Effect of Casing Movement on Flow and Heat Transfer Characteristics of Grooved Blade Tip

Yunsong Zhang¹, Yongbao Liu¹, ²*, Yujie Li¹, ² and Xing He¹, ²

¹ College of Power Engineering, Naval University of Engineering, Wuhan, China
² Military Key Laboratory for Naval Ship Power Engineering, Naval University of Engineering, Wuhan, China

*Corresponding author e-mail: yongbaoliu@aliyun.com

Abstract. Using numerical simulation method, the influence of casing movement on the flow and heat transfer characteristics in the groove tip has analysed, and the influence of rotation speed and blowing ratio was also considered. The results show that the casing movement reduces the casing temperature at the top of the rib and the Nu peak number at the bottom of the groove, it also reduces the mixing loss from the interaction of coolant and leakage flow. The higher the casing motion speed, the lower the average Nu on the wall surface and the smaller the mixing loss in the gap. Large blowing ratio cooling jet can effectively improve wall heat transfer.

1. Introduction

There must be a gap between unshrouded turbine blade tip and casing, high-speed leakage flow through the gap under the action of pressure difference, induces significant heat loads and efficiency loss[1]. Therefore, improving the flow and heat transfer performance in the tip region of the blade is a problem that turbine designers need to pay close attention to.

Turbine groove tip is one of the typical ways to control clearance leakage flow and improve tip heat transfer. Kwak et al [2] indicated that the suction side single rim has a lower wall heat transfer coefficient than the double rim with experimental methods. Ma [3] investigated the influence of turbine cascade blade tip configuration on aerodynamic loss by experimental method, further investigated the mechanism of reducing aerodynamic leakage loss by grooved blade tip. Wang et al [4] studied the tip rim width and found that increasing the rim width reduces the separation vortex in the tip groove and increases the average HTC of the tip wall.

By arranging the film cooling holes on the bottom of the groove b, the cooling gas can effectively block the direct contact of the high-temperature fluid to the wall surface [5]. Christophel et al [6] found that the film cooling efficiency increases with the increase of blowing ratio at the large tip gap height, while HTC increases with the increase of blowing ratio at the small tip gap height. Zhou et al [7] studied the influence of the cooling holes location on the heat transfer characteristics, and found that the heat transfer effect of cooling injection in the tip separation bubble region is better than the reattachment location of leakage flow.

Tip leakage flow affected by centrifugal force, Coriolis force and relative rotation of shroud. Yang et al [8] used numerical methods to study the flat tip and groove tip at the same rotating speed, and found that the relative rotation of the end wall plays a major role in tip leakage flow. Palafox et al [9] found
that the relative rotation of the tip wall changes the shape and size of the tip flow separation bubble and increases the Nusselt number near the tip wall, and this effect increases as the tip clearance decreases. Rezasoltani et al [10] investigated the four tip cooling models under different end wall rotation speeds. It was found that at the same rotation speed, the tip average film cooling efficiency increased with the increase of blowing ratio, while at the same blowing ratio, the tip average film cooling efficiency decreased with the increase of rotation speed. In this paper, numerical method is used to study the influence of casing movement on the flow and heat transfer performance in the gap of groove with air jet cooling on the pressure side, taking into account the influence of rotating speed and blowing ratio.

2. Numerical methods

2.1. Geometry and computational grid

As shown in figure 1(a), the calculation model in this paper approximately simulates the turbine blade tip groove clearance, with clearance height t=0.5mm, rib width w=1mm, bottom width W=10mm, groove height H=1.5mm and depth perpendicular to the paper surface is 2mm. The film cooling hole is located at the bottom of the groove. The most widely used round hole is adopted. The inclination angle α between the film cooling hole and the bottom surface. Cooling hole length is 2mm, the radius of the cooling hole is 0.254mm and the entrance of the model has a wedge angle of 70°. As shown in fig. 1(b), the computational grid is divided into unstructured grids and the grid at the outlet of the hole and the wall surface are encrypted to ensure the first layer grid y⁺<1.

