STUDY ON FLOW AND HEAT TRANSFER PERFORMANCE OF SINGLE JET IMPINGEMENT COOLING THROUGH VARIABLE-DIAMETER HOLE

by

Lei XI, Qicheng RUAN, Yuan GAO, Jianmin GAO, Liang XU^{*}, and Yunlong LI

School of Mechanical Engineering, Xi'an Jiaotong University, Xi'an, Shaanxi, China

Original scientific paper https://doi.org/10.2298/TSCI240109116X

In this work, the heat transfer and flow characteristics of single jet impinging cooling with three types of hole configurations, namely converging hole, straight hole, and expanded hole, were compared and analyzed numerically. The influence laws of Reynolds number, outlet-to-inlet diameter ratio, and impinging height ratio on the heat transfer, flow, and comprehensive thermal performance of the converging-hole impinging cooling were intensively investigated. Finally, the empirical correlations were fitted and the sensitivity of performance variables to influencing parameters were analyzed for the converging-hole impinging cooling. The results show that the converging hole exhibits the superlative heat transfer performance but the poorest flow performance. The expanded hole exhibits the poorest heat transfer performance but the superlative flow performance. As Re > 24,000, the comprehensive thermal performance of the three types of hole configurations is similar. For the converging-hole impinging cooling, when Reynolds number increases from 6000-30000 under various structural parameters, the average Nusselt number increases by approximately 1.62-2.65 times, and the comprehensive thermal coefficient increases by approximately 1.58-2.45 times. As the outlet-to-inlet diameter ratio increases from 0.5-0.9 under various Reynolds number, the pressure loss coefficient of the converging-hole impinging cooling decreases by approximately 91.47-92.95%, and the corresponding average Nusselt number decreases by 43.61-60.07%. When the impinging height ratio is 2.0, the converging-hole impinging cooling exhibits lower pressure loss coefficients. The parameter sensitivity analysis shows that the average Nusselt number of converging-hole impinging cooling is sensitive to changes in the outlet-to-inlet diameter ratio and Reynolds number, but insensitive to changes in the impinging height ratio. The pressure loss coefficient is highly sensitive to changes in the outlet-to-inlet diameter ratio, but insensitive to changes in the Reynolds number and impinging height ratio. The comprehensive thermal coefficient is highly sensitive to changes in the Reynolds number, but insensitive to changes in the outlet-to-inlet diameter ratio and impinging height ratio.

Key words: *impinging cooling, single jet, variable-diameter hole, flow performance, heat transfer performance*

Introduction

This poses a great challenge to the safe and reliable operation of gas turbines, hence there is an urgent need to study efficient turbine blade cooling technologies. Impinging cooling is one of the most efficient approaches to enhance the local cooling effectiveness of turbine

^{*}Corresponding author, e-mail: xuliang@mail.xjtu.edu.cn

blade inner walls. To achieve impinging cooling, essential impinging baffles with an array of small holes are generally arranged inside the turbine blades [1]. Therefore, impinging cooling is typically applied in areas of turbine blades with high thermal loads and ample space, such as the turbine blade mid-chord region or leading-edge region [2].

In turbine blades, traditional jet impinging holes are mostly equal-diameter circular holes [3]. Currently, a large number of literature reports have focused on numerical and experimental investigations regarding the cooling characteristics and capability of jet impinging cooling with equal-diameter circular holes [4]. These studies mainly analyze the effects of arrangement methods, operating conditions, and structural parameters on the cooling characteristics and capability of impinging cooling, and conduct optimization designs [5]. The arrangement methods mainly include perpendicular jet holes to the target wall [6], inclined jet holes to the target wall [7], single hole arrangement [8], array holes arrangement [9], and other forms. The operating conditions and structural parameters mainly include Reynolds number, inlet temperature, wall heat flux [10], impinging height, hole diameter, hole spacing, hole inclination angle, *etc.* [11]. Additionally, some researchers have inserted spiral strips into circular jet holes [12] or processed the inner wall surface of circular jet holes into a helical shape [13], thereby forming swirling jets to further improve the cooling capability of jet impinging.

