EFFECTS OF ROTATION ON FILM COOLING EFFECTIVENESS OF A GAS TURBINE ROTOR BLADE WITH CYLINDRICAL AND FAN-SHAPED FILM HOLES

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ABSTRACT
The effects of rotation on cylindrical and fan-shaped hole film cooling for a rotor blade were studied by means of CFD. The averaged blowing ratio is 0.8, and the rotation speed varies from 0 to 5000 rpm. The numerical results show that the film coolant trajectory is consistent with the limiting streamlines of the blade without film cooling under the same rotating speed. The deflection angle of the film trajectory towards the high-radius direction increases with the increase of the rotating speed, and the deflection angle of film trajectory on the pressure surface is much larger than that on the suction surface under the same rotating speed. The cooling effectiveness on suction surface is larger than that on pressure surface, and the suction surface cooling effectiveness reduces with the increase of the rotating speed. But the pressure surface cooling effectiveness increases with the increase of the rotating speed. The cooling effectiveness of fan-shaped hole is better than that of cylindrical hole under the same rotating speed.

INTRODUCTION
Film cooling has been an advanced technology widely used in gas turbine and aero-engine to protect the components from high heat loading. Researchers have studied the effects of flow and geometric parameters, such as blow ratio, momentum ratio, density ratio, hole shapes, the ratio of hole length to diameter, etc., on the film cooling effectiveness (Han, 2013). But most of the numerical and experimental studies were carried out on stationary flat-plate model or linear cascade model. The research on rotating cascade was very few because of the inherent difficulties. Some documents pointed out that the difference of the film trajectory and film cooling effectiveness between the stationary cascade and rotating cascade, but their conclusions are not consistent.

Ito et al. (1978) measured the film cooling effectiveness on a linear-cascade model. They found that the film cooling effectiveness of the suction surface is larger than that of the pressure surface. Dring et al. (1980) investigated the effect of rotation on the film cooling on the rotating cascade model with the shaft speed of 405 rpm. They pointed out that the film deflection on the pressure surface was larger than that on the suction surface, and the influence of rotation on film cooling effectiveness was more obvious on pressure surface than on suction surface. Takeshi et al. (1992) measured the film cooling effectiveness on the stationary linear cascade and the rotation cylindrical cascade by using the heat-mass transfer analogy. The conclusion they got was similar to that of Dring’s. Schobeiri (2006) and Yang et al. (2005), i.e., found that rotation change the position of stagnation line on the leading edge, and this phenomenon influences the distribution of coolant on the suction surface and pressure surfaces. Yang et al. (2007) explored the effect of rotation on the coolant coverage by using the flat-plate model. They found that film coolant on the suction surface always deflects to the high-radius direction under any rotating speed, and the deflection angle increases with the increase of rotating speed. However, the film coolant on the pressure surface deflects to the low-radius direction under low rotating speed and deflects to the high-radius direction under the high rotating speed. Li et al. (2010a) and Li et al. (2010b) found that film coolant on the suction surface deflects to the low-radius direction, but deflection angle is very small, and film on the pressure surface deflects to the high-radius direction.

To identify the influences of rotation on the distributions of film cooling, three dimensional numerical simulations of adiabatic film cooling effectiveness for a rotating blade were...
carried out in this investigation. The cooling effectiveness and flow field of the rotating cascade with different hole shapes and rotating speeds were simulated by using CFX with the SST turbulence model. The flow details and the film deflection are presented and discussed.

CALCULATION SETUP

Numerical Method

The CFX with SST turbulence model was adopted to calculate the three-dimension flow and temperature distribution on the blade surface. The $k-\omega$ model is used to solve the flow near the blade surface and $k-\varepsilon$ is used to solve the bulk flow away from the blade surface, and the blending function ensures a smooth transition between the two model. The SST turbulence has the advantage in predicting the turbulent heat transfer because of the effective capture of the onset of flow separation under adverse pressure gradient conditions.

Flat-plate film cooling model with cylindrical hole was used to validate the numerical method. Figure 1 shows the comparisons of laterally averaged film cooling effectiveness with P/D=3 between the numerical results and the experimental data from Wang (2012) at the blowing ratio of 0.5. It can be found that the numerical results are well consistent with the experimental data.

