Numerical study on heat transfer characteristics of swirling flow on dimpled surfaces with effusion holes at turbine blade leading edge

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Abstract. In this paper, we conducted a numerical study to investigate the effect of the offset of the jet holes on heat transfer of swirling flow in a concave target chamber with various dimple structures and effusion holes at the turbine blade leading edge. The distance of the jet holes off the centerline $e/d$ varies from 0 to 2.0. Four types of dimple structure, including spherical dimples (SDs) and oval-trench dimples (OTDs) in the inline and staggered arrangement, are considered. The heat transfer performance of the different leading-edge, impingement-effusion cooling structures is evaluated and compared at a Reynolds number of 30,000 based on the jet hole diameter. Results show that the offset of the jet holes provides 15% higher overall heat transfer performance and more uniform heat transfer of the target surface within the $e/d$ range of 0-2.0. The introduction of the dimple structures on the target surface slightly decreases the overall averaged Nusselt number but enhance the heat transfer quantity due to the clear increase of heat transfer areas. Under the same $e/d$, the OTD structure, especially with the staggered arrangement, is superior to SD structure.

1. Introduction
In order to achieve a high thrust-to-weight ratio and strong efficiency of aero engines, it is necessary to continuously increase the inlet temperature of the turbine, and excessive temperature will lead to ablation of the turbine blades. One of the main methods for creating high-temperature turbines is intensive cooling of the nozzle and working blades [1]. The leading edge of the turbine blade is directly subjected to the impact of the high-temperature mainstream gas and suffers an extremely high external heat load. The impingement-effusion composite cooling structure has been widely utilized in the gas turbine blade. Cooling air flowing through the impingement hole forms a high-speed jet and impacts the leading edge of the inner wall. Due to the destruction of the thermal boundary layer at the impingement region, jet impingement cooling has a significant improvement of the local heat transfer coefficient. In addition, as for internal cooling, vortex cooling is quite promising for its high cooling efficiency, low flow losses and even heat distribution [2-5]. Dimple technologies as a surface vortex generator occupy an important place among vortex technologies in power engineering. Many researchers have explored the effect of jet impingement over dimpled surfaces with different configurations and flow parameters on heat transfer as compared to that over a flat plate [6-10]. It has been found that the presence of spherical
Dimples on the target plate can induce higher energetic vortices, and this leads to better heat transfer performance compared to a flat plate.

As reported in a series of works of Isaev et al. [11-13], such oval-trench dimples can induce the phenomenon of abnormal intensification of turbulent separated flow and heat transfer in the narrow channel [11-12] and tube [13] due to the forming a large pressure drop between the closely spaced zones of stagnation and low pressure at a place where a spiral vortex is generated at the entrance portion of the semi-spherical segment of the dimple. As a result, a channel with oval-trench dimples-spiral vertex surface generators provides higher thermal and thermal-hydraulic efficiencies compared to channels with spherical dimples.

The objective of this study is to numerically investigate the effect of the offset of jet holes and dimpled target surface on heat transfer of swirling flow in an impingement-effusion cooling structure for a blade leading edge. An impingement chamber model with offset jets was set up, as described in [2]. Four cases with different dimple structures were studied. The Reynolds number was fixed at 30,000 based on the jet hole diameter, and the detailed internal local Nusselt number distributions and overall average Nusselt number of the leading edge were analysed.

2. Numerical methodology

2.1 Physical model

Fig. 1 shows the conventional, three-dimensional model of the impingement/effusion cooling system for the leading edge of a turbine blade, which was constructed based on the reference of [2]. The computational domain includes a concave shape cavity with film holes, an impinging plate, and a rectangular supply channel. Cooling air impinges on the target surface from a center array of jet holes and is then extracted out through the effusion holes. Thus, the crossflow effect is not considered in this paper. As shown in Fig.2, the geometric dimensions of these models are indicated. The target surface consists of a partially circular concave surface connected by a flat surface. The length and curvature of the target surface are 360 mm and 60 mm respectively. The diameter of jet holes d is 15 mm, and six impingement holes are evenly arranged horizontally with equal spacing of 60 mm along the target surface. The ratio of jet-to-target spacing to jet holes diameter is 4:1. The film holes diameter of D is 7.5 mm and the holes are arranged in the middle zones of the impingement cavity and its two side walls, showing the staggered arrangement relative to the jet holes (Fig. 2a). In addition, the outlet extension 6D in the length of film holes is set up to eliminate the outflow effect.

