufacture the pin fins and fix them to the internal tip surface as perfect circular cylinders. In the experiments by Bunker [37] this was also the case, which means that the pin fins had some base fillets closest to the tip surface. This will be commented on later.

This chapter aims to summarize the heat transfer enhancement and pressure drop of two-pass channels, and then compare the overall performance to provide useful information for blade-tip cooling-design concepts.

### 3 Computational Details

#### 3.1 Selection of turbulence model

The selection of a turbulence model for the simulations requires consideration of computational cost, predicted flow phenomena, and acceptable accuracy. This chapter mainly concerns tip-wall heat transfer, and therefore the agreement between the calculated Nusselt number and experimental data is the main criterion for selecting turbulence models. Six different turbulence models, that is, the
standard \( k-\varepsilon \) model, the Renormalization Group (RNG) \( k-\varepsilon \) model, the realizable \( k-\varepsilon \) model, the shear stress transport (SST) \( k-\omega \) model, the \( v^2f \) model, and the Reynolds stress model (RSM) were chosen to simulate the turbulent heat transfer of the smooth two-pass channel (as shown in Fig. 4a). Figure 5 compares the experimentally obtained and the predicted average Nusselt number at a Reynolds number of 200,000 with the different turbulence models. From this figure, the RNG \( k-\varepsilon \) model results approach the experimental data slightly better than the other tested turbulence models. The smallest difference of 14.7\% is found by the \( k-\varepsilon \) RNG model and the largest difference of 48\% is found by the standard \( v^2f \) model. Actually, the three \( k-\varepsilon \) models provide very similar results. The test computations showed that by using the realizable \( k-\varepsilon \) model, the CPU time can be shortened and the convergence behavior is better than for the other two models, especially for computing cases of pin-finned channels. The computed final results, which will be presented in this chapter, are all based on the realizable \( k-\varepsilon \) turbulence model. The corresponding governing equations are presented in the following sections and may also be found in the FLUENT documentation [47].

Standard wall functions for the realizable \( k-\varepsilon \) model are applied on the walls for the near-wall treatment because the Reynolds numbers are high and the geometrical models are complicated. Most of the \( y^+ \) values on the tip walls lie between 15 and 50.

### 3.2 Grid dependence

The preprocessor software GAMBIT was used to generate the grids. Figure 6 shows typical surface grids of Cases 1, 3, and 5. Because of the complex geometries, a multiblock grid technique was used to generate grids by dividing the whole domain into several subdomains. In the turn region due to the many small circular

**Figure 5:** Comparison of predicted Nusselt numbers by six turbulence models \((Re = 200,000)\). Experimental data from [37].
pins/dimples/protrusions or guide ribs, an unstructured grid has to be filled into the domain. This domain occupies about 95% of all cells.

To ensure the accuracy and reliability of the numerical results, several careful checks of the grid effect of the numerical solutions had to be carried out before the final calculations. Several candidate grid systems with large number of grid points were considered. For example with the pin-finned channel, four grid systems, that is, 989,996 cells, 1,889,055 cells, 2,494,201 cells, and 3,265,705 cells at a Reynolds number of 200,000 were tested. It was found that the relative deviation of both Nusselt numbers and pressure drops between 2.49M cells and 3.26M cells was 3.33% and 0.08%, respectively. Thus, to save computer resources and keeping a balance between computational economy and prediction accuracy, the grid with 2.49M cells has been chosen. Similarly for the other cases, grid systems of around 2.1M–2.4M cells have been chosen, respectively. These simulations are performed on a PC with two CPUs having a frequency of 3.0 GHz and a core memory of 8G. Typical running times for computation of one case are about 12 and 48 hours for the smooth-tip channel and the augmented tip channels, respectively.

3.3 Boundary conditions

Although the heat transfer on the leading and trailing walls is important for gas turbine blade design, the tip-wall heat transfer is a major concern. Therefore, in order to approach the experimental conditions by Bunker [37], in which a uniform heat flux was created by a heater, a constant heat flux is prescribed on the bottom walls while the remaining walls are assumed to be adiabatic. No-slip velocity conditions are applied at all walls. Uniform inlet velocity and temperature, that is, 300 K, are assigned at the inlet and an outflow condition is chosen at the outlet.

