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# Numerical simulation on a film-cooled rotating model with 30° injection holes

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**Abstract:** The flow and heat transfer characteristics were numerically investigated on a film cooling model under different rotating operating conditions. The computational model was originated from the mid span section of a typical turbine rotor with two rows of 14 staggered injection holes angled 30° both on the suction surface and pressure surface, and the flow through the coolant plenum and all the hole pipes were resolved as a part of the computational domain by specifying the coolant mass flux in the plenum. The computations primarily focus on understanding the rotational effect on film cooling performance in mechanism research approach. In the present study, the Reynolds number ( $Re$ ) based on mainstream velocity and injection hole diameter varied from 1835.5 to 5507.4, and the averaged blowing ratio ( $M$ ) ranges of 0.5 to 1.5. Results show that the coolant will move on to the high radius locations near the suction and pressure surfaces due to the strong centrifugal effect, which leads to the decrease in adiabatic effectiveness accordingly. The discharge coefficients ( $C_d$ ), on the pressure surface, are much higher than that on the suction surface under a given operating condition. In addition, the critical values of angular speed which represent the equilibrium of centrifugal force and Coriolis force near the pressure surface are also presented.

**Key words:** film cooling; rotating; adiabatic effectiveness; heat transfer; discharge coefficient

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## Nomenclature

|       |                                                              |
|-------|--------------------------------------------------------------|
| $B$   | Space between two rows                                       |
| $C_d$ | Discharge coefficient                                        |
| $D$   | Film cooling hole diameter                                   |
| $Ec$  | Eckert number                                                |
| $h_t$ | Heat transfer coefficient                                    |
| $L$   | Length of the hole pipe                                      |
| $m$   | Mass flow rate                                               |
| $M$   | Averaged blowing ratio ( $M = (\rho_c u_c) / (\rho_g u_g)$ ) |
| $Nu$  | Nusselt number ( $Nu = h_t D / \lambda_g$ )                  |
| $p$   | Pressure                                                     |
| $Pr$  | Prandtl number ( $Pr = \nu / a$ )                            |
| $q$   | Heat flux                                                    |
| $R$   | Gas constant                                                 |
| $Re$  | Reynolds number ( $Re = u_g D / \nu$ )                       |
| $R_t$ | Rotation number ( $R_t = \Omega D / u_g$ )                   |
| $S$   | Distance between holes in a row                              |
| $T$   | Temperature                                                  |

|           |                                                                |
|-----------|----------------------------------------------------------------|
| $T_u$     | Turbulence intensity                                           |
| $u$       | Velocity                                                       |
| $u^*$     | Shear velocity                                                 |
| $x, y, z$ | Cartesian coordinate system with $z$ coordinate along the span |
| $X^*$     | Dimensionless coordinates                                      |
| $y^+$     | Distance in wall coordinates ( $y^+ = y u^* / \nu$ )           |
| $a$       | Thermal diffusivity                                            |

## Greek symbols

|           |                                                                         |
|-----------|-------------------------------------------------------------------------|
| $\rho$    | Density                                                                 |
| $\nu$     | Kinematic viscosity                                                     |
| $\Omega$  | Angle speed                                                             |
| $\varphi$ | Inclination angle to wall surface                                       |
| $\eta$    | Film cooling effectiveness<br>( $\eta = (T_g - T_{aw}) / (T_g - T_c)$ ) |

## Subscripts

|    |                |
|----|----------------|
| aw | adiabatic wall |
| w  | wall           |

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c        coolant  
g        mainstream

## Introduction

From the thermodynamic analysis, it is revealed that the performance of gas turbine engines can be enhanced by increasing the turbine inlet temperature. Since the inlet temperatures of modern gas turbines are much higher than the melting temperatures of the materials used to make the turbine blades, cooling of the blades becomes a critical issue in turbine design. Film cooling which is widely used is probably the most effective cooling technique for protecting the high temperature turbine blades in aero engines up to now. In this technique, the cooler air is injected into the high temperature mainstream boundary layer on the blade surface and generates a thin coolant film acting as a buffer to isolate the thermal loads. With the aid of film cooling, a higher gas temperature at the turbine inlet is permitted, which in turn, improves the thermal efficiency of the turbine engines greatly.

Various fluid dynamics and geometric parameters can affect the performance of film cooling, such as the mainstream Reynolds number, blowing ratio, coolant to mainstream density ratio, turbulence intensity, the film hole shape, size and location, etc. Therefore, to achieve a comprehensive understanding of the film cooling technique as well as efficient design of the film cooling modes has become a very important issue.

Many studies have been conducted on the performance of film cooling over the past four decades, and a large amount of the valuable results is available in the open literature. An excellent review of the film cooling work up to 1971 was provided by Goldstein<sup>[1]</sup>. Many efforts have been devoted afterwards to this research area, and a comprehensive compilation of the achievements has been summarized by Han et al.<sup>[2]</sup>. From a view of experimental and numerical approaches, the development of film cooling for the most recent years is discussed here.

