Effects of Anticlockwise Swirling Inflow on the Aerothermal Performance of the Turbine Vane Endwall with Film Cooling Layouts

Zhiyu LI
Xi’an Jiaotong University, Institute of Turbomachinery
lzy001@stu.xjtu.edu.cn
Xi’an, Shaanxi, China

Kaiyuan ZHANG
Xi’an Jiaotong University, Institute of Turbomachinery
zhang_kaiyuan@126.com
Xi’an, Shaanxi, China

Zhigang LI
Xi’an Jiaotong University, Institute of Turbomachinery
zhigangli@mail.xjtu.edu.cn
Xi’an, Shaanxi, China

Jun LI*
Xi’an Jiaotong University, Institute of Turbomachinery
Corresponding author:
junli@mail.xjtu.edu.cn
Xi’an, Shaanxi, China

Liming SONG
Xi’an Jiaotong University, Institute of Turbomachinery
songlm@mail.xjtu.edu.cn
Xi’an, Shaanxi, China

ABSTRACT

The large swirling flow downstream the lean burn premix combustor significantly impacts on the aerothermal performance of the first stage turbine vane endwall. The effects of the anticlockwise swirling inflow on the aerothermal performance of the turbine vane endwall with film cooling layouts was numerically investigated using three-dimensional Reynolds-Averaged Navier-Stokes (RANS) and SST turbulence model. The heat transfer characteristics and film cooling effectiveness of the turbine vane endwall with three rows film holes along the axial direction at three kinds of inlet swirl number 0.6, 0.8 and 1.0 was analyzed and compared. The obtained results show that the anticlockwise swirling inflow leads to the horseshoe vortex of the vane endwall migrate to the downstream and generate the recirculation vortex of the mainstream by comparison to the uniform inflow condition. The anticlockwise swirling inflow changes the Nusselt number distribution of the vane endwall and results in the high heat transfer coefficients regions. The high heat transfer regions at the vane endwall increases with the swirling inflow strength increases. The anticlockwise swirling inflow enhances the Nu downstream the first row film holes at the x/Cax=0.31 and suppresses the coverage of the coolant jet from the film holes migration to the suction side. Comparison to the uniform flow condition, the averaged film cooling effectiveness of the first row film holes increases up to 27% and the third film holes decreases about 9%.

INTRODUCTION

Modern aircraft engines are equipped with lean premixed combustor for meeting the environmental protection and emission requirements. Swirl core generated by combustor migrates downstream to the turbine inlet which results in the high pressure turbine stage inlet has swirling flow characteristic. Swirling flow changes secondary flow pattern and influents cooling performance on the endwall. Therefore, swirling inlet condition brings new challenges for the design of turbine vane end wall film cooling (Johnson, B et al. 2014; Lazik, W et al. 2008).

As one of the most efficient cooling methods for turbine machinery, film cooling technology has been implemented widely and some related research have been carried out. The aerodynamic and cooling performance of the turbine vane endwall was given by (Friedrichs et al., 1997) and they pointed out that the film cooling layout of the endwall needs to consider the complicated secondary flow vortex structure and the intermixing effect of mainflow and film cooling airflow. In (Shiau et al. 2019), they used linear cascade experiments to study the heat transfer and cooling performance of the endwall with 3 rows of upstream film holes and 5 film cooling layouts. The results showed that the film cooling layout combined with the development characteristics of the secondary flow vortex system on the endwall had the excellent cooling performance. G. Barigozzi (G. Barigozzi, et al., 2018) reported an experimental investigation on aerodynamic and heat transfer performance of a high pressure nozzle vane cascade without and with different mass flow rate platform film cooling. In terms of numerical calculation research, Du (Du et al., 2017) carried out a study on the influence of the arrangement of
film holes near the front edge of the vane endwall on the cooling performance. Studies have shown that the film hole with a positive incident angle has a better cooling effectiveness on the endwall as well as the suction side of the vane. The cooling performance of the end wall with slot injection and slashface leakage under different misalignment modes were developed by Zhang K through numerical method (Zhang, K et al., 2019).