The aforementioned research indicates that raising Reynolds number or changing jet hole structure to increase the jet impinging velocity on the target wall is one of the important factors for enhancing the heat transfer effectiveness of impinging cooling [14, 15]. Changing the jet hole outlet diameter also affects the velocity and distribution of jet impinging on the target wall, thus influencing the heat transfer and flow performance of impinging cooling [16]. Therefore, in recent years, some investigators have started to focus on the effect of hole diameter variation (variation in inlet and outlet cross-section of the hole) on the heat transfer and flow characteristics and capability of impinging cooling [17], and have conducted some preliminary studies. Shuja et al. [18] analyzed the heat transfer and flow distribution behaviors of converging jet impinging on conical holes through numerical methods. The study indicated that the pressure loss coefficient and heat transfer coefficient both reach their highest values at the wall in the inlet and outlet areas of the conical hole. Carcasci et al. [19] compared the heat transfer capability of circular straight holes, circular inclined holes, and converging inclined holes through experimental methods. Their findings indicated that the heat transfer level of impinging cooling using converging inclined holes increases by about 10%-20%, as compared with circular straight holes. Ram and Sivamani [20, 21] analyzed the heat transfer characteristics of converging-hole impinging cooling by numerical calculation at different Reynolds numbers. They pointed out that converging-hole impinging cooling has higher heat transfer capability than that of straight-hole impinging cooling. Sathish et al. [22] researched the effect of angle $(10^\circ, 20^\circ, \text{ and } 30^\circ)$ of converging holes on the performance of impinging cooling at turbine blade leading-edge through numerical simulation. Their results showed that compared to the straight-hole impinging cooling, the converging-hole impinging cooling has better heat transfer performance. Zielinski et al. [23, 24] carried out experimental investigation on the flow characteristics of impinging cooling using variable-diameter holes. The study indicated that compared to jet impinging cooling using equal-diameter hole, the jet impinging cooling using variable-diameter hole can effectively enhance the fluid-flow momentum and achieve a heat transfer enhancement of about 35%. Chi et al. [25] conducted structural optimization design of variable-diameter jet hole under non-uniform thermal load conditions on the target wall. They obtained multiple sets of optimal structural parameter combinations that result in more even distribution of temperature on target wall.

Although some researchers have studied the heat transfer and flow behaviors of jet impinging using variable-diameter holes, most of these studies only focus on the effects of variable-diameter holes on the heat transfer enhancement of jet impinging cooling. Few researchers have investigated the influence of variable-diameter holes on the flow performance and comprehensive thermal performance of impinging cooling. Generally, the enhancement of heat transfer is inevitably accompanied by increased pressure loss and weakened flow performance. Therefore, in addition heat transfer performance, it is essential to assess the overall capability of impinging cooling using variable-diameter hole based on flow performance and comprehensive thermal performance.

To address the aforementioned issue, this study investigated the heat transfer and flow capabilities of impinging cooling using variable-diameter holes. Firstly, numerical calculation approach was used to compare and analyze the heat transfer and flow behaviors and performance of three different hole configurations (*i.e.*, converging hole with $D_2/D_{\rm in} = 0.5$, straight hole with $D_2/D_{in} = 1$, and expanded hole with $D_2/D_{in} = 1.5$) under different Reynolds numbers. Then, the effects of Reynolds number (ranging from 6000-30000), outlet-to-inlet diameter ratio $(D_2/D_{\rm in}, {\rm ranging from 0.5-0.9})$, and impinging height ratio $(H/D_{\rm in}, {\rm ranging from 1-4})$ on the heat transfer, flow, and comprehensive thermal performance of converging-hole impinging cooling were explored. Empirical correlations for the average Nusselt number, pressure loss coefficient, and comprehensive thermal coefficient of converging-hole impinging cooling were derived based on the Reynolds number, outlet-to-inlet diameter ratio, and impinging height ratio. Furthermore, the sensitivity of the performance variables of converging-hole impinging cooling to its input variables was analyzed. The research object and content of this article mainly focus on industrial applications of power equipment such as gas turbines and aviation engines. The research findings can provide valuable insights for the design of impinging cooling using variable-diameter holes in future advanced gas turbine blades.

Research object

The geometric model of the studied single jet impinging cooling using variable-diameter holes is displayed in fig. 1. The model includes the straight-hole section and variable-diameter section of the jet hole, the impinging zone, and the solid target wall. The solid target wall has a square cross-section with a width of W = 100 mm and a thickness of $\delta = 3$ mm. The total jet hole length is 20 mm, for which the first half being the straight-hole section of length $H_1 = 10$ mm and a diameter of $D_{in} = 10$ mm. The D_{in} is chosen as the characteristic length for calculating pressure loss coefficient, Nusselt number, Reynolds number, *etc.* The second half of the jet hole is the variable-diameter section of length $H_2 = 10$ mm. The jet hole diameter at the transition point is $D_1 = D_{in} = 10$ mm. The hole outlet diameter D_2 is the variable under investigation, ranging from 5 mm to 15 mm. The corresponding jet hole outlet-to-inlet diameter ratio (D_2/D_{in} , ratio of hole outlet diameter to hole inlet diameter) ranges from 0.5 to 1.5.