2.2. Boundary conditions

The two planes perpendicular to the Y axis of the main flow are set as periodic symmetry planes. The main flow inlet direction is set to be perpendicular to the arc surface, and the cold air inlet direction is negative to the Z axis. The casing wall is set with a speed along the negative direction of the x axis while the blade tip remains stationary to simulate the relative rotation of the moving blades. Inlet total temperature is 709K, inlet total pressure is 168kPa, outlet static pressure is 120kPa, coolant inlet total temperature is 344K and the inlet velocity of coolant defined by blowing ratio M. The speed of relative casing motion N=300m/s and the blade wall temperature is 496K.

2.3. Definitions

In this study, the bowing ratio is defined as

\[ M = \frac{\rho_c V_c}{\rho_m V_m} \]  

\( \rho_c \) and \( \rho_m \) indicates density of coolant and mainstream, \( V_c \) and \( V_m \) \( \rho_m \) indicates velocity of coolant and mainstream.

The Nusselt number is defined as
\[ \text{Nu} = \frac{qT}{k(T_w - T_i)} \] (2)

Where \( q \) is the heat flow rate, \( t \) is the gap height, \( k \) is the gas thermal conductivity, \( T_w \) and \( T_i \) is the wall temperature and the mainstream temperature respectively.

Total pressure coefficient is defined as
\[ C_{pt} = \left( \frac{P_{ave}^{ave} - P_o}{P_{ave}^{ave} - P_2} \right) \] (3)

\[ P_{ave}^{ave} = (m_in * P_{ave} + m_{jet} * P_{ave, jet})/ (m_in + m_{jet}) \] (4)

Where \( P_{ave}^{ave} \) represents the mass average of mainstream inlet total pressure and coolant inlet total pressure, \( m_in \) and \( m_{jet} \) represents the mass flow of mainstream and coolant, \( P_{ave} \) and \( P_{ave, jet} \) represents mainstream inlet total pressure and coolant inlet total pressure, \( P_o \) represents the total pressure at the measuring point, \( P_2 \) represents the static pressure at the lower end wall of the outlet.

The dimensionless temperature is defined as
\[ T_0 = \left( T - T_c \right) / \left( T_i - T_c \right) \] (5)

Where \( T_i \) and \( T_c \) represents the mainstream temperature and the coolant temperature.

2.4. Numerical method validations

Based on the experimental results of groove gap heat transfer without considering film cooling by Metzger et al [11], Nusselt number under different turbulence models are calculated numerically as shown in figure 2.

![Comparison of numerical and experiment data](image)

Figure 2. Comparison of numerical and experiment data

The \textit{SST}\gamma-\theta \ turbulence model has the best agreement between the numerical calculation results and the experimental results. The \textit{k - \omega} \ turbulence model overestimates the heat transfer effect.
3. Results and discussion

3.1. Effect of relative casing motion

Figure 3. Effect of casing motion on the flow and heat transfer characteristics

Figure 3 shows the effect of casing motion on the flow and heat transfer characteristics in the groove gap, the figure gives the Mach Number in the middle section, the Nusselt number of groove wall, the total pressure coefficient at the outlet and the dimensionless temperature distribution of the casing respectively. As shown in figure 3, the leakage flow impacts the bottom surface of the groove to form a local high Nusselt Number region. The viscous force generated by the casing movement hinders the flow of the leakage flow, and the leakage flow is pressed down at the gap inlet, thus increasing the Nu at the top of the inlet rib and reducing the peak value of the nu at the bottom. When the casing is stationary, the leakage flow shrinks and impacts the casing at the top of the rib, causing a local high temperature region, but a local low temperature region at the middle due to the influence of eddy current. The total pressure coefficient at the outlet of the gap is higher when the casing movement, it indicates the mixing loss in the gap is smaller.

Figure 4. 3D Turbulent Kinetic Energy Vortex Structure Distribution

Figure 4 shows a three-dimensional turbulent kinetic energy vortex structure distribution. Due to the influence of secondary flow on the wall surface, obvious high turbulent kinetic energy vortex structure is generated when the casing moves. The cooling jet and leakage flow form strong mixing at the outlet of the film hole, especially when the casing is stationary.
3.2. Effect of rotating speed

As shown in figure 5, with the increase of casing motion speed, the vortex phenomenon at the gap inlet becomes more obvious, the position where the leakage flow impacts the bottom surface is closer to the pressure surface, the flow velocity at the outlet decreases, and the total pressure coefficient increases. With the increase of motion speed, the influence of leakage flow is smaller on the casing wall temperature, the temperature distribution is more uniform.

