Impact of blade leading edge film coolant on the blade tip phantom cooling

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Abstract. Numerical simulations are carried out to predict the tip phantom cooling from blade leading edge (LE) coolant. The impact of spanwise angle of the blade LE film holes on the blade tip phantom cooling are investigated in detail under various the mass flow ratios (MFR) of coolant to mainstream from 0.3% to 1.3%. The results demonstrate that the LE film coolant could help cool the tip. These coolants mainly affect the aerothermal performance of the tip forepart and have little impact on that of the tip rear part. Increasing the coolant MFR has an obvious improvement on the tip phantom cooling level, the area-averaged cooling effectiveness could increase more than 2 times in comparing the cases with a higher MFR and a lower one. The cooling effectiveness on the LE surface first increase then decrease as a result of these coolants lift-off the LE surface with the increasing MFR. However, the LE film holes spanwise angle almost has no impact on the tip phantom cooling level at a lower MFR and even deteriorate the LE heat transfer procedure, where the cooling effectiveness decreases as the film holes spanwise angle increases.

1. Introduction
The gas turbine operating temperature is extremely high, and exceeds the limited temperature that the metal blade could normally operates. Consequently, many cooling methods has been applied to protect turbine components. However, the blade tip is difficult to cool as a result of some mainstream hot gas would across through the tip gap at a very high speed. The high-speed leakage flow would enhance the heat transfer for that the boundary layer is thin. Furthermore, the flow field structure in the gap is complex and is a three-dimensional problem even for a stationary case. Considering the above factors, the blade tip region is always exposed to a huge thermal load, which may damage the blade material. On the aerodynamic, the leakage flow would form the leakage vortex (LV) at the gap exit, causing the leakage flow loss and reducing the turbine efficiency. Denton [1] revealed that the tip leakage flow could reduce the work done by the mainstream, and could cause the leakage loss.

To better protect the tip, many open literatures have discussed aerothermal characteristics inside the tip gap. Newton and Lock [2], using oil-flow visualization method, found that the leakage flow would form a separation region on the tip near the pressure side (PS), where its heat transfer coefficient is the highest. Xiao and McCarter [3, 4] measured the flow field in the tip gap and in the passage. Their results showed that the highest total pressure loss value in the LV is much higher than that in the passage vortex. O’Dowd et al. [5] experimentally revealed that the tip gap size would greatly affect the size and strength of the tip LV. The blade in real turbine is a rotating component. Thus, it is vital to study the tip aerothermal performance under the rotating condition. Zhou [6], Srinivasan and Goldstein [7] analyzed the impact of relative casing movement on tip aerothermal performance. Zhou and Zhou [8] found that positive in-coming vortex could suppress the LV and reduce the aerodynamic
loss. Zhang et al. [9] experimentally found that both the turbulence intensity and the inlet boundary layer thickness don’t affect the tip heat transfer. Zhang and He [10], and Jiang et al. [11] demonstrated that the wall thermal boundary condition of the tip has an evident impact on the tip leakage flow structure. Arisi et al. [12] pointed out that higher exit Mach number enhances the tip thermal performance while it does not affect the LV.

What’s more, considering the high thermal loads in the tip region, cooling methods for the tip are very important and necessary. Many cooling methods has been applied on the tip to protect this region, such as tip injection, casing injection and PS near-tip injection, or their combined cooling injection. However, the application of cooling method will increase the consumption of coolant and reduce the turbine thermal efficiency. From this point of view, new cooling concept has been introduced and discussed in the 1980s in the turbine industry. Phantom cooling, firstly detailed studied by Roback and Dring [13, 14], is the coolant redistribution on the blade surfaces or the end-wall caused by the upstream coolant injection or the leakage flows. It means that local area surface could be cooled by upstream spent coolant, which is a kind of the second order and unintentional cooling effect. Although the coolant temperature increases a lot after having cooled its target area, it may have the potential to cool other regions. Thus, this kind of unintentional cooling effect could reduce coolant consumption and even there is no need to place film holes on the local surface. For the blade tip, its phantom cooling could be affected by the nozzle shroud cooling discharge, the nozzle surface film cooling, the nozzle trailing edge slot, the leakage flow between tip shoe and nozzle end-wall, and the blade LE and the PS film coolant. Although above factors are not validated by experimental or numerical method in open literatures, they may be considered in industrial design process, especially for the impacts of LE film coolant.