Sinha et al.<sup>[3]</sup> investigated two rows of 35° holes with a pitch to diameter ratio of 3 and a row spacing to diameter ratio of 40 film cooling performance. They found that the upstream row thickened the boundary layer between the two rows, reduced the momentum of the crossflow and the velocity gradients in the shear layers, and lowered the turbulence level for the second row. Baldauf et al.<sup>[4]</sup> employed a high resolution thermography system to study the local adiabatic film cooling effectiveness on a flat plate surface downstream a row of cylindrical injection holes. It was found that the optimum conditions are confined to a small band of blowing ratios around  $M = 0.85$  to 1.0 and momentum ratios of  $I = 0.4$  to 0.5. Saumweber and Schulz<sup>[5]</sup> conducted a comprehensive set of generic experiments to investigate the interaction of film cooling rows. They considered five different film cooling configurations on a large scale basis each consisting of two rows of injection holes in staggered arrangement and found that the decay of film cooling effectiveness with streamwise distance was much less pronounced downstream of the second row primarily due to pre-cooling of the boundary layer by the first row of holes. Kim et al.<sup>[6]</sup> investigated the detailed local heat transfer and film cooling effectiveness distributions from a trailing-edge model of a gas turbine airfoil in a low speed wind tunnel. They found that both local heat transfer coefficients and film cooling effectiveness displayed a strong dependency on blowing ratio and the mainstream acceleration. However, the discharge coefficients showed little dependency on the mainstream acceleration. Yuen and Martinez-Botas<sup>[7-8]</sup> studied the film cooling effectiveness and heat transfer coefficient distributions on rows of cylindrical holes with streamwise angles of 30°, 60° and 90° in a flat plate test facility with zero pressure gradient. They concluded that the optimum value of blowing ratio was about 0.5 for most configurations.

Besides the experimental film cooling studies outlined above, many efforts in computation

al approach have also been devoted to the film cooling characteristics. A number of parametric studies were provided by Garg et al.<sup>[9-12]</sup>.

Leylek and Zerkle<sup>[13]</sup> studied the discrete hole film cooling on a flat plate using the standard  $k-\epsilon$  turbulence model with wall functions for  $R_d = 2$ , and  $L/d$  of 1.75 and 3.5. Following the coolant from the plenum into the hole pipe, they presented the coolant velocity and temperature profiles at the hole exit. By using the same turbulence model, Lakehal et al.<sup>[14]</sup> investigated the film cooling effectiveness of a flat plate by a row of laterally injected jets using a Navier-Stokes equation solver. Compared to the results obtained with wall functions, they found that the resolution of the viscosity-affected near-wall region with a one-equation turbulence model yielded a noticeable improvement in the prediction of film cooling effectiveness. Azzi and Lakehal<sup>[15]</sup> simulated the streamwise injection of film coolant over a flat plate for density ratio of two and mass flux ratios of 0.5 and 1.0. Results showed that the isotropic two-layer  $k-\epsilon$  model was found to underpredict the lateral spreading of the temperature field, together with an exaggerated size of the kidney vortices. Garg and Rigby<sup>[16]</sup> used a multiblock, three-dimensional Navier-Stokes code with  $k-\omega$  turbulence model to study the within-hole and near-hole behaviors in relation to heat transfer on a film-cooled blade. It was found that the coolant velocity and temperature distributions at the hole exit on the blade side did not follow the 1/7th power-law profile, and were highly skewed by mainstream for the holes close to the leading edge. To obtain more accurate results, Muldoon and Acharya<sup>[17]</sup> computed the film cooling performance of an inclined injection hole by using the direct numerical simulation (DNS). Finally, they concluded that the great weakness in the standard  $k-\epsilon$  model was the choice of eddy viscosity.

All studies described so far have shown lots of significant results in static film cooling already. In the open literature, however, few academic researches have been reported to investigate

the film cooling performance under rotating engine conditions.

Dring et al.<sup>[18]</sup> studied the adiabatic film cooling effectiveness on the rotor blade of a large scale (low speed) model for a high pressure turbine first stage. Their results showed that the film coolant had only a small radial displacement, similar to flat plate results, on the suction surface. But on the pressure surface, the coolant trace had a large radial displacement toward the blade tip due to rotation, and a much faster decay of the effectiveness was observed. Takeishi et al.<sup>[19]</sup> measured the adiabatic effectiveness on a low-speed stationary cascade and a rotating blade by using the heat-mass transfer analogy. They reported that the adiabatic effectiveness on the suction surface of the rotating blade agreed well with that on the stationary blade, but a low level of effectiveness appeared on the pressure surface of the rotating blade. Abhari and Epstein<sup>[20]</sup> studied the time-resolved heat transfer characteristics for cooled and uncooled rotors by thin heat flux gages, and Ahn et al.<sup>[21]</sup> measured the film cooling effectiveness distributions on the leading edge region of a rotating blade using a pressure sensitive paint (PSP) technique. As a further study, Ahn et al.<sup>[22]</sup> reported that different rotating speeds can change the film cooling traces distinctly with the averaged film cooling effectiveness on the leading edge region increasing with blowing ratio. Yang et al.<sup>[23]</sup> employed a Reynolds stress turbulence model together with a non-equilibrium wall function to calculate the film cooling effectiveness and the associated heat transfer coefficients on the leading edge of a rotating blade in a particular turbine stage. Results showed that the heat transfer coefficients increased, but film cooling effectiveness decreased with increasing rotating speed. Furthermore, a series of computational studies on the heat transfer characteristics for a film-cooled rotating blade by using different turbulence models were also conducted by Garg and his co-workers<sup>[24-26]</sup>.