Figure 6(a) shows the distribution of Nu along the wall flow direction. The positions between the dashed lines indicate the rib sides, and the solid red lines indicate the outlet of the hole. With the increase of casing motion speed, the peak value of Nu at the bottom and the value of Nu in the first rib corner region decrease, the heat transfer is improved, but the overall effect on the heat transfer in the second rib region is small. Figure 6(b) shows the overall performance under different working conditions. With the increase of casing motion speed, the average heat transfer coefficient on the groove wall...
decreases while the average total pressure coefficient increases. It indicates that the wall heat transfer is improved and the mixing loss is reduced at high rotating speed.

3.3. Effect of blowing ratio

Figure 7 shows the distribution of $Nu$ along the wall flow direction with different blowing ratio (casing speed is $N$). $M=0.0$ represents that there is no film cooling, the existence of cooling jets effectively reduces $Nu$ on the side wall surface of the first rib and the bottom of the groove. With the increase of blowing ratio, the cooling airflow is pressed down by leakage flow, which makes the side wall surface of the first rib obtain better cooling effect. The cooling gas effectively reduces the $Nu$ at the impact position of leakage flow, making the distribution of $Nu$ more uniform, but the cooling gas has less influence on the second rib region.

![Figure 7](image)

**Figure 7.** Effects of blowing ratio on heat transfer in grooved tip

4. Conclusion

The secondary flow generated by the casing movement reduces the flow speed and impact of leakage flow, reduces the casing temperature at the top of the rib and $Nu$ peak value at the bottom of the groove. The movement of the casing promotes the formation of vortex in the gap, resulting in local high turbulence area and reducing the mixing loss from the interaction of cooling jet and leakage flow. The increase of casing motion speed reduces the average $Nu$ on the wall surface and the mixing loss in the gap. The cooling jet with large blowing ratio has the best cooling effect on the groove wall surface and $Nu$ distribution is improved.

Acknowledgments

This work was financially supported by national defense pre-research fund.

References

[1] R. S. Bunker. Axial Turbine Blade Tips: Function, Design and Durability[J].AIAA Journal of Propulsion and power, 2006, 22(2): 271-285.

[2] J. S. Kwak, J. C. Han. Heat transfer coefficients of a turbine blade-tip and near- tip regions[J]. Journal of Thermophysics Heat Transfer, 2003, 17: 297-303.

[3] H. Ma and L. Wang. Experimental study of effects of tip geometry on the flow field in a turbine cascade passage. Journal of Thermal Science, 2015, 24(1): 1-9.

[4] J. Wang, B. Sunden, M. Zeng. Influence of different rim widths and blowing ratios on film cooling characteristics for a blade tip[J]. Journal of Heat Transfer, 2012, 134(6).

[5] R. S. Bunker. A review of turbine blade tip heat transfer. Turbine 2000 Symposium on heat transfer in Gas Turbine System. Cesme Turkey, 2000.

[6] J. R. Christophel, E. Couch, K. A. Thole. Measured adiabatic effectiveness and heat transfer for blowing from the tip of a turbine blade[J]. Journal of Turbomachinery, 2005, 127(2): 251-262.

[7] C. Zhou, H. Hodson, G. Lock. Thermal performance of cooled tips in a high pressure turbine
cascade[J]. Journal of Propulsion and Power, 2012, 28(5):900-911.

[8] D. Yang, X. Yu, Z. Feng. Investigation of leakage flow and heat transfer in a gas turbine blade tip with emphasis on the effect of rotation[J]. Journal of Turbomachinery, 2010, 132(4).

[9] P. P. Palafox, M. G. Oldfield, P. T. Ireland. Blade tip heat transfer and aerodynamics in a large scale turbine cascade with moving endwall[J]. Journal of Turbomachinery, 2011, 134(2).

[10] M. Rezasoltani, K. Lu, M. T. Schobeiri. A combined experimental and numerical study of the turbine blade tip film cooling effectiveness under rotation condition[J]. Journal of Turbomachinery, 2015, 137(5).

[11] Metzger D E, Bunker R S, Chyu M K. Cavity Heat Transfer on a Transverse Grooved Wall in a Narrow Flow Channel[J]. Journal of Heat Transfer, 1989, 111(1):73-79.