Phantom cooling could save local coolant usage and increase the gas turbines output power and thermal efficiency. Zhang et al. [15], using pressure sensitive paint technology, measured the impact of end-wall hole diameter on the suction side (SS) phantom cooling. Zhang et al. [16] also pointed that decreasing the LE film holes spanwise angle and employing the compound angle would improve the end-wall cooling effectiveness. Liu et al. [17] also carried out similar studies and considered the impact of coolant from the trailing edge slots. Zhang et al. [18], Du et al. [19, 20, 21], Zhang and Yuan [22], and Papa et al. [23] investigated the impact of coolant from leakage slot on phantom cooling at the SS surface near the end-wall. Their studied variables mainly include the swirling jet angle, slot jet angle and blowing ratio. Zhang and Yuan [24], and Li et al. [25] found that both the end-wall and blade SS phantom cooling could be achieved by the coolant from trailing edge slots.

Previous work only focused on the tip aerothermal performance, or the phantom cooling on the blade SS or the end-wall surface. However, no attention was paid to the impacts of coolant from blade LE film hole on the tip phantom cooling effects up till the present moment. In this study, the phantom cooling level on the tip from LE film coolant is studied. The impacts of MFR and LE film hole spanwise angle on the tip phantom cooling and aerodynamic performance are detailly analyzed. To investigate these impacts, numerical simulations are performed under five different MFR values: 0.3%, 0.5%, 0.7%, 1.0%, and 1.3% at a LE film hole spanwise angle value of 25°; and five different LE film hole spanwise angle values of: 25°, 35°, 45°, 55° and 65° at an MFR of 0.5%.

2. Numerical methods

2.1. Geometry configuration

The calculation model derived from the first stage of the GE-E3 engine turbine (Ref. [26]). The blade height is 42.65mm, and the tip gap size is about 1% of the blade height. These dimensions match those in Ref. [26]. To simplify the computational model, the impingement cooling in the LE area is not concluded, and coolant is supplied by a coolant supply plenum (see figure 1(a)). The flat tip is imposed in the present study, while the original one from Ref. [26] is the squealer tip. The three rows LE film hole (Row 1-3) have a spanwise angle ranging from 25° to 65° (see figure 1(b)) and is arranged at a constant pitch with a diameter of 0.36mm. All the holes in Row 1-3 are the cylindrical
hole (see figure 1(c)). The location of film hole row is defined according to the percentage of arc length of the airfoil profile. The Row2 is located on the stagnation line of the LE and its percentage value is 0%, while the Row1 and Row3 is located on the SS (the minus value) and the PS (the plus value), respectively. Details about film hole definitions are presented in figure 1, and table 1 shows detailed geometric information.

2.2. Numerical simulations setting
The boundary conditions come from Ref. [26]. As displayed in figure 2, Total pressure and total temperature are set to the mainstream inlet, and the mainstream outlet is set to the average static pressure. Coolant temperature and its mass flow rate are set to the coolant inlet. The cascade passage either side are set to the rotational periodic boundary condition. and the mixing plane is the stator-rotor interface. The other surfaces are set to the no-slip and adiabatic wall. The blade rotational speed is 8450 rpm. The main fluid and the coolant air are ideal gases. The main fluid’s physical properties are functions of temperature. Table 2 presents the boundary conditions in detail.
Present numerical simulations are implied by solving the commercial CFD code ANSYS CFX. This code could solve the steady Reynolds-Averaged Navier-Stokes equations and discretize the conservation equation by applying the finite control volume. The convection term is discretized by employing the high-resolution method with a second order form. The whole calculations are operated under the second order. The convergent target is that the area-averaged $\eta$ on the tip remains constant with additional iterations.