In order to better understand the mecha-

nism of film cooling phenomenon under rotating operating conditions, both the studies concerning the performance on entire turbine blade and features on physical model are extremely important. Therefore, from the view of mechanism research, the general purpose of the present work aims to investigate the rotational effect on the film cooling performance. By means of the computational fluid dynamics (CFD) approach, the flow and cooling effectiveness on a simplified rotating film cooling model with two rows of 14 staggered injection holes are investigated, and the discharge coefficient distributions for all the hole pipes are also obtained. The computed results will be greatly helpful to explore the film cooling mechanism in gas turbine engines.

## 1 Methodology theoretical analysis

1) The cooling performance of a film-cooled surface is most commonly described by the adiabatic film cooling effectiveness  $\eta$ , which is defined as a non-dimensional temperature:

$$\eta = (T_g - T_{aw}) / (T_g - T_c) \quad (1)$$

where  $T_g$  is the mainstream temperature,  $T_c$  is the coolant temperature, and  $T_{aw}$  is the adiabatic wall temperature.

2) For design purposes, it is often required to know the relation between the heat fluxes to a wall and the temperature of the wall surface. With film cooling, the heat transfer coefficient  $h_t$  is customarily defined by

$$h_t = q / (T_{aw} - T_w) \quad (2)$$

where  $q$ ,  $T_{aw}$  and  $T_w$  denote the wall heat flux with film cooling, adiabatic wall temperature and the wall temperature, respectively. The  $h_t$  considers influence of the air interaction near the blade surface, and usually is obtained from a series of computations. Another way of defining the heat transfer coefficient  $h_g$  is given by

$$h_g = q / (T_g - T_w) \quad (3)$$

Dividing both sides of Eq. (3) by both sides of Eq. (2), and rearranging, it follows that:

$$h_g / h_t = 1 - \eta (T_g - T_c) / (T_g - T_w) \quad (4)$$

When  $T_g$  is equal to  $T_c$ , Eq. (4) can be written as:

$$h_g = h_t \quad (5)$$

The selection of constant wall heat flux  $q$  is independent of its value if the air is incompressible, and finally the values of  $h_t$  can be obtained from  $h_g$  when  $T_g$  is equal to  $T_c$ . This method is firstly analyzed by Eckert<sup>[27]</sup> and applied extensively in experimental researches for static film cooling characteristics. The present study will also employ the classical method to calculate the heat transfer coefficient and inspect its feasibility for rotating film cooling investigation.

3) After processing to the flow and heat transfer governing equations and boundary conditions in rotating reference frame on a film-cooled blade, the dominating normalized parameters to Nusselt number  $Nu$  and adiabatic effectiveness  $\eta$  can be summarized as follows:

$$Nu = f_1(Re, Ri, Pr, M, p_c/p_g, \rho_c/\rho_g, \varphi, X^*) \quad (6)$$

$$\eta = f_2(Re, Ri, Pr, M, p_c/p_g, \rho_c/\rho_g, \varphi, X^*) \quad (7)$$

where  $Re$  is the Reynolds number based on injection hole diameter,  $Ri$  is the rotation number,  $Pr$  is the Prandtl number,  $M$  is the blowing ratio,  $p_c$  and  $p_g$  are the coolant pressure and mainstream pressure,  $\rho_c$  and  $\rho_g$  are the coolant density and mainstream density,  $\varphi$  is the inclination angle, and  $X^*$  is the dimensionless coordinates. The present work aims to concentrate on the influence of leading parameters,  $Re$ ,  $Ri$ , and  $M$  on the performance of film cooling.

## 2 Rotating film cooling model

A schematic illustration of the rotating film cooling model, which is originated from the mid-span section of a typical turbine rotor and simplified properly, with two rows of staggered injection holes located both on pressure and suction surfaces is shown in Fig. 1. The coordinate system used is such that  $x$ -axis is along the stream-wise direction of mainstream,  $y$ -axis is along the direction of injection holes and  $z$ -axis conforms with the right-hand law. The entire model occupies a space of  $0.12\text{ m} \times 0.082\text{ m} \times 0.05\text{ m}$ , and the cutting plane normal to  $x$ -axis is a trapezoid