Figure 1. Geometric model

Where $0.5 \le D_2/D_{in} < 1$ represents converging hole, $D_2/D_{in} = 1$ represents straight hole, and $D_2/D_{in} = 1.5$ represents expanded hole. Under a fixed impinging height H = 10 mm, the heat transfer and flow behaviors of impinging cooling were compared for the converging hole $(D_2/D_{in} = 0.5)$, straight hole $(D_2/D_{in} = 1.0)$, and expanded hole $(D_2/D_{in} = 1.5)$. Because of the superior heat transfer capability of converging hole, this work particularly focuses on investigating the effects of different outlet-to-inlet diameter ratios and impinging height ratios on the heat transfer and flow performance of converging-hole impinging cooling. The values of outlet-to-inlet diameter ratios D_2/D_{in} are set to 0.5, 0.6, 0.7, 0.8, and 0.9, corresponding to outlet diameters D_2 of 5 mm, 6 mm, 7 mm, 8 mm, and 9 mm, respectively. The impinging height ratios $(H/D_{in}, \text{ratio of impinging height to jet hole inlet diameter)}$ are set to 1-4, with impinging heights H of 10 mm, 20 mm, 30 mm, and 40 mm, respectively. It is worth noting that the geometry parameters are varied while keeping the inlet cross-section of the jet hole constant to ensure the same inlet Reynolds number for the jet with the same cooling air mass-flow rate.

Research methods

Numerical calculation method

The heat transfer and flow behaviors of impinging cooling using variable-diameter holes were numerically solved using a steady-state, 3-D, and fluid-solid coupled approach in ANSYS CFX. The fluid-solid coupled computational model used in the study is shown in fig. 2(a). It consists of several parts, including the fluid domains of the straight-hole section, variable-diameter section, and impinging region, as well as the solid domain of the impinging target surface. The interface between the solid domain and fluid domain was processed as a coupled interface, with the same temperature and heat flux on both sides. In the numerical calculations, merely the equation of heat conduction was computed in the solid domain. The fluid domain was presumed as incompressible and gravity-independent turbulent flow. The finite volume approach on the basis of finite element was used to discretize and solve the conservation equations of mass, momentum and energy. The RANS equations were employed to calculate the turbulent momentum equations, and the Total Energy equation was used to solve the heat transfer problem. High precision discretization schemes were applied for the convective, source, and diffusion terms. The numerical solution was below 10⁻⁶.

The boundary conditions and parameters for the numerical calculations are: The inlet cooling air total temperature in the jet hole was set to 298.15 K. The inlet Reynolds number ranges from 6000-30000, corresponding to a cooling air velocity of 9.46-47.31 m/s. The inlet cooling air turbulence intensity was designated to 5%. The outlet boundary of the computational domain was specified with a given static pressure, the mean value and floating range of which are 101 kPa and $\pm 5\%$. An even heat flux of 1000 W/m² was given to the solid domain lower surface of the impinging target wall, which was the heated surface. The remaining surfaces were treated as no-slip and adiabatic walls. These boundary conditions were applied to simulate the heat transfer and flow behaviors of impinging cooling using variable-diameter holes.

The grid model for the numerical calculations is shown in fig. 2(b). The ANSYS ICEM was used to establish the grid model for the jet impinging cooling using variable-diameter holes. All computational domains were divided using the structured hexahedral mesh. The same division strategy was adopted for both the fluid and solid domains to minimize the numerical transmission errors between the meshes. The minimum mesh size was set as 0.05 mm. The total grid number was adjusted according to the maximum mesh size. The meshes near the wall surface were refined, with a mesh growth ratio of 1.2 and a first-layer mesh height of



0.001 mm. The aforementioned grid division method ensures that the wall y^+ is close to 1. Additionally, local mesh refinement was performed for the jet hole and its surrounding impinging region further enhance the accuracy of the numerical simulations.

Grid independence verification was conducted to assure the reliability and economy of the numerical approach for impinging cooling using variable-diameter holes. Five sets of grid models were examined in this work, with total mesh numbers of 500000, 710000, 1010000, 1510000, and 2270000. The specific case parameters for grid independence validation were set as Re = 18000, $D_2/D_{in} = 0.5$, and $H/D_{in} = 1$. The results in tab. 1 show that the mean Nusselt number on imping-

Table 1. Grid independence validation

Total mesh number/million	Nuave	Relative difference [%]
50	111.6	_
71	140.4	25.81
101	166.9	18.87
151	171.9	3.00
227	174.3	1.40

ing target wall raises with the increment of mesh number. When the mesh number is about 1510000, further increasing the mesh number brings a negligible influence on the average Nusselt number, with a relative variation within 1.5%. This indicates that grid independence is achieved. Therefore, the grid models for the jet impinging cooling using variable-diameter holes of different configurations will adopt the same division strategy as the fourth set of grids (1510000 meshes).

Numerical approach validation was also conducted to assure the accuracy of the numerical approach for impinging cooling using variable-diameter holes. Taking the test results of local Nusselt number for straight-hole impinging cooling in [26] as an example, the exactitude of numerical approach in the present work was validated. The parameters Re = 12000 and $H/D_{in} = 2$ were selected. Figure 2(c) presents the comparison among local Nusselt numbers gained by numerical calculations in this study using *k*- ω , RNG *k*- ε , and SST *k*- ω models, and the corresponding test data in [27]. From fig. 2(c), it can be observed that the distribution ten-

dency of local Nusselt numbers calculated by RNG k- ε model greatly differs from the test data, and within the range of $0 < R/D_{in} < 3$, its calculated values are much higher than the test data. The distribution trend of local Nusselt numbers calculated by k- ω turbulence model is also inconsistent with the test data to some extent, but the corresponding values are relatively close. The highest prediction deviation of k- ω model is approximately 16.37% when comparing the average values of local Nusselt numbers. The distribution trend and corresponding values of local Nusselt numbers computed by SST k- ω model are consistent to the test data. When comparing the mean values of local Nusselt numbers, the highest prediction deviation of SST k- ω model is approximately 5.89%. The aforementioned comparison indicates that the numerical approach in the current work, combined with SST k- ω model, can accurately simulate the heat transfer characteristics of single-hole jet impinging cooling. Thereby, the SST k- ω model was used for the numerical calculations of single-hole jet impinging cooling in all subsequent parts of this study.