![Figure 1 Comparisons of laterally averaged film cooling effectiveness between numerical results and experimental data (Wang, 2012) for flat plate film cooling with cylindrical hole(P/D=3)](image)

Geometry of Film Cooling Holes

Figure 2 illustrates the configuration of the blade for the current study. The current numerical study has been carried out by using linear blade with three film cooling holes located at 15.7%, 50% and 84.3% span height on the pressure surface or suction surface, respectively. The row of holes locates at the 18% of axial chord on the pressure surface and 18.3% of axial chord on the suction surface. The ratio of blade span to axial chord length is 1.84. The diameter of the hole is 0.5 mm and the ratio of length to diameter is 4. The film hole incline angles are 30 deg along the streamwise direction and 90 deg along the spanwise direction. The fan-shaped holes expand at the position of L/D=2 and the expansion angle is 14 deg.

Boundary Condition

The air of mainstream is considered as perfect gas. The total temperature of 450 K and total pressure of 130000 Pa were specified at the inlet of main stream and static pressure of 101325 Pa was specified at the out. The distribution of relative flow angle, with attack angle of zero, was specified at main flow inlet. The “Mass Flow Rate” is chosen at the coolant inlet to keep the average blowing ratio of 0.8, and the temperature of coolant at inlet is 300 K. The rotating speed is set to 0 rpm, 3000 rpm and 5000 rpm, respectively.

![Figure 2 Geometry of blade with film cooling holes](image)

Grid and Grid Independence Test

Unstructured grids were used because of the sophisticated geometry. The local unstructured grids used in current work are shown in Fig. 3. The height of the first prism layer near to the wall of blade and film hole is set to 0.001 mm in order to keep $y+$ less than 1. Grid independence test was carried out for the blade by using three different grid numbers at BR of 0.8. The laterally averaged film cooling effectiveness of the hole located on the middle span of the suction surface for P/D=3 are illustrated in Fig. 4. It can be observed that the effectiveness distributions are well agreed with each other between the result with about 15 million elements and that with 35 million elements. Therefore, the former grid is chosen for the following calculations.

![Figure 3 Local unstructured computational grids](image)
Figure 4 Laterally averaged film cooling effectiveness on suction surface for P/D=3

Test Cases

Table 1 lists the test cases carried out in this work. Case1 to Case3 are mainly used to observe the streamlines near the blade surface without the influence of film cooling under different rotating speed. Case4 to Case6 and Case7 to Case9 are used to study the coolant trajectory and cooling effectiveness distribution on pressure surface and suction surface with cylindrical film holes under different speed rotating speeds, respectively. Case10 to Case 15 represent the film cooling on pressure surface or suction surface with fan-shaped holes under different rotating speeds.

<table>
<thead>
<tr>
<th>Case</th>
<th>Hole shape</th>
<th>Hole position</th>
<th>Speed (rpm)</th>
<th>Notation</th>
</tr>
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<tbody>
<tr>
<td>Case1</td>
<td>-</td>
<td>-</td>
<td>0</td>
<td>NP/S_0</td>
</tr>
<tr>
<td>Case2</td>
<td>-</td>
<td>-</td>
<td>3000</td>
<td>NP/S_3000</td>
</tr>
<tr>
<td>Case3</td>
<td>-</td>
<td>-</td>
<td>5000</td>
<td>NP/S_5000</td>
</tr>
<tr>
<td>Case4</td>
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<td>P</td>
<td>0</td>
<td>CP_0</td>
</tr>
<tr>
<td>Case5</td>
<td>C</td>
<td>P</td>
<td>3000</td>
<td>CP_3000</td>
</tr>
<tr>
<td>Case6</td>
<td>C</td>
<td>P</td>
<td>5000</td>
<td>CP_5000</td>
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<td>Case7</td>
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<td>S</td>
<td>0</td>
<td>CS_0</td>
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<tr>
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</tr>
<tr>
<td>Case9</td>
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<td>Case15</td>
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<td>S</td>
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<td>FS_5000</td>
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</tbody>
</table>

RESULTS AND DISCUSSIONS

Limiting Streamlines without Film Cooling

Figure 5 shows the limiting streamlines near blade surface without film cooling under different rotating speed. The limiting streamlines on the suction surface deflect to the high-radius direction and the deflection angle increases with the increase of rotating speed. The streamlines on the pressure surface deflect to the low-radius direction when rotating speed is zero, but the streamlines deflect to the high-radius direction under the rotating speed of 3000 rpm and 5000 rpm, and the deflection angle increases with the increase of rotating speed. The deflections of streamlines on pressure surface are larger than that on suction surface when rotating speed is larger than zero.