Figure 1. Schematic views of impingement-film cooling for leading edge of a turbine blade
Fig. 2b presents the configurations of the swirl flow-film cooling structure. In the normal impingement-film cooling (Fig. 2a), the jet hole is located directly opposite the centerline of the target surface, while, to induce the swirl flow, the jet holes are arranged to offset the centerline along the y-direction. The offset distance $e$ ranges from 0 to 30mm. Four different jet offset ratios $e/d$ (0, 0.5, 1.0, 2.0) are examined in this paper.

![Figure 2. Schematic views of the leading edge model with effusion holes: (a) normal jet; (b) swirling flow (all dimensions in mm) ](image)

To obtain a better jet impingement cooling performance, various types of dimples, namely spherical dimples (SDs) and oval-trench dimples (OTDs), are introduced into the impingement chamber of the leading edge model, where spherical dimples are arranged on the target surface at an axial pitch of 30 mm for in-line arrangement (Fig. 3a) or 15mm for staggered arrangement (Fig. 3b). Similarly, the arrangements of oval-trench dimples are shown in Fig. 3c and Fig. 3d in the same way. The oval-trench dimples are all arranged at an angle of 45° to the axis of the target surface. Fig. 3e shows the schematic diagrams of the dimple structure. The depth and project diameter of spherical dimples are 3 mm and 15 mm, relatively, and the corresponding ratio of the depth to diameter are 0.2. The rounding-off of the radius of the dimple edge is taken as equal to 0.375mm. The OTD consists of two spherical dimple halves connected by a cylinder insert. The spot area of the oval dimple is kept equal to the spherical dimple with a length $l_k$ of 67.5 mm and a width $b$ of 6.435 mm.

2.2 Mesh procedure

Fig. 4 shows the global and local mesh structure of the swirl chamber with effusion holes generated by the commercial software $STAR-CCM+12.06$. The mesh consists of unstructured polyhedral cells in the core region and prism layers in the near-wall regions to enhance the accuracy of the calculation in these regions. There are 15 prism layers near the wall, and the mesh size is increased at a constant ratio of 1.4. The mesh in the boundary layer regions of the impingement holes, effusion holes and main swirl
chamber is refined to ensure the $y^+$ value of the first layer is less than 1. Moreover, due to the dramatic variations of flow parameters near the dimple edge, the grid is also densified around the dimples.

Figure 3. Configuration of the concave target surface with SDs (a, b) and OTDs (c, d) in inline (a, c) and staggered (b, d) arrangements (all dimensions in mm)
2.3 Boundary conditions and solution procedure

Numerical calculations on the turbulent heat transfer of swirling flow in impingement/effusion cooling for the leading edge of the gas turbine blade are performed in the ANSYS FLUENT. Air ideal gas is adopted as the working fluid, and the temperature of the inlet coolant air is 302K. At the inlet, the Reynolds number $Re$ based on the jet hole diameter is fixed at 30,000 and the associated coolant mass flow rate is about $3.93 \times 10^{-2} \, \text{kg.s}^{-1}$. The physical properties of the air, including the density and the thermal conductivity, are obtained using the linear interpolation method. Sutherland laws are used to revise the dynamic viscosity of the air. In all the cases, the inlet flows turbulent intensity is 5%. The outlet of the effusion holes is defined as a pressure outlet with the atmospheric pressure. To eliminate the effect of backflow, the effusion holes exit section is extended to some extent. A constant temperature of 332 K is applied on the target surface of the swirl chamber. The other surfaces are all adiabatic and all solid walls are set as no-slip boundary conditions.

The pressure-based coupled solver and the SIMPLEC algorithm is chosen for calculations, and the second-order upwind scheme is used in the spatial discretization. The minimum convergence criterion for the continuity equation, the velocity and the turbulence is $10^{-4}$, and that for the energy equation is $10^{-7}$.

![Figure 4. Schematic review of polyhedral mesh structures of leading-edge model](image)

![Figure 5. Comparison of experimentally measured and numerically predicted area-averaged Nusselt number for different turbulence models](image)
2.4 Turbulence model selection

The calculating accuracy is highly dependent on turbulence model selection, thus, it is quite important to choose a proper turbulence model for computations of complex turbulent flows with vortex and separation. Several two-equation turbulence closure models, such as standard $k$-$\epsilon$ with Kato-Launder correction (SKE-KL), RNG $k$-$\epsilon$, Realizable $k$-$\epsilon$, SST $k$-$\omega$ are employed for numerical validation. The predicted area-averaged Nusselt numbers with the variation of the Re number are compared to experimentally measured values of Rao et al. [2] in Fig. 5. We find that the turbulence models including the standard $k$-$\epsilon$ with Kato-Launder correction, RNG $k$-$\epsilon$ and Realizable $k$-$\epsilon$ significantly underestimate the area-averaged Nusselt number on the target surface at the high Re number. Compared with the other three turbulence models, the SST $k$-$\omega$ turbulence model provides the best agreement with the experimental data. The maximum deviation to the experimental data with a Re range of 20,000-50,000 is within 15%. Therefore, the shear stress transport (SST) $k$-$\omega$ turbulence model is used in this study.

