Conjugate Flow and Heat Transfer Mechanism between the Rib Cooling and Film Cooling

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ABSTRACT

Rib cooling and film cooling are the effective cooling schemes to meet the increasing demands of rising turbine inlet temperatures in modern gas turbine engines. Most studies have been carried out to find the flow and heat transfer characteristic of rib cooling or film cooling individually. While the conjugate heat transfer between these two cooling methods is rarely studied in the literature. In order to evaluate the interplay between the rib cooling and film cooling, the conjugate flow and heat transfer mechanism in two perpendicular flat plate channels connected by a conductive solid wall with a compound film hole through it is studied numerically. Ribs are arranged in the internal cooling channel and the heat conduction in the connecting wall and ribs are considered. The blowing ratios vary from 0.75 to 1.5. SST $\kappa-\omega$ and large-eddy simulation (LES) turbulence model are checked to find the applicability of the turbulence models in this simulation by comparing the numerically predicted results with the experimental results from literature. The results indicate that LES is better at capturing the fine structure of the flow field and has higher accuracy. The arrangement of film hole strengthens the heat transfer capacity of the internal cooling channel. However, the arrangement of ribs reduces the overall cooling efficiency of the mainstream cooling channel.

INTRODUCTION

Due to the increasing demands on the high efficiency and thrust of gas turbine engine, the turbine inlet temperature keeps rising. At present, the turbine inlet temperature of advanced aero engines [1] is 1800-2000K, while the highest heat resistance temperature of single crystal superalloy for blade material is 1376K, and the temperature difference is as high as more than 600 K. To ensure the safe and reliable operation of the engine at such a high temperature, both efforts on
developing high-temperature resistant materials and efficient cooling techniques for turbine blade are of vital importance. Turbine blade cooling techniques are often divided into internal cooling and film cooling, as shown in Fig. 1. In internal cooling, the turbulent intensity of coolant is increased through the arrangement of disturbing elements, so as to increase the heat transfer effect, including ribs, impingement holes and pin-fins. In film cooling, the coolant flows out through the film holes and covers the blade surface to form a diaphragm. The conjugate heat transfer of internal cooling and film cooling is a common cooling method.

So far, most studies in the open literature focused on the individual flow and heat transfer characteristics of various internal cooling or film cooling of the square or rectangular channel. In addition, most results were obtained at low temperature conditions.

As we know, ribs enhancement cooling has received much more attentions in the past decades. Han proposed that in a stationary channel with ribs, the heat transfer characteristics mainly depends on the channel aspect ratio, rib structures and Reynolds number [2]. The effect of the channel aspect ratio, rib angles to the airflow and rib shapes [3-4] on the internal cooling effect have been studied by experiment method. Yang et al. [5] experimentally and numerically explored the heat transfer and resistance characteristics of the square channel with symmetric and staggered ribs, respectively, and studied the effect of the ratios (e/H) of rib height to channel height and the rib pitch to height ratios (S/e). Ravi et al. [6] numerically studied the flow and heat transfer performance of the square channel with 45 deg angled, V-shaped, W-shaped and M-shaped ribs, respectively. Reynolds number vary from 20000 to 70000. The results show that the V-shaped and 45 deg ribs have better overall thermal performance than others. Besides, Kaewchoothong et al. [7] experimentally studied the effect of the rib arrangements and angles on the flow and heat transfer performance of the square channel. The rib arrangements included inclined ribs, V-shaped ribs and inverted V-shaped ribs. And Wang et al. [8] numerically studied the thermal performance of the rotating rectangular channel (2:1) with 45 deg angled, reverse 45 deg angled, 45 deg V-shaped and outer-leaning 45 deg V-shaped ribs, respectively. The above research shows that the rib cooling technology has been extensively studied, but more scholars currently study the flow and heat transfer characteristics of the internal cooling channel where the ribs are arranged, ignoring the effect of the film hole on the cooling characteristics of the rib cooling channel. As a result, the heat transfer effect of the internal cooling channel is underestimated.