Influenced by the complex exit conditions of the combustion chamber, turbine inlet conditions are usually not uniform but with strong swirl characteristics. Therefore, a lot of research work on the interaction between the combustion chamber and the turbine has been carried out in recent years. On the advanced experimental platform Oxford Turbine Research Facility (OTRF), Qureshi (Qureshi et al., 2012; Qureshi et al., 2013) studied the aerodynamics and heat transfer performance of high pressure turbine stages under inlet swirl conditions and found that Nusselt number of endwall was increased a lot under swirl inlet conditions. The robustness of the upstream endwall film cooling to swirling combustor inflow was investigated by Werschnik (Werschnik et al., 2017) on the Large Scale Turbine Rig (LSTR) at TU Darmstadt. The film cooling effectiveness was reduced by 30% according to the experimental results. As an effective analytical tool, numerical method has been used for solving inlet swirling problem. The effect of different inlet swirl intensities and orientations on the aerothermal performance of blade endwall with misalignment was reported by Zhang K (Zhang, K et al., 2019). In addition, Salvadori (Salvadori et al., 2012) compared the influence of different total temperature profile and swirl distribution on the film cooling characteristics of the high pressure turbine vane endwall by numerical method.

The interaction between the combustion chamber and the turbine must be considered for the aerothermal design of the gas turbine first stage vane. Therefore, the study that the influence of aerothermal and film cooling performance on turbine vane endwall under the strong swirling inlet needs in-depth research. Based on the experimental measurement of the linear cascade model, the aerodynamic, heat transfer and film cooling characteristics of endwall with the three rows of film holes under the anticlockwise direction by numerical method is carried out in this paper. Furthermore, the impact of swirling intensity is also analysed at three typical kinds of swirl number 0.6, 0.8 and 1.0 aiming to provide a reference for the design of the film cooling layout of the end wall of the high thermal load cascade under swirling inlet conditions.

### NUMERICAL MODEL AND METHOD

#### Geometric model

Figure 1 shows the experimental vane cascade model from (Shiau et al., 2019) of the Turbine Laboratory of Texas A&M University. Table 1 lists the main geometric parameters of the experimental vane cascade and film holes. There are three rows of film holes arranged on the vane endwall along the flow direction at $X/C_w=0.31$, 0.64, 1.0. Each row is arranged with 6 film holes with a diameter of 1.5mm, a total of 18 film holes and the distance between adjacent holes is 0.9092mm.

#### Numerical method

The inlet boundary condition was set to be uniform inflow in the experiment. In this paper, in order to study the influence of swirling inflow and intensity on the aerothermal and film cooling performance of the vane cascade endwall, the swirl inlet condition is established as shown in Figure 2. The swirl core is set at the centre of vane passage with a diameter of 0.4H. And the velocity of...
it is composed of axial velocity and tangential velocity. According to (Schmid and Schiffer, 2012), the definition of swirl Number is given for evaluating the swirling intensity and defined as follows:

\[
SN = \int_0^{r_0} \left( \rho \frac{u_a}{u_m} \right) r^2 dr
\]

where \( r_0 \) is the radius of inlet swirl, \( \rho \) is density of mainstream, \( u_a \) and \( u_m \) are axial component velocity and tangential component velocity of inlet swirl, respectively. Under the premise of keeping the mass flow constant, swirl number is variable by changing the speed ratio of \( u_a \) and \( u_m \). Three typical kinds of swirl number \( SN=0.6, 0.8, 1.0 \) as well as uniform inlet \( (SN=0) \) are selected in this study.

\[
\eta = \frac{T_{\infty} - T_{aw}}{T_{\infty} - T_c} \quad (2)
\]

where \( T_{\infty} \), \( T_{aw} \) and \( T_c \) is mainstream inlet temperature, endwall temperature for adiabatic condition and coolant inlet temperature, respectively. Furthermore, heat flux \( q \) and \( T_{aw} \) can be obtained under isothermal endwall conditions. Then the Nusselt number \( (Nu) \) can be calculated according to equation (3) and (4).

\[
h = \frac{q}{(T_{aw} - T_c)} \quad (3)
\]

\[
Nu = hC / \lambda \quad (4)
\]

where \( h \) is heat transfer coefficient and \( \lambda \) is fluid thermal conductivity.