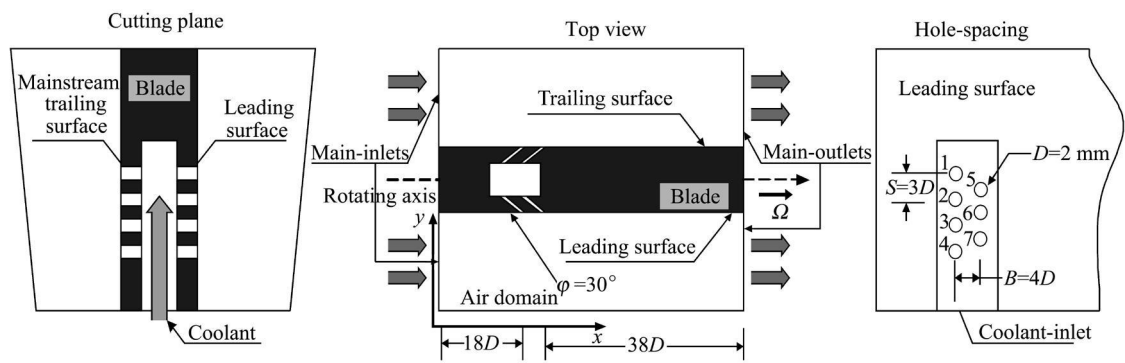


Fig. 1 Schematic of rotating model and film hole configuration

with a top angle of  $74.6^\circ$ . As can be seen, the air domain consists of mainstream and coolant which are mixed at the exit of injection holes, and the interior part forms the solid domain. The whole model is symmetric to the rotating axis that is parallel but at a distance of 0.5m under the  $x$ -axis. The mainstream has a high temperature and velocity when the hot air reaches the two main inlets with a cross-section of  $0.028\text{ m}\times 0.05\text{ m}$ , and the coolant has a low temperature and constant mass flux when the cooling air enters the coolant inlet with a cross-section of  $0.02\text{ m}\times 0.01\text{ m}$  which is at the bottom of inner plenum. All the air flow out from the two main outlets. A total of 14 discrete cylindrical holes are designed on the pressure and suction surfaces for film cooling. To investigate the coolant interaction and discharge coefficient distributions, the injection holes are arranged in two staggered rows. The first row of 4 holes and the second row of 3 holes have a diameter  $D$  of 2mm, and the holes in each row are spaced three-hole diameters apart ( $S/D=3$ ). The space between two rows is  $B/D=4$ , and the staggered distance is  $1.5D$ . The inclination angle  $\varphi$  to the pressure and suction surfaces is arranged at  $30^\circ$ , giving the length to diameter ratio  $L/D=6$ . In addition, this model rotates at constant angular speeds of 1000, 2000 and 3000 r/min, respectively.

3 Grid generation and boundary conditions

An excellent locally structured and body-fit-

ted grid with hexahedral cells is shown in Fig. 2. The grid consisted of  $122\times 130\times 126$  nodes in the  $x$ ,  $y$  and  $z$  directions, and still maintained high quality, smoothness and near orthogonality. Normal to the wall surface is the densely viscous grid, with  $y^+<1.2$  for the first grid point which was strictly kept within the laminar sublayer to satisfy the requirement of low- $Re$  turbulence model. To obtain a high precision grid solution, the O-grid technique was applied in all the injection holes and much attention was paid to keep the grid aspect ratios in any direction smaller than 1.3. Many tests were carried out to obtain an independent solution during the process of grids refining, including a grid which had 1.5 times as many cells in the streamwise direction as this one. The two grids yielded nearly identical adiabatic effectiveness and pressure distributions on the pressure and suction surfaces,

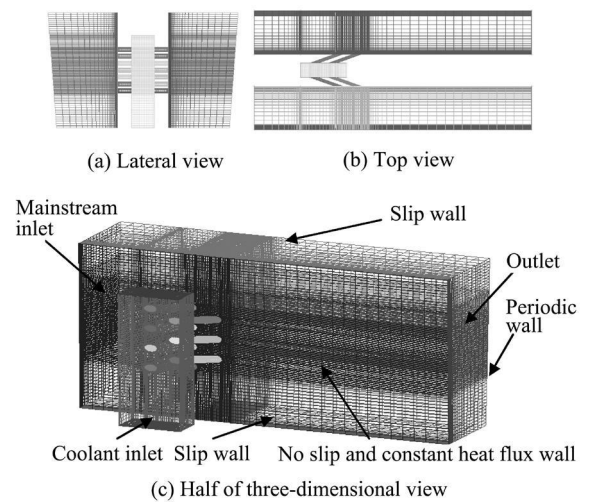


Fig.2 Grid used for the computation

and the maximum discrepancy in wall temperature prediction was within  $\pm 1\%$ .

The velocity boundary conditions to the mainstream and coolant inlet planes are summarized in Table 1. The mainstream had a temperature of 363.15 K with turbulence intensity of  $T_u=5\%$  after it reached the two main inlet planes, and the coolant had a temperature of 333.15 K with turbulence intensity of  $T_u=1\%$  in the plenum. When the mixed air streamed out from the two outlet planes, the back pressure was fixed to 101325 Pa accordingly. Because the mainstream is designed as simulating the flow between neighboring turbine blades, the top and bottom surfaces of mainstream were set to free-slip walls, and two planes in the exterior of this part were considered as periodic interfaces. The suction and pressure surfaces were firstly assumed to be adiabatic to calculate the film cooling effectiveness, and then changed to constant heat flux walls when calculating the heat transfer coefficient. According to the analysis of Eckert<sup>[27]</sup>, the temperatures of mainstream and coolant were altered to be equal, and the fluid was set to be incompressible in place of the ideal gas. In addition, the adiabatic no-slip condition was enforced at all the solid walls, including the inner surface of the film hole and the inner plenum.