The film cooling was first proposed by Goldstein in 1971 [2], and the effect of hole geometry, coolant density, laminar boundary layer and surface roughness on the film cooling effectiveness was studied by experimental methods. Cao et al. [9] experimentally and numerically studied the effect of film hole shapes on the film cooling effectiveness of two perpendicular flat plate channels, and found that the film cooling effectiveness of the sister hole is the best. Bashir et al. [10] studied the film cooling effectiveness of flat plate with three-row compound-angle film holes by using PSP technique. The results show that the blowing ratio effect is closely related to the geometric design. And film hole spacing effect is strongly dependent on the interaction between jet and mainstream. Zhang et al. [11] experimentally studied the effects of vortex generator height (H/D) and location (L/D) on the film cooling effectiveness of flat plate, and found that the vortex generator contributes to enhancing film cooling effect. Mcclintic et al. [12-13] and Stratton et al. [14] studied the effect of the direction of internal crossflow on the film cooling effectiveness of two perpendicular flat plate channels connected by a row of different shaped film holes by experimental method and CFD simulations, and found that LES is able to predict adiabatic film cooling effectiveness with reasonable accuracy. The above research shows that film cooling technology has been extensively studied, but the influence of the internal cooling structure of the internal cooling channel on the film cooling efficiency is neglected, so that the film cooling characteristics are underestimated or overvalued.

Recently, a few researchers has begun to pay attention to the conjugate heat transfer between internal cooling and film cooling. For the high temperature area of the blade on the pressure and suction surfaces, ribs enhancement internal cooling combined with the external film cooling is often an effective cooling scheme to reduce the blade surface temperature. However, there are few studies on the interaction between the internal and external cooling methods and the comprehensive influence on the cooling effect of turbine blades. Sakai et al. [15] and Agata et al. [16-17] compared the film cooling effectiveness and heat transfer coefficient of the flat plate with cylindrical film holes while the other side of the plate without or with ribs by experimental and numerical methods. The temperatures of the mainstream and coolant are 323K and 298K, respectively. Besides, Sakai et al. [18] experimentally and numerically studied the effect of internal ribs and rear bumps on the film cooling effectiveness of the laid-back-fan-shaped film holes. Based on the results in reference 1, Klavetter et al. [19] studied the effect of the angle distribution (Aligned, Perpendicular) and the relative positions between 45 deg ribs and compound-angle cylindrical film holes on the film cooling effectiveness by experimental method. The temperatures of the mainstream and coolant (liquid nitrogen) are 305K and 203K, respectively. And the thermal conductivity of material is 0.048 W/m•K. The study found that the case of perpendicular mid-pitch configuration has the best film cooling effectiveness.

As stated above, the practical cooling condition of the turbine blade includes the high heat transfer temperature difference between the mainstream and the coolant, thus the conjugate heat transfer between the mainstream and the coolant through the conduction of blade/rib material and the flow in film hole must have significant effect on the temperature distribution inside the turbine and the performance of film cooling and internal cooling. Therefore, the present work is to numerically study the conjugate flow and heat transfer mechanism between the ribs enhancement cooling and film cooling.
under medium temperature condition to check the detailed characteristics of turbine blade cooling close to the actual operation conditions.

**PHYSICAL MODEL AND BOUNDARY CONDITIONS**

Based on the E³ blade profile [20], a turbine blade model is established, as shown in Fig. 1. The third passage inside the blade is taken as the research channel in this paper. The rib pitch-to-height ratio (P/e) is 9 and the rib height-to-channel hydraulic diameter ratio (e/Dh) was 9.5%. For the first step, we try to find out an appropriate and reliable numerical method to simulate the conjugate heat transfer between the mainstream and the coolant through the conduction of blade/rib material and the flow in film hole, a simplified computational model based on the third passage in Fig.1 is established, which is composed of two perpendicular flat plate channels connected by the conductive solid wall with a film hole. Ribs are arranged in the internal cooling channel. The details about the computational model are shown in Fig. 2 and Fig. 3. Coordinate X is the flow direction of mainstream flow, coordinate Y is the direction of blade wall thickness, and coordinate Z is the flow direction of coolant.

The geometrical sizes of the model are listed in Table 1. To ensure the full development of the fluid at the inlet of the mainstream and internal cooling channels and avoid backflow at the outlet of the mainstream and internal cooling channels, the extended channels are placed upstream and downstream of the mainstream and internal cooling channels. The cross-section of the internal cooling channel is a non-rectangular section, the angle between the rib and the coolant is 90 deg, and the spacing-to-height ratio (P/e) of the ribs is 9. The compound film hole is a streamwise injection angle (α) of 30 deg and a compound angle (β) of 45 deg, as shown in Fig. 3. The diameter of the film hole (D) is 0.8mm. The thickness of the connecting wall is 1.4mm, which is from the practical thickness of a normal turbine blade.