**Table 2 Computational boundary conditions**

<table>
<thead>
<tr>
<th>Boundary condition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mainstream mass flow rate ( M ) /kg.s⁻¹</td>
<td>0.82</td>
</tr>
<tr>
<td>Mainstream inlet temperature ( T_{\infty} ) /K</td>
<td>298.0</td>
</tr>
<tr>
<td>Mainstream turbulence /%</td>
<td>10.0</td>
</tr>
<tr>
<td>Coolant mass flow rate ( m ) /kg.s⁻¹</td>
<td>0.0082</td>
</tr>
<tr>
<td>Coolant inlet temperature ( T_c ) /K</td>
<td>278.0</td>
</tr>
<tr>
<td>Outlet static pressure ( P_s ) /Pa</td>
<td>98000.0</td>
</tr>
<tr>
<td>Swirl number /SN</td>
<td>0,0.6,0.8,1.0</td>
</tr>
<tr>
<td>Swirl direction (looking downstream)</td>
<td>anticlockwise</td>
</tr>
</tbody>
</table>

Table 2 shows the detail boundary conditions of computational domain. Apart from swirl number, other parameters of boundary conditions are constants in all cases. The adiabatic and isothermal endwall boundary conditions are set respectively for evaluating film cooling and heat transfer performance. The isothermal end wall temperature \( T_{aw} \) used in this work is 275K. Periodic boundaries are set on both sides of computational domain, and the rest of the walls are set as adiabatic and non-slip walls. Adiabatic temperature of endwall can be achieved under adiabatic endwall boundary condition for calculating the film cooling effectiveness which is defined as:

\[
\eta = \frac{T_{\infty} - T_{aw}}{T_{\infty} - T_c} \quad (2)
\]

Figure 3 shows the calculation grid of the vane cascade and the film cooling holes. ANSYS ICEM is used to generate a multi-block structured grid, and O-shaped grids are used for a better mesh quality. The height of first layer for all case walls is less than 0.001 mm to ensure \( y^* < 1.0 \) which meets the requirements of the SST turbulence model. The total number of nodes is adjusted synchronously in three coordinate directions with the same stretching ratio and six different grid numbers are used to conduct the grid independence tests. As shown in
Figure 4, a total of 8.5 million nodes can predict laterally averaged $\eta$ accurately enough. Therefore, a total of 8.5 million nodes is selected for all cases in this work.

The reliability of the numerical method and turbulence model are verified through comparing with experimental data (Shiau et al., 2019) under the uniform inlet condition solved by CFX commercial software. Standard $k-\omega$ model, standard $k-\varepsilon$ model, RNG $k-\varepsilon$ model and SST $k-\omega$ model are selected in this comparison. Figure 5 and 6 give the comparisons of film cooling distribution and laterally averaged film cooling effectiveness of end wall respectively. And SST $k-\omega$ turbulence model has the best prediction of film cooling performance of endwall especially at section 1 and 2. It should be pointed out that the higher prediction of the $\eta$ at the position of $X/C_{ax}>0$ by all turbulence models is due to the influence of the neighboring passages film cooling under the periodic boundary conditions and there was no film cooling of neighboring passages in the EXP (see the difference in red triangle between EXP and CFD in Figure 5). Therefore, SST $k-\omega$ turbulence model is chosen for all computational cases in this paper.

RESULTS AND DISCUSSION
Effects of swirling inflow on aerodynamics

Figure 7 shows the flow pattern near the endwall under the uniform inlet and the anticlockwise swirling inlet with $SN=0.6$.

Obviously, flow pattern for the vane endwall has been significantly changed by swirling inlet according to Fig. 7. There is a mainstream recirculation zone near the inlet as a typical characteristic for swirl generator. On the other hand, instead of flowing straight into the vane passage under the uniform inlet condition, mainstream flows into the passage with a positive incident angle under
the swirl inlet condition after attaching vane endwall. Therefore, mainstream attachment results in heat transfer of endwall enhanced and positive incident angle makes the horseshoe vortex move to downstream along the vane pressure side. Furthermore, pressure side leg of the horseshoe vortex is strengthened and influenced the flow of coolant. The streamline of coolant moves away from the endwall surface can be observed in Figure 7(b).