2.5 Mesh independence check

In order to evaluate the mesh number on the computed values, the mesh independence check has been carried out. Four cases with the element numbers, varying from 3.40 million to 9.49 million, were created for the impingement/effusion cooling without dimples at the Reynolds number of 30,000. The influence of meshes on the computation results is assessed using both the area-averaged Nusselt number and pressure loss. It can be seen from Fig. 6 that, when the mesh cells number reaches 5.45 million, the further increase of mesh density has little effect on the performance parameters. The deviations of area-averaged Nusselt number and pressure drop between grid schemes of 5.45 and 7.84 million are only 1.40% and 0.25%, respectively. Therefore, the grid scheme of 5.45 million has met the accuracy requirements of the numerical calculation.

![Figure 6](image)

**Figure 6.** Variations of area-averaged Nusselt number and pressure drop four different element numbers

3. Results and discussion

Fig. 7 shows the local Nusselt number distributions on the target surface under various offset distances of the jet holes. As expected for all the cases, very high $Nu$ values appear near the stagnation points at the center of the target surface, whereas relatively low heat transfer is observed around the effusion holes. For the normal jets ($e/d=0$), the Nusselt number gradually decreases along the circumferential direction due to the development of the wall jet flow in all directions. Between the middle region of the adjacent jet stagnation points, low heat transfer stripe regions appear along the circumferential direction due to the interaction between adjacent wall jets.

When the jet holes are located offset from the centerline, the high heat transfer regions are also shifted toward the circumferential direction and the local Nu distributions are asymmetric relative to the axial
direction. This is because the offset of the jet holes leads to the development of a wall jet mainly along the negative direction of the Y-axis. As the $e/d$ increases from 0 to 2.0, the region with a high Nu gradually decreases and shrinks to a small crescent region at $e/d=2.0$. We note that the Nusselt number near the stagnation points does not decrease due to the effect of the offset of the jet holes and the region with relatively high heat transfer obviously increases. Thus, the heat transfer of the target surface becomes more uniform. In addition, high Nu values are also observed around the edges of effusion holes deviation from the stagnation point, which is caused by wall jet flow acceleration, leading to a thinner boundary layer around the effusion holes.

![Image](image.png)

**Figure 7.** Nusselt number distributions on target surfaces of leading-edge for different $e/d$

Fig. 8 gives the distributions of the Nusselt number on a dimpled target surface with various dimple structures. Compared to the target surface without dimples (Fig. 8a), the installation of the dimples almost does not changes the distribution pattern of the Nusselt number near the stagnation zones, while heat transfer is enhanced between the adjacent impingement stagnation zones. These are usually occupied by a low heat transfer stripe for the cases with a smooth target surface, due to the existence of the dimples around it, which can increase the intensity of the flow disturbance. The peak value near the impingement stagnation point at the center region of the dimple is lower than that of the smooth target surface, which is consistent with the experimental results of [6] in the case of a higher ratio of jet-to-jet spacing to jet diameter. However, high heat transfer is observed near the edge of the dimple due to the thinner boundary layers when the wall jet flows leaving the dimple edge. For the case with OTDs, as shown in Fig. 8(c) and (d), in the inline and staggered configurations, staggered configuration gives better heat transfer performance because of the enlargement of the high heat transfer regions. In addition, it should be noted that a relatively low Nu is observed at the semi-spherical segments of the OTDs near the stagnation points that is occupied by the separated flow zone which results in a reduction of heat transfer.