![Fig. 1 blade model](image1)

![Fig. 2 Model for the present work](image2)

![Fig. 3 Perpendicular flat plate channels](image3)

| Table 1 Geometrical sizes of channels ( units: mm) | Lc | 10 | Lmi | 20.23 | Lco | 23 | P | 4.5 | Lw | 6.11 | e | 0.5 | Lcoo | 6 | t | 1.4 | Lmi | 23 | D | 0.8 | Lmo | 50.66 |

Fig. 1 blade model
Fig. 2 Model for the present work
Fig. 3 Perpendicular flat plate channels
COMPUTATIONAL CONDITIONS

For the match of parameters inside the two channels, the parameters at the boundaries of the computational model are defined based on the turbine test rig from NASA [20]. The annular cascade in NASA’s test has 76 blades. However, the paper only considers one blade of the annular cascade. Therefore, the mass-flow-inlet of 0.01865 kg/s and inlet temperature of 709 K are set at the inlet of the mainstream channel. The Reynolds number is 1.08×10^5. The outlet of the mainstream channel is set to be pressure-outlet boundary condition, and the outlet static pressure is 206431 Pa. The two side planes of the mainstream channel are set as periodic boundary, and the top surface is set as symmetric boundary. The inlet of the internal cooling channel is set to be mass-flow-inlet boundary condition with inlet temperature of 339 K, and the mass flow is 0.00231 kg/s. The outlet of the internal cooling channel is also set to be pressure-outlet boundary condition, and the two side planes and bottom wall is set as the adiabatic non-slip boundary. The contacting wall between fluid and solid is set as the fluid-solid coupled surface. The value of blowing ratio is controlled by adjusting the outlet pressure of internal cooling channel. The range of blowing ratios is 0.75-1.5 in the present study.

The fluid is ideal compressible air, and specific heat, viscosity and thermal conductivity vary with temperature. As we all know, single crystal superalloy DD6 has the advantages of high temperature strength, good comprehensive performance, stable structure and good casting process performance, etc., and it is suitable for the production of high-temperature parts such as gas turbine blades with complex cavities working below 1100℃[21], which is thermal conductive, so the paper selects crystal superalloy DD6 as the solid material to account for the influence of the thermal conductivity inside the channel wall and ribs.

DEFINITION OF FEATURE PARAMETERS

For the post processing of the results, the following parameters and dimensionless number are defined.

(1) The blowing ratio of the mainstream channel

\[ M = \frac{\rho_c u_c}{\rho_\infty u_\infty} \]

Where \( \rho_c \) is jet density, kg•m⁻³, which varies with temperature and pressure. \( u_c \) is the jet velocity, m•s⁻¹, which is a result of this conjugated computation. \( \rho_\infty \) is mainstream density, kg•m⁻³, which varies with mainstream inlet temperature and pressure. \( u_\infty \) is the mainstream inlet velocity, m•s⁻¹.

(2) For turbine blades under real operating conditions, the thermal conductivity of the blade material cannot be ignored. Therefore, similar to the definition of adiabatic cooling effectiveness, in this paper, the overall cooling effectiveness is defined as Eq.(2)

\[ \phi = \frac{T_s - T_x}{T_s - T_c} \]

where \( T_s \) is the surface temperature of the connecting wall on the side of the mainstream channel when considering the thermal conductivity of solid materials, K. \( T_x \) is the mainstream temperature, K. \( T_c \) is the coolant temperature, K.

MESHING AND GRID INDEPENDENCE ASSESSMENT

ANSYS ICEM 18.0 is used to generate hexahedral structure grid to mesh the fluid and solid domains as detailed in Fig. 4. Boundary layer grids are drawn around the film hole, ribs and near all the walls of the channel for the fluid domain. The height of the first-layer of boundary layer grids is set to be small enough to satisfy the criterion that \( y^+ \) value be less than 1 required by the turbulence model for the current study.