Parameter definition

The inlet Reynolds number, for a jet hole is defined:

$$\operatorname{Re} = \frac{uD_{\operatorname{in}}}{v} \tag{1}$$

where $u \text{ [ms}^{-1]}$ is the cooling air velocity at the inlet of jet hole and $v \text{ [Pa} \cdot \text{s]}$ – the dynamic viscosity.

The local Nusselt number, based on the equivalent diameter of jet hole inlet is defined:

$$Nu = \frac{qD_{in}}{(T_w - T_{in})\lambda}$$
(2)

where $q \, [Wm^{-2}]$ is the target wall heat flux, $T_w \, [K]$ – the target wall temperature, and $T_{in} \, [K]$ – the cooling air temperature at jet hole inlet.

The pressure loss coefficient, C_p , is defined:

$$C_p = \frac{2(p_{\rm in} - p_{\rm out})}{\rho u^2} \tag{3}$$

where p_{in} [Pa] is the area-averaged pressure at jet hole inlet and p_{out} [Pa] – the area-averaged pressure at the computational domain outlet.

The comprehensive thermal coefficient, G, is defined [1]:

$$G = \frac{\operatorname{Nu}_{\operatorname{ave}}}{C_p^{-1/3}} \tag{4}$$

Result analysis and discussion

Comparison of impinging cooling performance of three types of holes

Figure 3 shows the distributions of heat transfer and flow characteristics for impinging cooling using three different types of holes, when Reynolds number is 18000 and H/D_{in} is 1. Figure 3(a) displays the distributions of flow velocity and fig. 3(b) displays the distributions of local Nusselt number. As depicted in fig. 3(a), the flow distribution characteristics of impinging cooling of the three types of holes are similar. Specifically, the cooling air-flow velocity in the free jet zone is high. The cooling air-flow velocity near target wall in the stagnation zone is very low. After that, the cooling air-flow velocity near target wall in the wall jet zone first rapidly increases and then gradually decreases. Finally, a vortex with large size and low velocity is

formed above the target surface near the target outlet side. As exhibited in fig. 3(a), the cooling air velocities of converging-hole impinging cooling in the free jet zone and wall jet zone are significantly higher than those of straight-hole impinging cooling and expanded-hole impinging cooling in the corresponding zones. Especially in the zone near target wall in the wall jet zone with $R/D_{in} < 3$, the cooling air velocity of converging-hole impinging cooling is even higher. This phenomenon may bring higher C_p for the converging-hole impinging cooling. As exhibited in fig. 3(b), the distribution characteristics of Nusselt number on the target wall for impinging cooling using three different types of holes are relatively consistent. Because the direct impinging of high speed jet destroys the heat transfer boundary-layer around the stagnation point and increases the disturbance effect of the surrounding fluid, the Nusselt number on the target wall directly below the impinging hole is very high, and a circular high heat transfer area is formed at the central region of target wall. After that, the Nusselt number of target wall progressively drops along the radial direction of impinging hole, and four low heat transfer regions are formed in the four corners of target wall. As also exhibited in from fig. 3(b), the Nusselt number of target wall for converging-hole impinging cooling are significantly higher than those for straight-hole impinging cooling and expanded-hole impinging cooling. Especially in the high heat transfer region in the center region of target wall, the Nusselt number of converging-hole impinging cooling improves more significantly. This is because the smaller outlet section of the converging hole makes the cooling air in the free jet zone impinge to the target wall at a higher flow velocity, and also effectively improves the cooling air velocity near target wall in the wall jet zone, thereby improving the local heat transfer performance of the target wall. In addition, the Nusselt number of target wall of straight-hole impinging cooling is also higher than that of expanded-hole impinging cooling. This is because the larger outlet section of expanded hole reduces the cooling air-flow velocity in the free jet zone and wall jet zone, weakening the disturbance of the cooling air and the damage to the boundary-layer, thus reducing the local heat transfer performance of target wall.



Figure 3. Heat transfer and flow characteristics distribution for three different hole configurations; (a) flow velocity distribution and (b) local Nusselt nimber distribution

In general, the heat transfer capability of converging-hole impinging cooling is the best, followed by straight-hole impinging cooling, while the heat transfer capability of expanded-hole impinging cooling is the worst.