The fluid is assumed to be incompressible with constant thermal physical properties and the flow is assumed to be three dimensional, turbulent, steady, and nonrotating. Although for augmented straight passages the flows might behave unsteady, it is believed that, for the present augmented-tip two-pass channels, unsteady flow will not occur as the pins, dimples, protrusions, and guide ribs are assumed to stabilize the flow to some extent. The working fluid is dry air. The minimum convergence criterion for continuity, momentum quantities error, and $k$ and $\varepsilon$ equations is $10^{-4}$, whereas it is $10^{-7}$ for the energy equation.
4 Fluid Flow and Heat Transfer Characteristics

4.1 Definitions of friction factor and Nusselt number

The pressure drop is transferred to the Fanning friction factor $f$ which is defined as follows:

$$f = \frac{\Delta p}{\frac{u_i^2}{2}D}$$

(1)

where $u_i$ is inlet velocity, $L$ is the two-pass channel total length.

The overall/averaged Nusselt number can be calculated in the following way. The local Nusselt number of every cell vertex is first calculated as

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

(2)

where $T_f$ is the mass-weighted average fluid temperature between the inlet and outlet fluid temperatures, and $T(i)$ is the local surface temperature. The overall/averaged Nusselt number is determined by area-weighted averaging of all local Nusselt numbers on the bottom wall, which is expressed as follows:

$$Nu = \frac{\sum A(i)Nu(i)}{\sum A(i)}$$

(3)

4.2 Model validation

To validate the computational approach, the turbulent heat transfer in the smooth two-pass channel was computed. The averaged Nusselt numbers and pressure drop of Case 0 are compared and listed in Table 1. It is found that the largest deviation in the Nusselt number between the simulations by the Realizable $k$–$\varepsilon$ turbulence model and the experimental data is less than 15% and the smallest deviation is about 7%. In the work by Bunker [37], the experimental uncertainty in the heat transfer coefficient was between 8% and 15%. The decent agreement between the predicted and tested results shows the reliability of the physical model. Thus, the validation and reliability of the computational approach were ensured.

<table>
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4.3 Flow and temperature fields

To facilitate a better understanding of the enhanced heat transfer over the tips, velocity vectors and temperature contours are presented. As an example, Fig. 7 shows the cross-sectional velocity vectors of Case 0 and Case 1F. It can be seen that for both channels, high-momentum fluid enters the turn from the inlet channel and impinges on the outer walls. Before the sharp turn, the velocity profiles are flat. Upon entering the turn, the flow accelerates near the inner wall while it decelerates near the outer wall. This distribution is reversed at the beginning of the second channel. When the coolant travels through the 180° turn of the two-pass channels, areas of impingement, separation, and recirculation are created. Because of the inability of the flow to follow the turn, a separation bubble appears downstream of the tip of the divider wall followed by a reattachment. Two circulating regions occur at the first-pass channel corner and the second-pass corner, respectively. As the fluid flows through the 180° turn, the centrifugal forces arising from the curvature and pressure difference (low pressure at the inner wall, while high pressure at the outer wall) produce 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 toward the outer wall. For the smooth-tip channel, only vortices remain on the upstream side of the second pass. For the pin-finned tip channel, the cores of the pair of counter-rotating vortices in the turn move downwards close to the pin-fins, and the zone of separation is larger than that of the smooth-tip channel. At the upstream side of the second pass, both the vortices and the separation bubble exist. The vortices force the cold fluid to impinge toward the pin fins and then induce turbulent mixing of the coming cold fluid with the hot fluid near the pin-fins and endwall. At this point, the heat transfer over the tip wall is increased due to the continuous mixing process.