| Table 1 Velocity at inlet planes in computation |             |         |         |
|-------------------------------------------------|-------------|---------|---------|
| $u_g/(m/s)$                                     | $u_c/(m/s)$ |         |         |
|                                                 | $M=0.5$     | $M=1.0$ | $M=1.5$ |
| 20.3                                            | 2.06        | 4.13    | 6.19    |
| 40.6                                            | 4.13        | 8.25    | 12.38   |
| 60.9                                            | 6.12        | 12.23   | 18.34   |

In the previous studies, many investigations have been conducted to inspect the actual effect of turbulence models (Azzi and Lakehal<sup>[15]</sup>, Garg and Rigby<sup>[16]</sup>, Garg and Abhari<sup>[24]</sup>, Garg<sup>[25]</sup>). Following the advice of Garg<sup>[25]</sup>, the  $k-\omega$  turbulence model was chosen because it satisfies many constraints such as asymptotic consistency near walls without the need for additional corrections and it can also give reasonable

results for subsonic flows with adverse pressure gradients, separation, and streamline curvature.

The numerical simulations were performed using the commercial CFD code Star-CD, a finite control volume based, multiblock and multigrid solver. A high resolution scheme was employed for the discretization of advection term, and the standard  $k-\omega$  model in conjunction with the scalable wall function was used for turbulence closure. Convergence in all cases was declared only when both of strict criteria were satisfied: (1) reduction in all residuals of four orders of magnitude at least, and (2) no observable change in surface temperature prediction for an additional 40 iterations.

#### 4 Results and discussion

A summary of the leading fluid dynamics parameters in the present study is shown in Table 2.

| Table 2 Leading parameters in computation |                        |
|-------------------------------------------|------------------------|
| Parameters                                | Values                 |
| $Re$                                      | 1835.5, 3671.9, 5507.4 |
| $R_t$                                     | 0.0034~0.0309          |
| $M$                                       | 0.5, 1.0, 1.5          |

#### 5 Flow and temperature field

Fig. 3 presents the adiabatic film cooling effectiveness distributions on the suction surface for various  $M$  and  $R_t$ . The particular cases shown here are  $Re=1835.5$ . To deliver a better understanding of the flow field, a horizontal line is marked through the center of hole 6 as a reference and the downstream exit of this hole is located at  $x=0$ . A particular phenomenon in rotating film cooling performance is that the coolant will move on to the high-radius locations after it streams out from the cooling holes especially at high angular speeds. It is well known that there are three governing forces in rotating system, i. e., the centrifugal force, the Coriolis force and the buoyancy force. The buoyancy force is induced through the former two forces and temperature field, and in the current study,