Figure 4 shows the comparison of heat transfer and flow performance for impinging cooling using three different types of holes. Figure 4(a) shows the C_p distribution, fig. 4(b) shows the Nusselt number distribution on target wall, fig. 4(c) shows the Nuave distribution on target wall, and fig. 4(d) shows the G distribution, where Nu_{ave} is the area-averaged Nusselt number. As drawn in fig. 4(a), under various Reynolds number, the C_p of straight-hole impinging cooling and expanded-hole impinging cooling is close, but the C_p of straight-hole impinging cooling is relatively higher. The C_p of converging-hole impinging cooling is significantly higher than that of straight-hole impinging cooling and expanded-hole impinging cooling, and this phenomenon is more obvious with the increment of Reynolds number. Specifically, when Reynolds number raises from 6000-30000, the C_p of converging-hole impinging cooling is about 16.67-21.09 times higher than that of straight-hole impinging cooling. The C_p of expanded-hole impinging cooling are 22.55%-33.68% lower than that of straight-hole impinging cooling. Therefore, the flow performance of expanded-hole impinging cooling is the best, followed by straight-hole impinging cooling, and the flow performance of converging-hole impinging cooling is the worst. As exhibited in fig. 4(b), the Nusselt number of straight-hole impinging cooling and expanded-hole impinging cooling is close in whole target wall. While comparatively, the Nusselt number of straight-hole impinging cooling is higher than that of expanded-hole impinging cooling. Compared with the Nusselt number of straight-hole impinging cooling and expanded-hole impinging cooling, the Nusselt number of converging-hole impinging cooling has very high increase



Figure 4. Comparison of heat transfer and flow performance for three different hole configurations; (a) distribution of C_{ρ} , (b) distribution of Nusselt number, (c) distribution of Nu_{ave}, and (d) distribution of *G*

amplitudes in the range of $-3 < R/D_{in} < 3$, with the maximum increase rate of 3.26 times (relative to straight hole) and 3.82 times (relative to expanded hole), respectively. The increase amplitudes of Nusselt number of converging-hole impinging cooling in the ranges of $R/D_{in} < -3$ and $R/D_{\rm in} > 3$ are small, about 28.08% (relative to straight hole) and 39.01% (relative to expanded hole), respectively. From fig. 4(c), it can be found that under various Reynolds number, the Nu_{ave} of target wall of converging-hole impinging cooling is significantly higher than that of straight-hole impinging cooling and expanded-hole impinging cooling. The Nu_{ave} of target wall of straight-hole impinging cooling and expanded-hole impinging cooling is close. While comparatively, the Nu_{ave} of target wall of straight-hole impinging cooling is also slightly higher than that of expanded-hole impinging cooling. Specifically, when Reynolds number is 6000-30000, the Nu_{ave} of target wall of converging-hole impinging cooling is about 0.96-1.85 times higher than that of straight-hole impinging cooling. The Nuave of target wall of expanded-hole impinging cooling is reduced by about 5.62%-10.85% compared with that of straight-hole impinging cooling. Therefore, the heat transfer capability of converging-hole impinging cooling is the best, followed by straight-hole impinging cooling, while the heat transfer capability of expanded-hole impinging cooling is the worst. From fig. 4(d), it can be observed that when Reyniolds number is less than 24000, the expanded-hole impinging cooling exhibits the highest G, followed by straight-hole impinging cooling, while the converging-hole impinging cooling shows the lowest G. This indicates that when Reynolds number is less than 24000, the expanded-hole impinging cooling has the best comprehensive thermal performance, next is straight-hole impinging cooling, while the converging-hole impinging cooling has the poorest comprehensive thermal performance. However, when Reynolds number is equal to or greater than 24000, the G of impinging cooling under the three types of hole configurations is very similar, indicating a close similarity in their comprehensive thermal performance.

Because the converging-hole impinging cooling can bring great heat transfer enhancement, the subsequent content of this investigation will research the heat transfer and flow performance of converging-hole impinging cooling at different parameters.

Flow performance analysis of converging-hole impinging cooling

Figure 5 depicts the flow distributions of converging-hole impinging cooling under different parameters. As depicted in fig. 5(a), when Re = 6000, two vortexes with opposite directions appear in the wall jet zone of converging-hole impinging cooling. Taking the zone on the right side of the impinging hole as an example, the rotation direction of the vortex near the impinging hole is counterclockwise and the size is large, and the rotation direction of vortex near the outlet side is clockwise and the size is small. As Reynolds number raises from 12000-30000, only one large-scale counterclockwise vortex appears in the wall jet zone of converging-hole impinging cooling, and the size of the vortex almost remains unchanged under various Reynolds number. As also depicted in fig. 5(a), with the increment of Reynolds number, the range of high velocity region in the wall jet zone gradually increases. As can be found from fig. 5(b), when $D_2/D_{\rm in}$ is 0.5-0.8, the position and size of vortex in the wall jet zone of converging-hole impinging cooling do not significantly change. When D_2/D_{in} is 0.9, the size of vortex in the wall jet zone significantly decreases, and the position is closer to the target wall and target outlet. Besides, with the increase of D_2/D_{in} , the cooling air velocity near target wall in the free jet zone and wall jet zone significantly decreases, and the range of high velocity region in the wall jet zone also gradually decreases. As can be found from fig. 5(c), with the increment of H/D_{in} , the size of vortex in the wall jet zone of converging-hole impinging cooling becomes larger and larger, while the width of high velocity cooling air region in the free jet zone gradually decreases. Specifically, as $H/D_{\rm in}$ increases from 0.5-4, the corresponding width of high velocity cooling air region in the free jet zone decreases from the range of $R/D_{\rm in} \approx 0.39$ to the range of $R/D_{\rm in} \approx 0.34$. Besides, with the increment of $H/D_{\rm in}$, the range of high velocity region in the wall jet zone also gradually decreases.