2.3. Numerical simulations setting

To find out the best turbulence model to carry out this study, a linear cascade experimentally investigated by Kwak [27] is chosen to validate the accuracy of numerical methods with a flat tip case. Figure 3 compared the difference of $h$ distribution between the measurement and predictions. The results reveal that the standard $k$-$\omega$ model have similar $h$ distribution with experimental data, which achieves the best precise predictions in the separation region near the PS. The SST $k$-$\omega$ model also could capture the separation region but underpredicts the high-$h$ region. However, the standard $k$-$\varepsilon$ model overpredict the high-$h$ region and could not catch the high-$h$ region near LE. The RNG $k$-$\varepsilon$ mis-predicts the separation region and underpredicts the high-$h$ region. It is showed that the standard $k$-$\omega$ is the most suitable model and it is employed in the presented simulations.

![Figure 3](image)

Figure 3. The $h$ distribution between predictions and measurement

Three different meshes for the calculation domains with grid numbers of 5.75 million, 8.20 million, and 10.20 million were employed to carry on the grid independence analysis. The unstructured mesh in the calculation domains were generated. The mainstream region consists of tetrahedral mesh and the near-wall regions are full of prism mesh. To ensure that there is no poor-quality mesh, all the mesh elements had been refined. To catch the high velocity gradient near-wall regions, the number of boundary layers is 15, and the first boundary layer height is 0.001mm with a growth factor of 1.1, the wall $y^+$ in all cases is about 1.0.

Figure 4 presents the $\eta$ level on the tip and its pitch-averaged $\eta$ along the axial-chord direction at three different numbers of nodes for the case with $\theta=25^\circ$ and $MFR=1.0\%$. The results show that there is almost no difference between the cooling effectiveness with 8.20 million (figure 4(a) (2)) and that with 10.20 million (figure 4(a) (3)). The pitch-averaged $\eta$ also show the same distributions among the cases with 8.20 million and 10.20 million. It could be concluded that the increasing number of nodes has almost no impact on the predictions. Considering the numerical simulations precision and the calculation cost, the mesh with 8.20 million nodes was employed in present simulations.
3. Results and discussions

3.1. The tip phantom cooling features

Figure 5 shows the LE film coolant flow path and its phantom cooling effects on the blade tip with a case of $\theta=25^\circ$ at an MFR of 0.5%. From the figure 5(a), first, the coolant from the LE film hole cool the LE surface, and form a high-$\eta$ region on local surface. Then, the coolant, mixing the mainstream, flow into the tip gap. Although the coolant’s temperature increases a lot, it still has the potential to cool the tip surface. Thus, it is observed that there are several bar-shaped high-$\eta$ regions. It is also found that the coolant flowing into tip gap mainly comes from the No. 1, No. 2 and No. 3 hole in Row2 and Row3. These coolant flow into the tip gap for that the pressure difference between the PS and the SS. These coolant from No. 1 in Row1 also form a high-$\eta$ region on the tip. The coolant is suppressed by the mainstream and become a part of leakage flow. While it is caused by the mainstream incidence angle rather than the pressure difference. Besides, From the figure 5(b), the coolant will form the LV and passage vortex and it will affect the aerodynamic characteristics in the passage.

Figure 5. The LE film injection coolant flow path and its phantom cooling effects on the blade tip
3.2. Impact of MFR