Figure 6 shows the 3D streamlines starting from the computational domain inlet at the middle span under rotating speed of 5000 rpm. It intuitively displays the fluid flow near the suction and pressure surface.

Figure 7 shows the variation of flow parameters near the blade surface (0.1 mm away from the surface) along the streamwise direction at middle span. It can be seen from Fig.7a that on the pressure surface the radial velocity is markedly increaded with the increase of rotating speed. The
radial velocity on the suction surface also increases with the rotating speed increase, except near the trailing edge. The streamwise velocity on the suction surface is approximately three times of that on the pressure surface at the same axial position as observed in Fig.7b. And the streamwise velocities both on the suction surface and pressure surface are decreased with the increase of rotating speed. This explains that the deflection of limiting streamlines on pressure surface is larger than that on suction surface at rotating speeds of 3000 rpm and 5000 rpm.

The radial acceleration is defined as,

\[
a_r = \frac{\text{Radial Resultant Force}}{\text{Mass}}
\]

The Radial Resultant Force represents the sum of the radial component of Coriolis Force, Centrifugal Force and Pressure Difference in the radial direction. The three components are defined as follow,

\[
\text{Coriolis Force} = -\frac{\rho V_n \pi n \text{Vol}}{15}
\]

\[
\text{Centrifugal Force} = \rho \text{Vol} \left( \frac{V_z^2 + \left( \frac{\pi n}{30} \right)^2 r}{r} \right)
\]

\[
\text{Pressure Difference} = \left( \text{Pressure}_{\text{down}} - \text{Pressure}_{\text{up}} \right) A
\]

It can be observed from Fig.7c that the radial acceleration is generally increased on the pressure surface with the increase of rotating speed, but it is decreased on the suction surface with the increase of rotating speed.

The deflection of fluid particle in the radial direction can be qualitatively estimated by

\[
\Delta R = V_n t + \frac{1}{2} a_t t^2
\]

The time of fluid particle passing through the blade surface is increased with the increase of rotating speed as indicated in Fig.7b. Both the radial velocity and radial acceleration on the pressure surface are increased with the increase of rotating speed. So it is easy to understand that the radial deflection \(\Delta R\) on the pressure surface is increased when the rotating speed is increased. However, the opposite change of radial velocity and radial acceleration on the suction surface results in small variations in the deflection \(\Delta R\) with the increase of rotating speed.

Coolant Trajectory

Figure 8 shows the limiting streamlines and coolant trajectory represented by the “cold streak” on the blade surface both for cylindrical film holes and fan shaped film holes under different rotating speeds. The coolant is generally carried by the mainstream. However, the coolant trajectory is not fully consistent with mainstream because of the coolant inertia. Table 2 presents the deflection angles of coolant trajectory of Hole 2 on the suction surface and pressure surface for cylindrical and fan shaped film holes under different rotating speeds. As described above, the coolant deflection angle on the pressure surface is much larger than that on the suction surface both for cylindrical and fan shaped holes under rotating speed of 3000 and 5000 rpm. The deflection angle of coolant trajectory for the fan-shaped hole on pressure surface is slightly larger than that of cylindrical hole. This can be explained that the coolant inertia or momentum from the fan-shaped hole is smaller than that from the cylindrical hole at the hole exit. So the coolant from the fan shaped holes is more easily carried by the mainstream.
Adiabatic Film Cooling Effectiveness

The relative total temperature under rotating condition is not evenly distributed along the span. So the whole span is divided into three sections and the spanwise averaged adiabatic film cooling effectiveness is calculated by using the length weighted method.

Figure 9 shows the distributions of spanwise averaged film cooling effectiveness along the streamwise direction on the blade surface for the fan-shaped hole and cylindrical hole under different rotating speed. The averaged cooling effectiveness gradually reduces along the streamwise direction both on the suction and pressure surfaces for a given rotating speed. The film cooling effectiveness on the suction surface at the same axial location is gradually reduced for both cylindrical and fan shaped film holes when the rotating speed increases from 0 rpm to 5000 rpm. The averaged film cooling effectiveness on the pressure surface at the same axial locations show the complicated trends when the rotating speed increases from 0 to 5000 rpm, although the variations at the same axial location are quite small when S/D is larger than 10.

It is can be seen from Figs. 8 and 9 that the expanded exit of fan shape holes reduces the coolant momentum and widens the coolant coverage. It is obvious that the film cooling effectiveness of the fan-shaped hole is generally larger than that of cylindrical hole.