![Mesh of the computational domain](image1)

Fig. 4 Mesh of the computational domain

![Overall cooling effectiveness with M=0.75](image2)

Fig. 5 Overall cooling effectiveness with M=0.75
The overall cooling effectiveness with $M=0.75$ computed by LES turbulence model under three grid systems (1.16, 1.92 and 2.25 million) are compared to verify grid independence, as shown in Fig. 5. It is found that when the grid number exceeds 1.92 million, the overall cooling effectiveness is basically the same, which indicates that grid number around 1.9 million should be acceptable for obtaining grid independent solution and saving computing time. Therefore, the model with 1.9 million grids will be adopted for the following calculation.

**TURBULENCE MODEL VERIFICATION**

To verify the applicability of turbulence model in the present problem, the paper selected two turbulence models often used for film cooling in the literature: SST $\kappa$-$\omega$ and LES, and the numerical simulation results were compared with Agata's experiment [16]. The numerical model is simplified based on the experiment setup, and the boundary conditions are according to the experimental conditions. Figure 6 shows the comparison of the overall cooling efficiency of the experimental results and the simulation results with SST $\kappa$-$\omega$ and LES turbulence models when $M=1.0$. It can be found that the simulation results of LES are closer to the experimental results.

![Overall cooling effectiveness comparison between experimental, LES and SST results](image)

**RESULTS AND DISCUSSION**

*Flow field analysis of internal cooling channel*

Figure 7 shows the effect of ribs on the flow and heat transfer characteristics of the internal cooling channel with film hole. The ribs play two roles in the internal cooling channel. The first is to guide the flow and change the direction of the jet entering the film hole, and the second is to generate vortices to enhance the heat transfer effect of the wall.

![Transient streamline of the cross section in the internal cooling channel when $M=0.75$](image)

Figure 8 shows the influence of the arrangement of film hole on the heat transfer characteristics of the internal cooling channel when $M=0.75$. As can be seen from Fig. 8 (a), the internal cooling channel where the ribs are arranged has two separation areas, the first one is downstream area of the upstream rib, and the other is upstream area of the downstream rib. Heat transfer enhancement is caused by the separation and reattachment of the boundary layer. It can be seen from Fig. 8(b) that the arrangement of the film hole has a disturbing effect on the coolant, affects the flow and heat transfer characteristics of the internal cooling channel, and strengthens the heat transfer effect of the wall. In the upstream area of the film hole, the heat transfer is enhanced because the outflow of the film hole accelerates the fluid in the upstream boundary layer of the film hole, thereby reducing the thickness of the boundary layer. The reason for the enhanced heat transfer in the downstream area of the film hole is that the arrangement of the film hole destroys the fully developed boundary layer, and the downstream boundary layer is being generated. In addition, the coolant in the downstream of the film hole produces a
partial velocity against the wall, which has a certain impact and disturbance effect on the wall, so that the heat transfer is enhanced. As shown in Table 2, the arrangement of the film holes enhances the heat transfer on the coupled wall of the internal cooling channel by 11.26%.

![Image](a) Without film hole

![Image](b) With film hole

Fig. 8 Time-average temperature distribution of the wall and transient streamline of the cross section in the internal cooling channel (units: K)

Table 2 $h_{ave}$ on the coupled wall of the internal cooling channel (units: W/m²K)

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<tr>
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<th>Without film hole</th>
<th>With film hole</th>
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<tr>
<td>$h_{ave}$</td>
<td>385.23</td>
<td>428.6</td>
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Flow field analysis of mainstream channel

Vortex structure.

Due to the interaction between the jet and the mainstream, a complex vortex system structure will be formed near the film hole, as shown in Fig. 9. In this paper, Q Criterion ($Q = \frac{1}{2}(\Omega_{ij} S_{ij} - S_{ij}^2)$) proposed by Hunt et al. [22] is used as the vortex criterion to characterize the vortex system structure in the flow field. Fig. 10 clearly shows the horseshoe vortex and shear vortex located at the upstream of the film hole, the counter rotating vortex pair that dominates the flow characteristics of the far flow field, and the wake vortex near the plate and the downstream area of the film hole. The vortex system structures will affect the development of the main body of the jet. Among them, the counter rotating vortex pair has the greatest influence on the mixing characteristics of the film cooling, which is mainly formed by the entrainment of the boundary layer vortex on the wall.