3.2.1. Thermal performance. Figure 6 shows the $\eta$ level on the tip surface for different MFR values at a $\theta$ value of $25^\circ$. The results show that there are two higher $\eta$ regions (A and B) at the tip forepart. Region A starts from the $z/C_{ax}=0$ near the SS and ends at the about $z/C_{ax}=0.25$ near the PS. This region is cooled by the coolants from the No.1 hole of Row1 and Row2 of the LE. These coolants are suppressed by the mainstream and directly flow into the tip gap, becoming a part of the leakage flow. Region B starts from the $z/C_{ax}=0.1$–0.2 near the PS and then across the tip gap and ends at the $0.35$–$0.45$ $z/C_{ax}$ near the SS. The coolant from the No. 1 hole of Row2 and Row3 cool this area, mixing with the mainstream. It is also observed that two low-$\eta$ regions (C and D) at the tip central. These regions are cooled by the coolant from the No. 2 and No. 3 hole in Row2 and Row3, respectively. The temperature of coolant rises due to it mixes with mainstream, its second order cooling effect on the tip become weak. Thus, the $\eta$ in regions C and D is lower than that in region A and B. The $\eta$ level at other areas on the tip is almost zero except region A, B, C and D. This is because almost no coolant could cover these areas in the studied cases. At a low MFR of 0.3%, the area of high-$\eta$ is small. The highest $\eta$ level is observed in Region B, and Region D almost cannot be detected. As MFR increases to 0.5% and 0.7%, all the high-$\eta$ regions expand. Regions C and D integrate and the low-$\eta$ regions shrink. At a high MFR of 1.0% and 1.3%, the area that $\eta=0$ almost disappear and the results show a high-level phantom cooling performance. At all MFR values, the high-$\eta$ regions are detected in tip forepart ($z/C_{ax}=0$–0.5) and the Region B always shows the highest $\eta$ value.

![Figure 6. The $\eta$ level on the tip under different MFR at a $\theta$ value of $25^\circ$](image)

To quantitatively compare the $\eta$ level for different MFR values, figure 7 shows the pitch-averaged $\eta$ distribution along the axial-chord direction. At lower MFR, two peak values of $\eta$ appear between the $z/C_{ax}=0.1$–0.2 and $z/C_{ax}=0.3$–0.4, which agree with the high $\eta$ regions in figure 6. While there is no obvious peak value of $\eta$ at higher MFR, and the $\eta$ level decreases along the axial-chord direction. On the tip rear part($z/C_{ax}=0.5$–1.0), the $\eta$ value is much lower than that in the forepart of tip, which means that the LE coolant mainly affects the heat transfer of the tip forepart and have little impact on the tip.
rear part. In general, the MFR has an evident impact on the η level of the tip, particularly in the aera \( z/C_{ch}=0-0.5 \). The peak value of the pitch-averaged η in the case with the MFR=0.3% is 0.1, and this value increases by 190% when MFR=1.3%. Combining the figure 6 and figure 7, with an increase of MFR, both the higher η regions (A and B) and the lower η regions (C and D) expand (integrate at higher MFR).

Figure 8 presents the η level on the LE for different MFR values. The left side of each picture is the SS and the right one is the PS. The results show that the high-η regions located on the coolant flow path at any MFR, especially on the hole exit the η could reach to 0.9. At a low MFR of 0.3%, its η level is relatively high. However, the η value on the hub region of the LE is much lower than other areas and no coolant eject on the stagnation line of the hub region. As increases to 0.5%, the η level both on the PS and the SS enhance, and the hub region also could achieve a better film coverage. However, although the high-η areas on the tip region expand, its highest η value decrease from about 0.9 to about 0.8. This is owing to that the more coolant discharge in the tip region for that the centrifugal force and the coolant jet off from the LE surface. As the MFR continues to increase, the η value in these high-η areas decreases, especially on the PS and hub region. At a high MFR of 1.3%, the high-η areas on the PS almost disappear, which means that the PS would be difficult to cool under a higher MFR. The coolant on the tip region integrate and form a larger high-η areas, but its level is lower than that in the cases with a low MFR. With the increasing MFR, the η level on the LE first increases and then decrease, achieving the highest η level with the MFR of 0.5%.