Figure 7: Sectional flow fields at $Re = 200,000$ of Case 0 and Case 1F.
Figure 8 shows the tip-wall temperature contours of Case 0 and Case 1F. It can be observed that due to the cold fluid impinging toward the tip wall, the temperature of the first tip part is lower than that of the second tip part. By comparing Figs 8a and b, it is found that a much lower temperature region occurs at the bottom walls of the pin-finned tip channels due to the cold fluid impingement and pin-fin cross-flow. Besides, through the heat conduction in the pins and tips, a more uniform temperature is observed in the pin-finned tip channels compared with the smooth-tip channel. It is then expected that much higher Nusselt numbers and hence heat transfer are produced by the pin-finned tip channels. Similar distributions for the other considered cases are not presented.

4.4 Pressure drop and heat transfer

The surface-averaged Nusselt number and inlet-to-outlet pressure drop for all Reynolds numbers are summarized in Fig. 9. Concerning heat transfer, the heat transfer coefficients of Cases 1–5 are higher than that of Case 0. Among them, Case 1F offers the largest heat transfer followed by Case 1P. The averaged heat transfer coefficients of Case 2 and Case 3 are similar, and at high Reynolds numbers Case 5 offers higher heat transfer than Case 2 and Case 3. This indicates that a tip with pins can provide better heat transfer than a tip with dimples or protrusions. Case 1FM offers 10–60% higher heat transfer than Case 0. This indicates that even if the pins are made of an insulating material, the tip heat transfer can be enhanced by the pins especially at low Reynolds numbers.

For the pressure loss, the largest pressure drop is produced by Case 4, followed by Case 1F. Similar pressure drops are produced by Cases 1P, 2, and 3. The pressure drop of Case 5 is smaller than that of Case 0. This indicates that the pressure drop will be reduced when the guide ribs are designed properly otherwise the pressure drop will be severely increased. For the same geometry, the material property does not affect the pressure loss.
Figure 10 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 \( \text{Nu}_0 \) and \( f_0 \) of fully developed channel flow are obtained from \( \text{Nu}_0 = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} \) and \( f_0 = (0.79 \ln \text{Re} - 1.64)^{-2/4} \). It is found that the largest heat transfer enhancement is about a factor of 3.6, and the friction factor is increased by around 4–6 times.

![Figure 9: Nusselt number and friction factor.](image_url)
Examples of typical distributions of the local heat transfer enhancement of Case 1F, Case 2, and Case 5 at $Re = 200,000$ are plotted in Fig. 11. It is distinctly observed that slightly higher local heat transfer enhancement is produced at the first-pass tip region, while at the second-pass tip region much higher local heat transfer enhancement is achieved. Especially by adding the pins on the tip cap, the enhancement factor of the second-pass tip region lies between 2.5 and 4.5.
Figure 11: Profiles of local heat transfer enhancement.
5 Overall Comparisons of Tip-Cooling Concepts

The augmented two-pass channels provide higher Nusselt number associated with slightly higher pressure drop. Accordingly, it is essential to compare the heat transfer enhancement performance of the two-pass channels. Figure 12 presents a comparison of the globally averaged Nusselt number ratio divided by the normalized friction factor ratio for the three channels. $\frac{Nu}{Nu_0}/(ff_0)$ parameter is referred to as the Reynolds analogy performance parameter, and $\frac{Nu}{Nu_0}/(ff_0)^{1/3}$ parameter is to provide a heat transfer augmentation quantity. It relates the heat transfer augmentation $\frac{Nu}{Nu_0}$ and the friction factor increase $ff_0$ at the same ratio of mass flux in a channel with augmentation devices to a channel with smooth surfaces. From Fig. 12, it is found that compared to the smooth channel, all the channels with augmented devices provide higher values of the parameter $\frac{Nu}{Nu_0}/(ff_0)$ and $\frac{Nu}{Nu_0}/(ff_0)^{1/3}$ at all Reynolds numbers from 100,000 to 600,000. The pinned-tip two-pass channel, Case 1F, exhibits the best performance, followed by Case 5 at high Reynolds numbers.