For investigating the influence of $SN$ on endwall aerodynamics, the comparison of flow structure under different $SN$ boundary conditions is presented in Figure 8. As the $SN$ increases, the horseshoe vortex moves from leading edge to downstream along the vane pressure side and eventually mixes with coolant as shown in Figure 8(d) red circle. The film cooling jet is affected by the horseshoe vortex mixing with a larger action area, and more cooling jets are entrained and taken away from the endwall to develop downstream.

Comparison of the surface flow structure at the XZ plane upstream of the vane leading edge under different inlet conditions is shown in Figure 9. Corner recirculation vertex near the inlet is generated by mainstream recirculation under the swirl inlet conditions and recirculation vortex will occur when $SN$ is bigger than 0.8. They are both typical characteristics under swirl inlet conditions for getting a stable flame. However, high heat transfer zone is appeared near the attachment point. Due to the horseshoe vortex moves downstream according to Figure 8, when $SN$=0.8 and 1.0, the complete horseshoe vortex and secondary vortex cannot be fully captured at this plane, and the smaller vortex system formed is the suction side leg of the horseshoe vortex.

Figure 8 Flow patterns near endwall for different inlet conditions

Figure 9 The upstream flow structure of the vane leading edge with different inlet conditions

Figure 10 Flow field structure downstream of the first row film hole with different inlet conditions

Figure 10 shows the flow structure downstream of the first row of film hole on vane endwall under the different inlet condition. The YZ plane is set at the axial position of $X/C_{ax}=0.36$. When the inlet flows uniformly, the flow of this plane is only a simple lateral flow formed by the influence of the pressure gradient. Differing from uniform
inlet, the mainflow vortex can be observed clearly under the swirling inlet. While the mainstream vortex develops from the inlet to the downstream, momentum is gradually dissipated. Therefore, the larger the $SN$, the greater the intensity of the mainstream vortex at this plane and the farther away from the endwall. At the same time, under the influence of the mainstream vortex, a smaller vortex will be generated downstream of each film hole and causes the increase of the heat transfer coefficient.

**Effects of swirling inflow on heat transfer**

Contours plot of $Nu$ for vane endwall under different inlet conditions are shown in Figure 11. Obviously, the $Nu$ of vane endwall is increased by swirling inflow. Specifically, because of the attachment of the mainflow at $X/C_x = -2$ as described in aerodynamic part, the $Nu$ of region A is increased a lot. At the same time, the anticlockwise swirling inlet increases the vane incident angle, resulting in a higher heat transfer region B along the flow direction, and a low heat transfer region C with less airflow. With the increase of the $SN$, the recirculation vortex is strengthened and the two regions A and B gradually develop and expand downstream, causing the area of the low heat transfer region C to gradually decrease. As for the region D downstream of the cascade, affected by the swirling effect, the mainstream gas is more likely to entrain the coolant to develop downstream, thus increasing the $Nu$ of this region.

The distributions of the laterally average $Nu$ along the axial position on the endwall under different inlet conditions are compared in Figure 12. Affected by the mainflow attachment on the endwall, region A is an extremely high heat transfer area with maximum $Nu$ over 10000. Furthermore, the peak value of $Nu$ in this area increases with $SN$ increases. The average vane endwall $Nu$ upstream of the leading edge ($X/C_x < 0$) under swirling inlet conditions are about 4 times that under the uniform inlet condition. It is worth pointing out that the reason for the difference between $SN=0.8$ and $SN=1.0$ is larger than that between $SN=0.8$ and $SN=0.6$ is a recirculation zone has been generated at the $SN=1.0$. The interaction between endwall boundary layer and swirling mainflow is strongly increased results in the $Nu$ is greatly increased when $SN=1.0$. For another aspect, the swirl inflow also enhances the laterally average $Nu$ downstream of every film hole in the passage and this trend is intensified with $SN$ increase. The reason is that the vortex behind the film hole produced by the mainstream vortex mentioned in aerodynamic part enhances the heat transfer of endwall. Under the swirl inflow conditions, the average $Nu$ downstream of the first row film hole is increased by 1.5 times.