Figure 6 depicts the variation curves of C_p of converging-hole impinging cooling under different parameters. As depicted in fig. 6(a), for cases $D_2/D_{\rm in} = 0.5$ and $H/D_{\rm in} = 1$, the increment of Reynolds number increases the C_p of converging-hole impinging cooling to a certain extent (as Reynolds number raises from 6000-30000, the C_p increases by 18.26%), and reduce its flow performance. Except for case at $D_2/D_{\rm in} = 0.5$ and $H/D_{\rm in} = 1$, the increase of Reynolds number indistinctively affects the C_p (*i.e.* flow performance) of converging-hole impinging cooling. As displayed in fig. 6(b), under various Reynolds number, with the increase in $D_2/D_{\rm in}$,

the C_p of converging-hole impinging cooling rapidly decreases. This is because when the inlet section of impinging hole is fixed, the outlet section of impinging hole is smaller when the D_2/D_{in} is small. This makes the cooling air velocity near target wall in the free jet zone and the wall jet zone very high, thus increasing the C_p in the whole process of jet impinging. When the $D_2/D_{\rm in}$ raises, the outlet section of impinging hole synchronously raises, and the cooling air velocity near target wall in the corresponding free jet zone and wall jet zone significantly decreases, thus reducing the C_p in the whole jet impinging process. Specifically, compared with $D_2/D_{\rm in} = 0.5$, the C_p of converging-hole impinging cooling at $D_2/D_{\rm in} = 0.9$ is reduced by about 91.47%-92.95% at various Reynolds number. Therefore, the increase of $D_2/D_{\rm in}$ can effectively improve the flow performance of converging-hole impinging cooling. As illustrated in fig. 6(c), when H/D_{in} is less than 2.0, the increase of H/D_{in} under different Reynolds number significantly reduces the C_p of converging-hole impinging cooling. When H/D_{in} is greater than 2.0, increasing H/D_{in} indistinctively affects the C_p of converging-hole impinging cooling, and the variation laws of C_p are not completely consistent under different Reynolds number. On the whole, when H/D_{in} is 2.0, the C_p of converging-hole impinging cooling is lower and the flow performance is better.



Figure 6. Variation curves of C_p of converging-hole impinging cooling under different parameters; (a) effect of Reynolds number, (b) effect of D_2/D_{in} , and (c) effect of H/D_{in}

Heat transfer performance analysis of converging-hole impinging cooling

Figure 7 displays the distributions of Nusselt number of converging-hole impinging cooling under different parameters. As illustrated in fig. 7, the distribution characteristics of Nusselt number on target wall of converging-hole impinging cooling under different parameters are similar. Specifically, a circular high Nusselt number region is formed at the central region of target wall, and four low Nusselt number regions are formed at the four corners of target wall. As for the influence of Reynolds number, it is shown in fig. 7(a) that with the increment of Reynolds number, the range of high Nusselt number region on target wall of converging-hole impinging cooling significantly increases. This shows that the increment of Reynolds number can greatly improve the heat transfer performance of whole target wall, especially in the central region of target wall. It is because the increment of Reynolds number strengthens the cooling air velocity near the target wall in the free jet zone and the wall jet zone, as depicted in fig. 5(a), intensifies the heat exchange between the target wall and the cooling air, and thereby improves the heat transfer effect of target wall. As for the influence of $D_2/D_{\rm in}$, it is shown in fig. 7(b) that with the increment of D_2/D_{in} , the range of high Nusselt number region on target wall of converging-hole impinging cooling significantly decreases. This shows that increasing $D_2/D_{\rm in}$ will weaken the heat transfer performance of whole target wall, particularly the heat transfer performance of target wall central region. This is because the increase of $D_2/D_{\rm in}$ reduces the cooling air velocity near the target wall in the free jet zone and the wall jet zone ($-2.5 \le R/D_{in}$) < 2.5), as shown in fig. 5(b), thereby weakening the local heat transfer capacity of target wall.