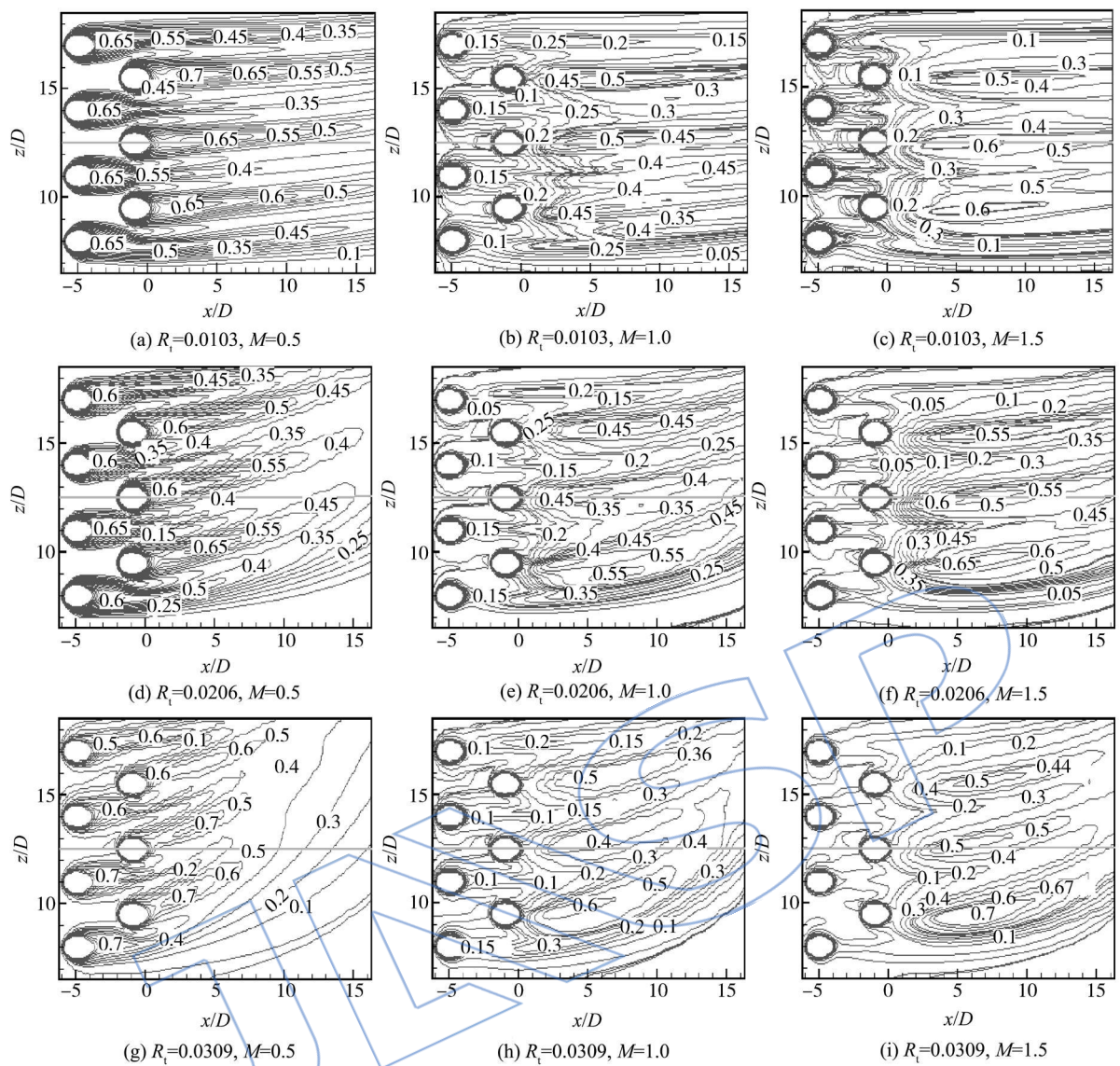


Fig. 3 Contours for  $\eta$  on suction surface at  $Re=1835.5$

it can be negligible for the small differential of temperature between the mainstream and coolant. The centrifugal force is prone to drive the air to deflect toward the high radius locations especially for the high-density coolant, while the Coriolis force pushes the cooling jets upwards and downwards along the  $z$ -axis direction near the suction and the pressure surfaces, respectively. As a result, the coolant deflection near the suction surface is much evident. The adiabatic effectiveness distributions in Fig. 3 can be very helpful to analyze the coolant deflection phenomenon. It can be obviously seen that as  $R_i$  increases from 0.0103 to 0.0309, the contour at  $\eta=0.5$  downstream of the second row of holes is deflected about  $3.5D$  upwards at  $x/D=8$ , and

the effective region covered by cooling film tends to decrease both in  $x$  and  $z$  directions (see Fig. 3 (a), (d), (g)). Because of the interaction of cooling jets from two rows of injection holes, the thermal protection between two neighboring holes of the second row along  $z$ -axis direction is still sufficient for this blowing ratio. When the value of  $M$  increases, a discrete region with low  $\eta$  values but high temperatures surrounding all the injection holes appears and reaches its maximum area at  $M=1.5$ . This is because the coolant at high blowing ratios has much great momentum so that it penetrates deeper into the mainstream easily at the hole exit and reattaches the wall surface after passing an appropriate distance along streamwise direction. Therefore,

high values of  $M$  are not suitable for the thermal protection to wall surfaces near the exit of injection holes.

6 Film cooling effectiveness

Fig. 4 presents the laterally averaged film cooling effectiveness results as a function of  $x/D$  at the centerline of hole 6 on the suction and pressure surfaces under different operating conditions. Through a detailed comparison, the tendencies of  $\eta$  are summarized as follows: (1) The high values of  $M$  will weaken the thermal protection to wall surfaces especially near the exit of cooling holes, but it will increase the adiabatic effectiveness properly at far downstream locations. The numerical result (Fig. 4(a)) reveals that an abrupt decrease in the cooling effectiveness  $\eta$  occurs at  $x/D < 2$ , while a further increase in  $\eta$  appears at  $x/D = 2 \sim 8$ . This is due to the fact that the cooling jets detach from the wall surfaces immediately after effusing from the injection holes and then reattach at the downstream locations after separation. In this region, the highest  $\eta$  appears at  $M = 0.5$ . While in the region of  $x/D \geq 14$ , the  $\eta$  level of  $M = 1.0$  and  $1.5$  is higher than  $M = 0.5$  case because the high  $M$  values bring an enhanceive momentum in coolant, which results in an increase in  $\eta$  at far downstream locations. (2) Although the values of  $\eta$  next to the downstream of injection holes on suction surface are higher than that on pressure surface, the adiabatic effectiveness on pressure surface is much superior to that on suction surface within most of regions because of the strong deflection of coolant near the suction surface. Both the centrifugal force and Coriolis force point towards the high radius locations near the suction surface, which allows more coolant to be deflected upwards. On the pressure surface, the centrifugal effect will be weakened due to the opposite effect of Coriolis force, and it leads to the high values of  $\eta$  finally. (3) As  $R_i$  increases, the values of  $\eta$  both on suction surface and pressure surface decrease simultaneously for the same reason of coolant deflection (see Fig. 4

(b)). The cooling jets from neighboring injection holes can provide additional compensations for the adiabatic effectiveness due to their intense mixing, which can result in the recovery of  $\eta$  in the far downstream region of  $x/D > 18$ . (4) It appears that the augmentation of  $Re$  can lead to a slight decrease of  $\eta$  at the downstream locations (see Fig. 4(c)). To keep a constant value of  $R_i$ , the angular speed should be elevated simultaneously as  $Re$  increases, and which is the exact reason for the reduction of adiabatic effectiveness. (5) After reaching the far downstream locations, the tendencies of  $\eta$  become much uniform on the suction and pressure surfaces.