![Figure 7. Pitch-averaged η distribution along the axial chord direction on the tip at a \( \theta \) value of 25°](image)

![Figure 8. The η level on the LE surface for different MFR at a \( \theta \) value of 25°](image)

To better understand how the MFR influence the phantom cooling level on the tip and the film cooling performance on the LE, figure 9 shows the coolant flow structure and the non-dimensional temperature distributions on the cross-section of Row3 near the tip for different MFR values. The non-dimensional temperature distributions could reveal the coolant distribution on the LE and in the tip gap. The left side of each picture is the PS, and the right side the SS. The top side is the tip gap. For a low MFR of 0.3%, the high-ε region could be detected nearby the hole exit and the ε level in the tip gap is very low. As MFR increases to 0.7%, the high ε region on the LE expands and a high-ε region could be found in the tip gap near the SS. As MFR continue to increase, some coolant lifts off the LE
surface and become a part of the cross flow. Therefore, the $\eta$-level on the LE decrease, which agree with the results in the figure 8. However, more coolant flow into the tip gap and the $\varepsilon$ level in the tip gap is relatively higher, it means that increasing the MFR would enhance the phantom cooling performance on the tip.

![Figure 9](image)

**Figure 9.** The coolant flow structure and the non-dimensional temperature distributions on the cross-section of row3 near the tip.

### 3.2.2. Aerodynamic performance

Figure 10 displays the normalized tip leakage mass flow rate distribution for different MFR. The minus tip leakage mass flow rate values, existing at about $z/C_{az}$ = 0–0.13, are caused by the mainstream incidence angle. It means that some mainstream directly flows into the tip gap, mixing with the coolant. With an increase of MFR, the tip leakage mass flow rate increase at $z/C_{az}$ = 0.25–0.6. While the MFR has little impact on it at other parts of tip.

![Figure 10](image)

**Figure 10.** The normalized tip leakage mass flow rate distribution along the chordwise direction at a $\theta$ value of 25°

Figure 11 shows the distribution of the dimensionless specific entropy increment at several cut-planes of $z/C_{az}$ = 0.2, 0.4, 0.6 and 0.8. The LV begins to appear at the $z/C_{az}$ = 0.2 and develops along the streamwise direction. The upper passage vortex also develops along the streamwise and its size is smaller than that the LV. With increasing MFR, the size of both the LV and the upper passage vortex almost remains constant, but its values of entropy increment increases, especially at 0.6 and 0.8 $z/C_{az}$ cut-planes.

![Figure 11](image)
**Figure 11.** The distribution of the dimensionless specific entropy increment at different cut-planes at a \( \theta \) value of 25°

Figure 12 shows the contours of the \( C_p \) on the \( z/C_{az}=1.1 \) cut-plane for different MFR at a \( \theta \) value of 25°. The upper boundary of each figure is casing, and the lower boundary of each figure is hub. There are three obvious higher \( C_p \) regions corresponding with loss cores, including the LV, the upper passage vortex and the lower passage vortex. Besides, there are boundary layer losses near the hub and shroud regions and profile losses in the middle part of this picture. With the increase of MFR, the \( C_p \) values in the upper vortex decrease, while there is no much difference in other loss cores. To quantitatively compare the \( C_p \) at different MFR values, figure 13 presents the pitch-averaged \( C_p \) distribution in the spanwise direction. The results show that the MFR just have little impact on the \( C_p \) at 0~0.3 and 0.60~0.96 relative blade height. At the 0.90~0.96 and the 0.60~0.80 relative blade height, corresponding to the LV and upper vortex, the pitch-averaged \( C_p \) decrease with increasing MFR. At the 0~0.3 relative blade height, corresponding to the lower passage vortex, the pitch-averaged \( C_p \) also decrease with increasing MFR.

![Figure 12](image-url)
3.3. Impact of $\theta$

*Thermal performance.* Figure 14 shows the $\eta$ level on the tip for different $\theta$ at a constant MFR of 0.5%. At a smaller $\theta$ value, the radial component of the coolant velocity is relatively large, and under the oppression of the leakage flow, the coolant would cover the tip surface better. With an increase of $\theta$, both the area of all high-$\eta$ regions and its highest $\eta$ value decrease. It is obviously found that the Region A nearly vanished in case of $\theta$ value of 65° and the Region D disappears in cases of $\theta$ values, 55° and 65°. This is because the radial component of coolant velocity decreases, and the axial component increases with an increase $\theta$, then coolant mainly jets into the mainstream hot gas. This will weaken the second order cooling effect on the tip due to mixing with the mainstream hot gas. Figure 15 shows the pitch-averaged $\eta$ along the axial chord direction for different $\theta$ values. There are two peak values of $\eta$, located at about $z/C_{az}=0.16$ and $z/C_{az}=0.4$. It is observed that the $\theta$ has an evident impact in these areas $z/C_{az}=0.1-0.2$ and $z/C_{az}=0.3-0.4$, which are corresponding to the high $\eta$ values in Regions A, B and C in figure 13. With an increasing $\theta$, the pitch-averaged $\eta$ decrease in above three regions, while it almost remains constant in other regions.