As for the H/D_{in} , it is illustrated in fig. 7(c) that with the increment of H/D_{in} , the range of the highest Nusselt number region (red circle) on target wall of converging-hole impinging cooling slightly increases, while the range of the second highest Nusselt number region (yellow circle) gradually decreases. This is mainly because the distribution state of the cooling air velocity in the free jet zone, as shown in fig. 5(c), changes with the change of H/D_{in} . It is worth noting that when $H/D_{in} = 2$, a red ring in the high Nusselt number area appears outside the red circle in the center of the target wall. That is, when $H/D_{in} = 2$, the Nusselt number of the target wall shows a trend of first decreasing, then increasing, and then decreasing outward from the center. On the whole, the H/D_{in} inconspicuously impacts the local heat transfer performance of target wall of converging-hole impinging cooling.



Figure 7. Heat transfer distribution of converging-hole impinging cooling; (a) effect of Reynolds number, (b) effect of D_2/D_{in} , and (c) effect of H/D_{in}

Figure 8 shows the change curves of Nusselt number and Nu_{ave} of converging-hole impinging cooling under different parameters. It is demonstrated in fig. 8(a) that the Nusselt number distribution laws of converging-hole impinging cooling are basically the same at different Reynolds number. With the increment of Reynolds number, the Nusselt number of converging-hole impinging cooling significantly increases. When the Reynolds number raises from 6000-18000, the increase range of Nusselt number of converging-hole impinging cooling is relatively large. When the Reynolds number increases from 18000-30000, the increase range of Nusselt number increases from 18000-30000, the increase range of Nusselt number of converging-hole impinging cooling is relatively smaller. As depicted in



Figure 8. Variation curves of Nusselt number and Nu_{ave} of converging-hole impinging cooling; (a) effect of Reynolds number, (b) effect of D_2/D_{in} , and (c) effect of H/D_{in}

fig. 8(a), with the increment of Reynolds number, the Nu_{ave} of converging-hole impinging cooling gradually increases, and the increase rate gradually decreases. Compared with Re = 6000, the Nu_{ave} of converging-hole impinging cooling at Re = 30000 is increased by about 1.62-2.65 times under different structural parameters. As demonstrated in fig. 8(b), the Nusselt number distribution laws of converging-hole impinging cooling are similar at different D_2/D_{in} , and the main difference is that there is a significant difference in Nusselt number at the high heat transfer region. Specifically, in the ranges of $R/D_{in} < -2$ and $R/D_{in} > 2$, the D_2/D_{in} inconspicuously impacts the Nusselt number of converging-hole impinging cooling. However, in the range of $-2 < R/D_{in} < 2$, the increase of D_2/D_{in} significantly weakens the Nusselt number of converging-hole impinging cooling. However, in the range of $-2 < R/D_{in} < 2$, the increase of D_2/D_{in} significantly weakens the Nusselt number of converging-hole impinging cooling. However, in the range of $-2 < R/D_{in} < 2$, the increase of D_2/D_{in} , the Nu_{ave} of converging-hole impinging cooling gradually decreases, and the decrease amplitude also gradually decreases. Compared with $D_2/D_{in} = 0.5$, the Nu_{ave} of converging-hole impinging cooling are similar at different Reynolds number is decreased by 43.61%-

60.07%. As displayed in fig. 8(c), the Nusselt number distribution laws of converging-hole impinging cooling are not completely consistent under different H/D_{in} . When H/D_{in} is 0.5 to 3, the Nu-sselt numberdistribution curves are double peak, while when H/D_{in} is 4, the Nusselt number distribution curve becomes single peak. In addition, in the ranges of $-3.1 < R/D_{in} < -1.2$ and $1.2 < R/D_{in} < 3.1$, the Nusselt number corresponding to H/D_{in} of 0.5 is higher than that corresponding to H/D_{in} of 1-4. In the ranges of $R/D_{in} < -3.1$ and $R/D_{in} > 3.1$, the Nusselt number corresponding to H/D_{in} of 1-4. As also displayed in fig. 8(c), with the increase of H/D_{in} , the Nu_{ave} of converging-hole impinging cooling under different Reynolds number all shows slow decreasing trends. Compared with H/D_{in} of 0.5, the Nu_{ave} of converging-hole impinging cooling at H/D_{in} of 4 is reduced by about 1.06%-6.25%, under different Reynolds number. Therefore, the H/D_{in} has little effect on the heat transfer capability of converging-hole impinging cooling.

Comprehensive thermal performance analysis of converging-hole impinging cooling

Figure 9 shows the curves of G of converging-hole impinging cooling under different parameters. As displayed in fig. 9(a), the G of converging-hole impinging cooling under different structural parameters increase with the increment of Reynolds number. It is because that the change rate of Nu_{ave} with respect to Reynolds number is greater than the change rate of C_p with respect to Reynolds number. Compared with Re = 6000, the G of converging-hole impinging cooling at Re = 30000 under different structural parameters is increased by about 1.58-2.45 times. As demonstrated in fig. 9(b), when Re = 6000 and 12000, the G of converging-hole impinging cooling gradually increases with the increment of D_2/D_{in} . When Re = 18000 and 24000, the G of converging-hole impinging cooling slightly decreases at first and then gradually increases with the increment of D_2/D_{in} . When Re = 30000, the G of converging-hole impinging cooling first raises and then drops with the increment of $D_2/D_{\rm in}$. The reason for the inconsistency is that the change rates of Nu_{ave} and C_p are not consistent at different D_2/D_{in} . As illustrated in fig. 9(c), the variations of the G of converging-hole impinging cooling with H/D_{in} are not completely consistent under different Reynolds number. But overall, the change curves of G are relatively flat, with a maximum change range of less than 4%. This shows that the impinging height ratio inconspicuously affects on the comprehensive thermal performance of converging-hole impinging cooling.