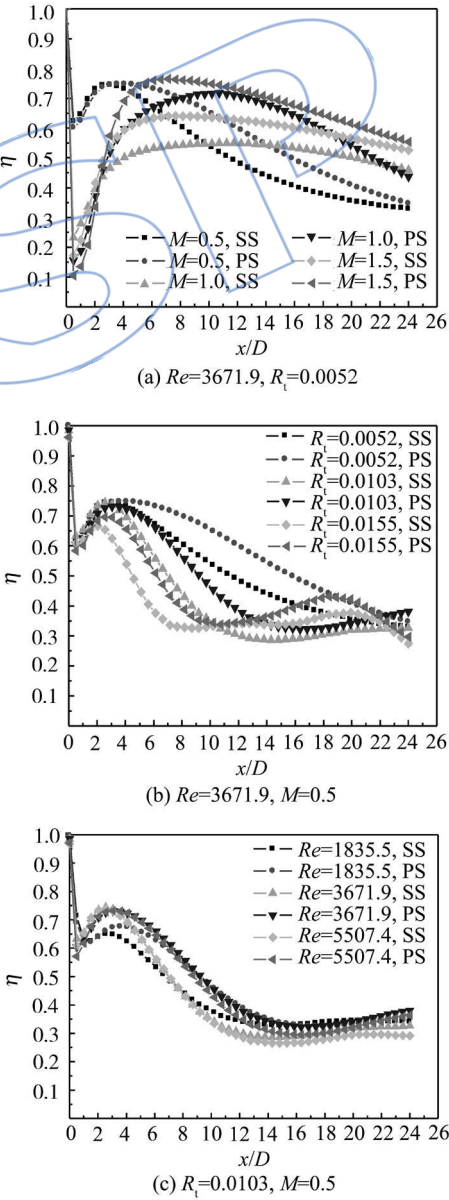


Fig. 4 Values of  $\eta$  downstream of hole 6

7 Heat transfer characteristics

Fig. 5 presents the detailed local Nusselt number distributions on the suction surface at various operating conditions for  $M = 0.5$ . As seen from the figure, the Nusselt number distributions clearly indicate higher  $Nu$  appears along the downstream of each injection holes due to the strong interaction of mainstream and coolant, while lower  $Nu$  appears in the region among the injection holes due to the lack of coolant. The  $Nu$  decreases along the streamline direction to far downstream locations and its magnitude can be evaluated as  $Re$  increases from 1835.5 to 5507.4. When comparing with the results illustrated in Fig. 3 at the same operating condition of  $Re = 1835.5$  and  $R_t = 0.0309$ , it is interesting to note that the phenomenon of coolant deflection is vanished. The reason could be the mainstream temperature and coolant temperature were assumed to be equal during the calculations by using the classical method of Eckert<sup>[27]</sup>.

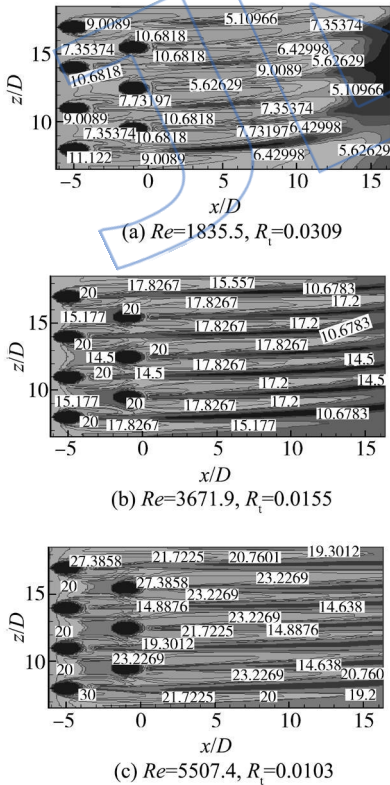


Fig. 5  $Nu$  distributions for  $M = 0.5$

8 Discharge coefficient

Knowledge of the discharge coefficient  $C_d$  is vital in sizing film cooling holes at the design stage to determine the requisite amount of coolant flow to produce the desired cooling effectiveness. The discharge coefficient  $C_d$  is the ratio of actual mass flow rate and ideal mass flow rate through the film cooling hole. Therefore,  $C_d$  can be expressed as

$$C_d = \frac{m}{p_{tc} \left( \frac{p_g}{p_{tc}} \right)^{\frac{k-1}{k}} \left\{ \frac{2k}{(k-1)RT_{tc}} \left[ \left( \frac{p_{tc}}{p_g} \right)^{\frac{k-1}{k}} - 1 \right] \right\}^{\frac{1}{2}} \frac{\pi}{4} D^2} \quad (8)$$