**Figure 13.** The pitch-averaged $C_{p_t}$ distribution along the spanwise direction on the $z/C_{az}=1.1$ cut-plane at a $\theta$ value of 25°

**Figure 14.** The $\eta$ level on the tip in different $\theta$ at an MFR value of 0.5%
Figure 15. The pitch-averaged $\eta$ distribution along the axial-chord direction on the tip at an MFR value of 0.5%.

Figure 16 shows the $\eta$ level on the LE for different $\theta$ values at an MFR value of 0.5%. The features of high-$\eta$ areas in Figure 16(a) are clearly shown and analyzed in Figure 8. The results show that with the increasing $\theta$, both the $\eta$ value in the high-$\eta$ regions and the its area decrease. And on the PS, the coolant flow path almost disappears at high $\theta$ values. This is because that the radial component of the coolant velocity for the case with a small $\theta$ is large, and the coolant would flow along the LE surface under the mainstream suppression. As the $\theta$ become larger, the radial component of coolant velocity decreases, and the axial component increases. Thus, the coolant mainly jets into the mainstream hot gas rather than cover the LE surface. Furthermore, there is a low-$\eta$ region between neighboring film holes in the same row, especially for the Row2 and Row3, for that there is no coolant coverage in these regions.

Figure 16. The $\eta$ level on the LE in different $\theta$ at an MFR value of 0.5%.

Figure 17 presents the coolant flow structure and the non-dimensional temperature distributions on the cross-section of Row3 near the tip for different $\theta$ values. For a low $\theta$ value of 25°, the high-$\varepsilon$ region could be detected nearby the hole exit and the $\varepsilon$ level in the tip gap is very low. As the $\theta$ increase, the $\varepsilon$ level near the hole exit still remains high. However, the component of coolant velocity that towards to the tip decreases due to the increasing $\theta$. Therefore, the $\eta$-level on the LE decrease, which agree with the results in the figure 16. Meanwhile, the coolant that flows into the tip gap become less, which leads to the phantom cooling performance on the tip become weak. In general, increasing $\theta$ would deteriorate the heat transfer process both on the LE and the tip.
Figure 17. The coolant flow structure and the non-dimensional temperature distributions on the cross-section of row3 near the tip

3.3.1. Aerodynamic performance. Figure 18 displays the normalized tip leakage mass flow rate for different $\theta$ at an MFR value of 0.5%. With the increasing $\theta$, the tip leakage mass flow rate almost remains constant, which means that the $\theta$ would not affect the leakage flow in the case with an MFR value of 0.5%.

Figure 18. The normalized tip leakage mass flow rate distribution along the chordwise direction at an MFR value of 0.5%

Figure 19 shows the distribution of the dimensionless specific entropy increment at several cut-planes of $z/C_{ax}=0.2$, 0.4, 0.6 and 0.8. At any cut-plane, the size of both the LV and the upper passage vortex, and the high values of entropy increment are almost the same with increasing $\theta$. This is because that the entropy increment is mainly affected by the LV, which caused by the leakage flow interacts with the across secondary. However, the $\theta$ would not affect the leakage flow as displayed in figure 18. Thus, the dimensionless specific entropy increase distribution at any cut-plane remains the same with increasing $\theta$.