Figure 13 presents a comparison of the heat transfer coefficient versus the required pumping power for the four channels. From this figure, it is found that at identical required pumping power the pinned-tip channels (Case 1F and Case 1P) and Case 5 at high Reynolds number offer a higher heat transfer coefficient, and thereby higher heat transfer enhancement. This indicates that the use of pins on the tip or adding guide ribs in the turn region could increase the heat transfer intensity significantly associated with low additional pressure losses.
Figure 14 provides an alternative comparison of the heat transfer enhancement of the augmented tips. The Nusselt number ratio of the tips of the augmented channels over the smooth channel is normalized by the active surface area enhancement factor of the tips. Since the heat transfer enhancement value is mostly attributed by the active surface area increase obtained from the augmented geometry, it is essential to evaluate the overall performance disregarding the area increase to describe
the ability of augmented heat transfer devices over the primary surface. From the figure, it can be clearly seen that the tips with hemispherical dimples or protrusions, Case 2 and Case 3, are superior to the tip with pin-fins when they are used to augment the smooth tip heat transfer at low Reynolds numbers. However, at high Reynolds number, Case 5 provides the highest value of this ratio. On the other hand, for practical operation, minimizing blade weight is an objective for designing rotating blades. The added weight implies possible larger stress, which results in reduced reliability and life.

6 Concluding Remarks

In order to explore the effective cooling technology for gas turbine blade tips, attempts have been tried to improve the tip heat transfer. In the presented investigations, scaled models of a rectangular two-pass channel with various augmented tip-caps have been simulated numerically by so-called Computational Fluid Dynamics (CFD) technique. The main conclusions are summarized as follows.

1. By adding pins/dimples/protrusions on the internal tip, the tip heat transfer can be increased about 2.0–3.0 times associated with less than 10% higher pressure loss. The intensity of heat transfer enhancement depends upon pins/dimples/protrusions configuration and arrangement.

2. Using augmentation devices, the heat transfer at the side edge regions can be improved, especially at the second-pass tip side region.

3. The larger the thermal conductivity of the pins is, the higher is the tip heat transfer. At low Reynolds numbers, the tip heat transfer is enhanced even by using insulating pins with a low thermal conductivity.
4. Base fillets may provide a small additional heat transfer increase without affecting the pressure drop significantly.
5. The presence of guide ribs in the turn regions can improve the tip heat transfer and the pressure loss might be reduced if the ribs are properly designed and placed.
6. Pins are preferable devices to augment the tip heat transfer when the added tip weight is acceptable for the mechanical design.
7. While disregarding the active area ratio or considering added weight ratio, it is found that the arc-like guide ribs have the best potential to improve the tip cooling.

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Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Greek symbols</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>tip-wall surface area</td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>Diameter of pin fins</td>
<td>ε</td>
</tr>
<tr>
<td>D_h</td>
<td>hydraulic diameter</td>
<td>λ</td>
</tr>
<tr>
<td>E</td>
<td>height of guide ribs</td>
<td>µ</td>
</tr>
<tr>
<td>F</td>
<td>Fanning friction factor</td>
<td>ρ</td>
</tr>
<tr>
<td>h</td>
<td>heat transfer coefficient</td>
<td>Δp</td>
</tr>
<tr>
<td>H</td>
<td>height of pin fins</td>
<td></td>
</tr>
<tr>
<td>L</td>
<td>length of two-pass channel</td>
<td></td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number</td>
<td>0 fully developed flow channel</td>
</tr>
<tr>
<td>P</td>
<td>Pumping power</td>
<td>f fluid</td>
</tr>
<tr>
<td>q_w</td>
<td>wall heat flux</td>
<td>i inlet</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number, ( Re = \rho u_i D_h/\mu )</td>
<td>o outlet</td>
</tr>
<tr>
<td>u_i</td>
<td>inlet velocity</td>
<td>s smooth-tip channel</td>
</tr>
<tr>
<td>W</td>
<td>channel height</td>
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</tbody>
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References