For a quantitative comparison, Fig. 9 shows the spanwise averaged Nusselt number distribution along the axial direction for a case with different dimple configurations under the normal jet and swirling flow. Under the normal jet (Fig. 9a), It can be seen that the $Nussp$ quantitative distributions are similar to each
other. The case with staggered arrangement OTDs has the highest peak value near the stagnation points, while much lower \( \text{Nu}_{\text{sp}} \) are observed in the case with an in-line arrangement OTDs at this region. When the jet holes are off the center with \( e/d = 2.0 \) (Fig. 9b), the \( \text{Nu}_{\text{sp}} \) distribution shows a little change, and the maximum value of the \( \text{Nu}_{\text{sp}} \) is higher for the case with SDs. In this situation, the case with staggered arrangement OTDs, in turn, provides the lowest peak value near the stagnation points.

Figure 8. Nusselt number distributions on target surfaces with SDs (a, b) and OTDs (c, d) in in-line (a, c) and staggered (b, d) arrangements of leading edge under normal impingement (\( e/d = 0 \))

Fig. 10 (a) present the variations of relative overall averaged Nusselt numbers with \( e/d \), where \( \text{Nu}_0 \) are the overall averaged Nusselt number of the case with normal impingement jet and smooth target surface, and are referred to as the baseline. It can be seen that the overall heat transfer performance of the target surface for each configuration increases with an increase of \( e/d \), indicating that swirling flow is beneficial for leading-edge cooling. Among the five configurations of the target surface, the introduction of an inline arrangement of OTDs on the target surface has the lowest averaged heat transfer coefficient under the condition of normal impingement and swirl flow with a small offset of the jet hole (\( e/d \leq 1.0 \)). When \( e/d = 2.0 \), the heat transfer performance for the case with staggered OTDs is slightly lower than that of the case with inline OTDs. In comparison, the values of the case with SDs in the inline and staggered arrangement are almost the same and both are higher than those in the case with OTDs within an \( e/d \) range of 0-2.0. In addition, it should be noted that the overall heat transfer level of the dimpled target wall is lower than that of the smooth target surface with the \( e/d \) range of 0-2.0, demonstrating that the introduction of the dimples (SDs and OTDs) deteriorate the overall heat transfer.
The introduction of turbulators on the target surface can not only induce additional turbulence vortices but also increases the effective heat transfer areas. As pointed out by Chang et al. [7, 8] and Rao et al. [14], the additional heat transfer areas provided by the dimples or pin fin surface need to be considered in consideration of the capability for convective heat flux transferred by the roughened target surface. In order to highlight the relative enhancement of heat transfer quantity over the dimpled wall from the smooth-walled reference condition, Chang et al. proposed a new parameter, namely index factor $\phi/\phi_0$, which can be expressed as $(\overline{Nu}/\overline{Nu}_0) \times (A/A_0)$. In this paper, the area ratio between the dimpled and the smooth target surfaces $A/A_0$ is approximately 1.04 and 1.13 for the spherical dimpled and oval-trench dimpled surfaces, respectively. Fig. 10 (b) presents the variations of index factor $\phi/\phi_0$ with $e/d$, where $\phi_0$, $\phi_0$ are the heat transfer quantity and friction factor of the case with normal impingement jet and smooth target surface, respectively, and are referred to as the baseline. As can be seen, the introduction of the OTDs on the target surface has the highest value of index factor due to the significant increment of the heat transfer areas. The maximum value of the index factor reaches 1.26 for the case with OTDs at a larger $e/d$ of 2.0. On the contrary, when comparing the index factor, the dimpled target surface configuration has a higher value than that of the smooth target surface, showing that the introduction of the dimples on the target surface is helpful for taking more heat flux away from the target surface.
Figure 10. Variations of relative overall averaged Nusselt number (a) and index factor (b) with $e/d$

4. Conclusions
In this paper, a three-dimensional steady numerical simulation was conducted to investigate the heat transfer of swirling flow in a concave target chamber with effusion holes at the turbine blade leading edge. Four different configurations of dimple structures, i.e. SDs and OTDs in in-line and staggered arrangements, respectively, are introduced on the target wall. The local and overall averaged heat transfer characteristics on the dimpled target surface under different offset distances of jet holes $e/d$ are investigated at Re of 30,000. It is shown that, for the smooth target surface, with an increase of the $e/d$, the heat transfer of the target surface increases, and the highest Nusselt number is obtained by $e/d=2.0$ with a 15% improvement compared to the baseline under the normal jet. The introduction of dimple structures on the target surface produces a slight decrease in the overall averaged Nusselt number with the $e/d$ ranging from 0 to 2.0 but obviously enhance the heat transfer quantity from the target surface by 24% compared to the baseline under the normal jet due to the significant increment of the heat transfer area. Under a small offset of the jet holes ($e/d \leq 1$), the case with staggered arrangement OTDs has the highest index factor.

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