Figure 19. The distribution of the dimensionless specific entropy increment at different cut-planes at an MFR value of 0.5%

Figure 20 and Figure 21 show the contours of $C_{p_s}$ at the $z/C_{ax}=1.1$ cut-plane and its pitch-averaged values along the spanwise direction for different $\theta$ with an MFR value of 0.5%, respectively. The loss cores and their corresponding vortex are shown and analysis results are in figure 12. There is no obvious difference in the contours the $C_{p_s}$ for different $\theta$ and the figure 20 reveals that $\theta$ has no impact on the $C_{p_s}$ distributions.
3.4. Overall performance

Figure 22 shows the area-averaged $\eta$ on the tip for various $\theta$ and MFR values. The area-averaged $\eta$ increases proportionally with the increasing MFR under all $\theta$ values. For a small $\theta$ value of $25^\circ$, the area-averaged $\eta$ in the case of MFR=1.3% increase by 220% in comparing with that in the case of MFR=0.3%, while for the cases of $\theta=65^\circ$, the rate of increase is 75%. It means that MFR could obviously enhance the phantom cooling performance under a smaller $\theta$. Furthermore, the impact of $\theta$ on the area-averaged $\eta$ depends on the value of MFR. For a low MFR of 0.3%, the $\theta$ almost has no impact on the phantom cooling performance. However, as MFR increase to 1.3%, the area-averaged $\eta$ in the case of $\theta=25^\circ$ is about 2 times as high as that in the case of $\theta=65^\circ$. It indicates that the impact of $\theta$ on the phantom cooling performance of the tip is more obvious at higher MFR.

Figure 23 shows the area-averaged $\eta$ on the LE for various $\theta$ and MFR values. The results show that the area-averaged $\eta$ on the LE first increase then decrease with increasing MFR. The peak value of
the area-averaged $\eta$ appear at $MFR=0.5\%$ for all cases, which means that the optimal $MFR$ for the LE is 0.5\%. When $\theta$ increases, the cooling performance on the LE become worse under any $MFR$ value. As $MFR$ increases from 0.3\% to 1.3\%, the difference between the area-averaged $\eta$ of the cases of $\theta=25^\circ$ and $\theta=65^\circ$ increases from 50\% to 172\%. It shows that the $\theta$ has a more obvious impact on the film cooling performance on the LE at higher $MFR$.

Figure 24 shows the area-averaged stage efficacy on the $z/C_{ax}=1.1$ cut-plane for various $\theta$ and $MFR$ values. The results indicate that the area-averaged $\Phi$ increases proportionally as $MFR$ increases at any $\theta$ values. As $MFR$ increases from lower to a higher one (0.3\% to 1.3\%), the area-averaged $\Phi$ increases by about 2.5\%. But, the area-averaged $\Phi$ remains constant with the increase of $\theta$, which means that $\theta$ has no impact on the stage efficacy.

4. Conclusions

In present study, numerical simulations are implied to investigate the blade tip phantom cooling performance due to the blade LE film discharge. The features of phantom cooling on the tip are demonstrated. The impacts of coolant $MFR$ and LE film hole spanwise angle on the tip phantom cooling are analysed. Turbulence models are validated by contrasting with the experimental measurements in an open literature. Present results reveal that the standard $k-\omega$ model is closest to the experimental measurements and it is employed in present studies. A grid independence analysis demonstrates that 8.20 million mesh nodes for the calculation domain could give the reasonable results. The main results can be listed as:

These coolant from blade LE could produce phantom cooling effect on the tip and they mainly affect the cooling performance on the tip forepart and have little impact on the tip rear part. Increasing the $MFR$ could not only enhance the tip phantom cooling, but also influence the $\eta$ distribution on the tip. The peak values of pitch-averaged $\eta$ mainly exist at the $z/C_{ax}=0.1\sim0.2$ and $z/C_{ax}=0.3\sim0.4$ on the tip at low $MFR$. While there is no obvious peak value of pitch-averaged $\eta$ at high $MFR$ cases. The impact of $\theta$ on the phantom cooling of the tip is more obvious at higher $MFR$. For a low $MFR$ of 0.3\%, the $\theta$ almost has no impact on the phantom cooling performance. However, as $MFR$ increase to 1.3\%, the area-averaged $\eta$ in the case of $\theta=25^\circ$ is about 2 times as high as that in the case of $\theta=65^\circ$.