![Image](Fig. 9 Four types of vortex structure of the jet [23])

![Image](Fig. 10 Vortex structure of the jet for numerical simulation)

Figure 11-12 show the effect of the ribs arrangement on the temperature and overall cooling efficiency of the downstream area of film hole when $M=0.75$. As can be seen from the streamline in Fig. 7 that the diversion effect of the ribs changes the direction of the jet entering the film hole, which increases the deflection of the jet entering the film hole, thereby increasing the flow loss in the film hole, decreasing the outlet velocity of the film hole, as shown in Fig. 11. Therefore, the jet near the downstream of the film hole fits more closely to the wall, and the overall cooling efficiency is higher. When
the jet flows downstream, the outward shift distance of the jet becomes shorter. Therefore, the overall cooling efficiency far away from the downstream of the film hole decreases, as shown in Fig. 12-13.

Fig. 11 Time-average temperature distribution and velocity vector downstream of the compound film hole at $x/D=3$ when $M=0.75$ (units: K)

Fig. 12 Overall cooling efficiency contour downstream of the compound film hole when $M=0.75$

Fig. 13 Overall cooling efficiency curve downstream of the compound film hole when $M=0.75$

Temperature field analysis.

Figure 14 shows the time-average temperature field at different positions downstream of the film hole when $M=1.0$. It can be found that at $x/D=3$, the core of the jet has not been diluted, the temperature of the cooling fluid is lower and stays near the surface, and the overall cooling efficiency is higher. When the jet flows downstream, the cooling fluid is continuously heated by the mainstream, and the temperature becomes higher. Therefore, the overall cooling efficiency of the downstream of the jet is reduced, as shown in Fig. 15. The arrangement of the film hole increases the overall cooling efficiency in the area covered by the jet, so the overall cooling efficiency first increases along the blade height and then decreases. Because the compound film hole is inclined at 45 deg in the blade height direction, the maximum overall cooling efficiency along the blade height direction at different $x/D$ positions downstream of the film hole is shifted.

Fig. 14 Time-average temperature distribution downstream of the film hole with $M=1.0$ (units: K)
CONCLUSIONS

The paper study the conjugate heat transfer between rib cooling and film cooling on the flow and heat transfer mechanism of in two perpendicular flat plate channels connected by film hole by numerical method. The conclusions are as follows:

(1) The LES turbulence model can more accurately predict the flow field structure of film cooling.
(2) The arrangement of film hole affects the flow and heat transfer characteristics of the internal cooling channel, and the heat transfer effect of the wall becomes better.
(3) The arrangement of ribs reduces the overall cooling efficiency of the mainstream channel.

NOMENCLATURE

\[ D \]  film hole diameter, mm
\[ D_h \]  equivalent diameter of the internal cooling channel, m
\[ e \]  height of the rib , mm
\[ h_{ave} \]  averaged heat transfer coefficient, W·m\(^{-2}\)·K\(^{-1}\)
\[ k \]  thermal conductivity, W·m\(^{-1}\)·K\(^{-1}\)
\[ L_c \]  length of the conjugated section of the internal cooling channel, mm
\[ L_{ci} \]  length of the development section of the internal cooling channel, mm
\[ L_{co} \]  length of the outlet section of the internal cooling channel, mm
\[ L_m \]  length of the conjugated section of the mainstream channel, mm
\[ L_{mi} \]  length of the development section of the mainstream channel, mm
\[ L_{mo} \]  length of the outlet section of the internal cooling channel, mm
\[ M \]  blowing ratio
\[ P \]  spacing of the rib , mm
\[ t \]  blade wall thickness, mm
\[ T_w \]  wall temperature of mainstream channel, K
\[ T_\infty \]  mainstream temperature, K
\[ u_c \]  velocity of jet, m·s\(^{-1}\)
\[ u_\infty \]  velocity of mainstream, m·s\(^{-1}\)
\[ \alpha \]  streamwise injection angle, deg
\[ \beta \]  compound angle, deg
\[ \rho \]  density, kg·m\(^{-3}\)
\[ \rho_c \]  density of jet, kg·m\(^{-3}\)
\[ \rho_\infty \]  density of mainstream, kg·m\(^{-3}\)
\[ \phi \]  overall cooling effectiveness
\[ \bar{\phi} \]  averaged overall cooling effectiveness

ACKNOWLEDGMENTS

The work is supported by the National Science and Technology Major Project (2017-III-0003-0027).

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