Fig. 6 shows the distributions of  $C_d$  for all the injection holes under different operating conditions. As can be seen in Fig. 6, the  $C_d$  values for each injection hole are quite different, and this denotes that the  $M$  is only a qualitative parameter for the distribution of total coolant mass flux in this study. All the  $C_d$  groups decrease both on suction surface and pressure surface as  $R_t$  is increased for a constant  $Re$ , and the  $C_d$  always increases with increasing  $M$ . This is due to the fact that when the  $M$  increases, the coolant can break away from the repression of mainstream and flow out easily along  $y$  direction. The rotation can always cause the pressure increase along the radial direction in the plenum, and this brings to the accretion of  $p_{tc}$ , leading to an increscent difference between  $p_{tc}$  and  $p_g$ . Finally, the  $C_d$  shows a decrease as the  $R_t$  increases from 0.0103 to 0.0309. It can be seen that the  $Re$  shows no effect on  $C_d$  at a fixed value of  $M$  and  $R_t$  (see Fig. 6(c)), and this result also corresponds with the conclusion by Kim et al<sup>[8]</sup>. The reason could be attributed to the synchronous variation of angular speed to keep the constant value of  $R_t$  when  $Re$  varies. In this figure, it is also demonstrated that the  $C_d$  on pressure surface is much higher than that on suction surface under each operating condition. The Coriolis force drives a majority of coolant to deviate from the suction surface in the plenum, causing more

coolant discharge from the hole-pipes on the pressure surface. Considering that there has no obvious difference in  $p_g$  near the two flat surfaces, as a consequence, the  $C_d$  values on the pressure surface are much higher. In addition, the  $C_d$  shows only a moderate variation along the radial direction, and the values in cooling holes (5) ~ (7) are slightly higher than others.

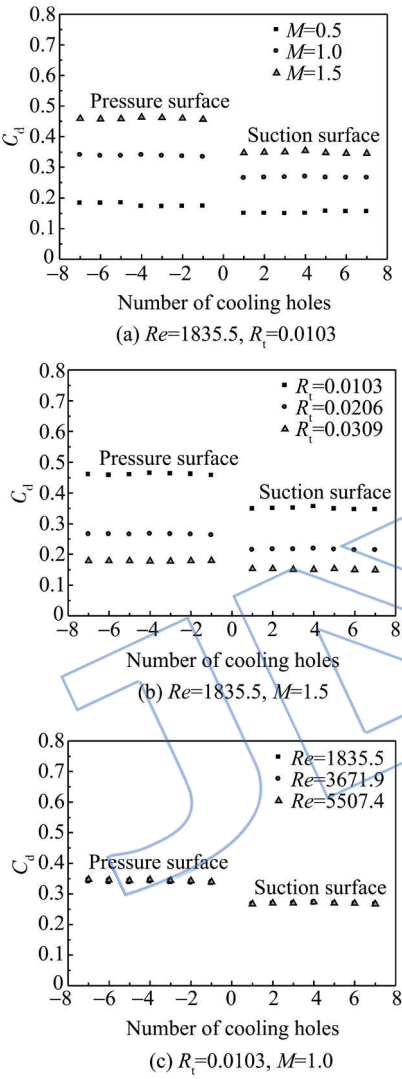


Fig. 6 Discharge coefficients  $C_d$  for all the hole pipes

9 Conclusions

Numerical simulations were performed to investigate the effect of rotation on the film cooling mechanism over a simplified rotating model. The computational model is originated from the mid-span section of a typical turbine rotor with two rows of 14 staggered injection holes on the suction surface and pressure surface, and the

inclination angle to suction and pressure surfaces is fixed at  $\varphi=30^\circ$ . The operating conditions include various values of Reynolds number  $Re$ , rotation number  $R_t$ , and the averaged blowing ratio  $M$ . The key findings of present study can be summarized as follows:

- 1) The rotational effect has a significant influence on the flow field. The centrifugal force and Coriolis force have the same direction near the suction surface to point towards the high-radius locations, but the Coriolis force can weaken the centrifugal effect due to its opposite direction near the pressure surface.
- 2) Although the high values of  $M$  will weaken the thermal protection to wall surfaces near the exit of cooling holes, it can increase the adiabatic effectiveness properly at downstream locations. As  $R_t$  increases, the values of  $\eta$  both on suction surface and pressure surface decrease simultaneously. In addition, the augmentation of  $Re$  can lead to a slight decrease in  $\eta$ .
- 3) For discharge coefficient distributions, in all cases an increase in  $R_t$  leads to a decrease of  $C_d$  for a constant  $Re$ . The augmentation of  $M$  will always aid to elevate the  $C_d$  values. When  $R_t$  and  $M$  are fixed, the  $C_d$  values are independent on the variation of  $Re$  under a given operating condition. Furthermore, the  $C_d$  values on the pressure surface are much higher than that on the suction surface under the same operating condition.

4) Although the method which assumes the mainstream temperature and coolant temperature are equal to obtain the heat transfer coefficient in film cooling has been proved rather correctly in static study both in theoretical and experimental approaches, it is not suitable for the rotating film cooling investigation due to the inaccurate prediction of coolant deflection phenomenon.

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