Figure 12 Laterally averaged $Nu$ distribution along axial direction with uniform and swirling inlet

**Effects of swirling inflow on film cooling**

Figure 13 shows the contour of film cooling effectiveness distribution on the endwall under the uniform inlet and the swirling flow inlet conditions. The $\eta$ between the first and the second film holes is increased under the swirling inlet conditions and it can be explained from two parts. For the size of coverage area, film cooling coverage area is expanded under swirling inlet conditions especially for the hole nearing the suction surface. Meanwhile, compared with uniform inlet condition, the high $\eta$ can be achieved near the downstream of the first row. From the boundary position where the $\eta$ of the second row downstream is 0 (marked by the red dashed line in the Figure 13), it can be seen that the effective coverage area of the coolant on the endwall is easier to develop downstream along the passage, and coolant will not deviate to the suction surface due to the lateral
pressure gradient. It is because the swirl core is directly on the center of the cascade passage, which weakens the influence of the lateral pressure gradient in the cascade passage. However, as the SN number increases, the tendency of the coolant deflection to the suction surface will increase again.

The average $\eta$ of the endwall between the first row and the second row is increased from 0.11 to 0.14, which is an increase of 27% under the $SN=1.0$ compared to uniform inlet condition. Within 0.5 times the chord length downstream of the third row, there is a 9% reduction of averaged $\eta$ of endwall from 0.33 under uniform inlet to 0.30 under $SN=1.0$ swirling inlet. It is due to the mainstream swirling flow and the vortex downstream of the first row enhance the $\eta$ downstream of the first row, but this effect leads to coolant from the third row cannot cover the surface of the endwall, resulting in a sharp drop of $\eta$.

**CONCLUSIONS**

This paper carried out a research on the effects of anticlockwise swirling inflow on the film cooling performance and aerothermal characteristics of the turbine vane endwall. The flow structure, heat transfer characteristics and film cooling effectiveness of the vane endwall under uniform inlet and swirl inlet conditions with three swirl number are compared and analysed. The following conclusions can be achieved:

1. Compared with uniform inlet condition, mainflow recirculation and attachment are the typical characteristic under swirl inlet conditions. The incident angle is increased and the horseshoe vortex moves to downstream along the pressure surface under the swirl inlet conditions.

2. The heat transfer of endwall is increased a lot under swirling inlet condition especially at the mainflow attachment region. The average $Nu$ of the upstream endwall is increased by 4 times compared with it under the uniform inlet condition. At the same time, the endwall average $Nu$ downstream of the vane is also increased slightly. These phenomena is more obvious under higher $SN$ condition.

3. The tendency of the coolant introduced from film holes flows to suction surface is suppressed by swirling inflow. Effected by mainflow vortex and vortex downstream of film holes, the $\eta$ of first row is increased but $\eta$ of the third row is decreased. The average $\eta$ of endwall is increased by 27% for the area between the first and the second row and reduced by 9% for the area within 0.5$C_{ax}$ downstream of the third row.

**ACKNOWLEDGMENTS**

This work was supported by the National Natural Science Foundation of China (No. 51936008 and No. 51776151).

**NOMENCLATURE**

$C$  
Chord

$C_{ax}$  
Axial chord

$d$  
Diameter of film hole

$h$  
Heat transfer coefficient

$H$  
Height of vane

$m$  
Coolant mass flow rate

$M$  
Mainstream mass flow rate

$Nu$  
Nusselt number
$P$  
Pitch

$q$  
Isothermal endwall heat flux

$SN$  
Swirl number

$T_{aw}$  
Wall temperature for adiabatic

$T_{i}$  
Coolant inlet temperature

$T_{w}$  
Wall temperature for isothermal

$T_{e}$  
Inlet temperature

$\alpha$  
Inlet flow angle

$\beta$  
Outlet flow angle

$\phi_1$  
Film hole jet angle

$\phi_2$  
Film hole compound angle

$\lambda$  
Fluid thermal conductivity

$\eta$  
Film cooling effectiveness

REFERENCES