![Figure 8 Limiting streamlines and coolant trajectory on blade surfaces for cylindrical and fan shaped film holes under different rotating speed](image)
Redistribution of Coolant Mass Flowrate

The film holes on the pressure surface or suction surface connect to the same coolant plenum, but the coolant mass flowrate is not evenly distributed among the film holes. Figure 10 shows that the fractions of coolant mass flowrate for the film holes under different rotating speed. The maximum coolant mass flowrate among the three holes on the suction surface varies from Hole 3 (under 0 and 3000 rpm) to Hole 1 (under 5000 rpm), but the coolant mass flowrates among the three holes are changed a little and kept between 0.31 and 0.35. However, the maximum coolant mass flowrate on the pressure surface varies from Hole 1 (under 0 rpm) to Hole 3 (under 3000 and 5000 rpm), and the coolant mass flowrates among the three holes are changed a lot. The coolant mass flowrate of Hole 1 on pressure surface is extremely small under the rotating speed of 5000 rpm, and the coolant coverage of Hole 1 is very poor, as shown in Figs.8j and 8m.

The pressure changes at the inlet (plenum) and outlet of film holes under different rotating speed are the main reasons of the redistribution of coolant mass flowrate among the film holes. Figure 11 shows the pressure (p/p2) distributions in the coolant plenum and near the holes on the pressure surface. The pressure ratio keeps uniform along the spanwise direction in the plenum under rotating speed of zero, meanwhile the pressure at the hole exit slightly decreases from Hole 1 to Hole 3 along the spanwise direction, so the coolant mass flowrate is mildly increased from Hole 1 to Hole 3 (Fig.10b). With the increase of rotating speed, the pressure in the coolant plenum and near the holes on the pressure surface increases along the spanwise direction due to the effects of centrifugal force. The pressure difference between the film hole inlet and the outlet among the three film holes on the pressure surface varies from Hole 1 under 0 rpm to Hole 3 under 3000 rpm and 5000 rpm. So the coolant mass flowrate for the Hole 3 is the maximum under the rotating speed of 3000 rpm and 5000 rpm.
The film coolant trajectory and film cooling effectiveness for both cylindrical holes and fan-shaped holes under the rotating speed of 0 rpm, 3000 rpm and 5000 rpm. The following conclusions are drawn,

1) The film coolant is mainly carried by the mainstream, so the coolant trajectory is consistent with the limiting streamlines. The deflection of coolant trajectory is towards the high-radius direction both on the suction surface and pressure surface under rotating speed of 3000 rpm and 5000 rpm. The deflection angle is increased with the increase of rotating speed, and the deflection angle on the pressure surface is much larger than that on the suction surface.

2) The film cooling effectiveness on the suction surface is reduced for both cylindrical and fan shaped film holes with the increase of rotating speed. The film cooling effectiveness on the pressure surface shows complicated trends with the increase of rotating speed, and the variations caused by rotating are not as obvious as on the suction surface.

3) There are redistributions of coolant mass flowrate among the three film holes on the suction or pressure surface under different rotating speed, due to the changes of pressure at the inlet and outlet of the film holes. The differences of coolant mass flowrate for the film holes on the suction surface are within 4% under different rotating speed studied. In contrast, the differences of coolant mass flowrate for the film holes on the pressure surface are up to 50% under rotating speed of 5000 rpm, which lead to unsatisfactory coolant coverage for the film hole near the hub.

NOMENCLATURE

- \( A_{hole} \): cross sectional area of film cooling hole
- \( A \): area of each surface of assumptive control volume
- \( a \): radial acceleration
- \( BR \): blowing ratio
- \( C \): cylindrical hole
- \( D \): diameter of film cooling hole
- \( d \): axial chord length of blade
- \( F \): fan-shaped hole
- \( L \): length of film cooling hole
- \( N \): number of film cooling hole
- \( n \): rotating speed
- \( P \): pressure surface or laterally distance between holes
- \( p \): static pressure
- \( S \): suction surface or streamwise distance
- \( T_c \): temperature of coolant
- \( T_{total} \): temperature of wall without film cooling
- \( T_w \): temperature of wall with film cooling
- \( t \): time
- \( u \): velocity
- \( Vol \): volume of assumptive control volume
- \( V_{rr} \): initial radial velocity
- \( V_r \): circumferential velocity
- \( z \): axial distance
- \( \Delta R \): deflection in the radial direction
- \( \eta \): laterally averaged film cooling effectiveness
- \( \theta \): angle
- \( \rho \): density
- \( m \): main flow
- \( 2 \): main flow outlet

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