Both the $MFR$ and $\theta$ have a critical impact on cooling performance of the LE. With the increasing $MFR$, the film coverage on the LE first improve then become worse. The peak value of the area-averaged $\eta$ on the LE appear at $MFR=0.5\%$ for all cases. Increasing the $\theta$ would worsen the film cooling performance on the LE. This is because that as the $\theta$ become larger, the radial component of coolant velocity decreases, and its axial component increases. And the coolant mainly jets into the mainstream hot gas rather than cover the LE surface. At different $MFR$ values, the area-averaged $\eta$ on the LE of the cases of $\theta=25^\circ$ and $\theta=65^\circ$ decreases from 50\% to 172\%.

The impact of $MFR$ on the aerodynamic performance is greater than that of the LE film hole spanwise angle. The tip leakage flow, corresponding leakage flow loss and the $C_p$ increase with increasing $MFR$, while those parameters almost remain constant with increasing $\theta$. As $MFR$ increases,
the area-averaged $\Phi$ increases by about 2.5% at most. The area-averaged $\Phi$ also remains constant with the increase of $\theta$, which means that $\theta$ has no impact on the stage efficacy.

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**Nomenclature**

- $cp$: specific heat at constant pressure, [J/(kg·K)]
- $Cp_s$: pressure loss coefficient, $[\frac{(p_{s,\text{ref}}-p_{s,\text{local}})}{(p_{s,\text{ref}}-p_{\text{local}}})]$  
- $D$: diameter of film hole, [mm]
- $h$: heat transfer coefficient, [W/(m²·K)]  
- $L$: length of film hole, [mm]
- LE: leading edge
- LV: leakage vortex
- $m$: mainstream, mass flow rate, [kg/s]
- $MFR$: mass flow ratio, [mₖ/mₘ]
- $N$: number of film hole
- $P$: hole pitch in each row, [mm]
- PS: pressure side
- $p$: pressure, [kPa]
- $p_{s,\text{ref}}$: reference total pressure, $[(m_{\text{c}}\cdot P_{s,\text{c}}+m_{\text{c}}\cdot P_{s,\text{c}})/(m_{\text{c}}+m_{\text{w}})]$
- $R$: universal gas constant, [J/(mol·K)]
- $s$: entropy, $[\Delta s = cp \ln(T/T_1)-R \ln(P/P_1)]$
- SS: suction side
- $T$: temperature, [K]
- $Tu$: turbulence intensity, [%]
- $V$: velocity, [m/s]
- $y^+$: non-dimensional distance, $[Y\mu/\nu]$
- $Y$: transverse coordinates from wall, [mm]
- $Z$: axial chord direction, [mm]

**Greek symbols**

- $\alpha$: hole angle in the axial direction, [°]
- $\gamma$: specific heat ratio
- $\varepsilon$: non-dimensional temperature, $[(T_{aw}-T)/(T_{aw}-T_c)]$
- $\zeta$: dimensionless specific entropy increase, $[T_{2}-As/0.5V_{z}^{2}]$
- $\eta$: film cooling effectiveness, $[(T_{aw}-T_0)/(T_{aw}-T_3)]$
- $\theta$: LE film hole angle in the spanwise direction, [°]
- $\mu$: dynamic viscosity, [Pa·s]
- $\nu$: kinematic viscosity, [m²/K]
- $\tau$: shear stress
- $\Phi$: stage efficiency, $[(T_{s,0}T_{s,0})/(T_{s,0}P_{s,0})(P_{s,0}P_{s,0})^{((\gamma+1)/\gamma)}]]$
- $\omega$: speed of rotation, [rpm]

**Subscripts**

- $aw$: adiabatic wall
- $az$: axial chord direction in Z coordinate, [mm]
- $c$: coolant
- $is$: isentropic
- ref: reference
- $s$: total
- $w$: walls
0 stage inlet
1 blade inlet
2 stage outlet

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