Figure 9. Variations of comprehensive thermal coefficient of converging-hole impinging cooling; (a) effect of Reynolds number, (b) effect of D_2/D_{in} , and (c) effect of H/D_{in}

Empirical correlations fitting

In order to facilitate the application of the research results of this study in the field of turbine blade cooling technology, the empirical correlations of Nu_{ave} , C_p , and G of the straight-

hole and converging hole impinging cooling were fitted. Due to the poor effect of expanded hole, the empirical correlation fitting in this study was only for straight hole and converging hole, that is, the range of outlet-to-inlet diameter ratio is $0.5 \le D_2/D_{in} \le 1$. Because the variation laws of Nu_{ave}, C_p , and G of the straight-hole impinging cooling and converging hole with Reynolds number, outlet-to-inlet diameter ratio and impinging height ratio are relatively simple, the power function [3] widely utilized in literature was adopted to fit the correlations.

According to the 50 sets of calculation results of the impinging cooling using straight and converging holes in this study, empirical correlation fitting was performed using the curve_ fit module in Python software. The specific fitting results are expressed:

$$Nu_{ave} = 0.0727 \operatorname{Re}^{0.689} \left(\frac{D_2}{D_{in}}\right)^{-1.474} \left(\frac{H}{D_{in}}\right)^{-0.0368}$$
(5)

$$C_p = 0.345 \operatorname{Re} 0.126 \left(\frac{D_2}{D_{\text{in}}}\right)^{-4.454} \left(\frac{H}{D_{\text{in}}}\right)^{-0.0337}$$
(6)

$$G = 0.11 \operatorname{Re}^{0.633} \left(\frac{D_2}{D_{\text{in}}}\right)^{0.0667} \left(\frac{H}{D_{\text{in}}}\right)^{-0.0232}$$
(7)

The application ranges of empirical correlations (5)-(7) is: $6000 \le \text{Re} \le 30000$, $0.5 \le D_2/D_{\text{in}} \le 1$, $0.5 \le H/D_{\text{in}} \le 4$. The maximum fitting deviations of the three empirical correlations (Nu_{ave}, C_p , and G) are 17.38%, 15.33%, and 16.17%, respectively, and the average fitting deviations are 5.73%, 8.78%, and 5.66%, respectively.

Analysis of parameter sensitivity

Sensitivity analysis is an approach that studies the extent to which changes in input variables affect output variables. Through sensitivity analysis, it can determine the importance of each input variable on the output variable, where a higher sensitivity index indicates a more important input variable. In this study, the widely applied Sobol approach on the basis of variance decomposition was adopted to execute the sensitivity analysis on the flow and heat transfer performance of the jet impinging cooling using variable-diameter holes. The Sobol approach can simultaneously compute the total sensitivity index, first order sensitivity index, and second order sensitivity index of each input variable.

The total variance, D, of output variable can be defined:

$$D = \sum_{i} D_{i} + \sum_{i \neq j} D_{i,j} + \dots + \sum_{i \neq j \neq \dots \neq n} D_{i,j,\dots,n}$$

$$\tag{8}$$

where D_i is the variance contribution of i^{th} input variable to output variable, $D_{i,j}$ – the variance contribution of the interaction between i^{th} and j^{th} input variables to output variable, and $D_{i,j,\dots,n}$ – the variance contribution of interactions among all input variables to output variable.

The calculation formula for first order sensitivity index S_i can be written:

$$S_i = \frac{D_i}{D} \tag{9}$$

The calculation formula for second order sensitivity index S_{ij} can be written:

$$S_{i,j} = \frac{D_{i,j}}{D} \tag{10}$$

Total sensitivity index S_{Ti} is the sum of the sensitivity indices of all orders, and it is given:

$$S_{\text{T}i} = S_i + S_{i,j} + \dots + S_{i,j,\dots,n}$$
(11)

where $S_{i, j, \dots, n}$ represents the higher-order sensitivity indices.

Figure 10 shows the sensitivity index distribution of performance variables (Nu_{ave}, C_p , and G) of the converging-hole impinging cooling to input variables (Reynolds number, D_2/D_{in} , and H/D_{in}). Figure 10(a) shows that for the Nu_{ave} of converging-hole impinging cooling, the first order sensitivity index of $D_2/D_{\rm in}$ is the highest, followed by the first order sensitivity index of Reynolds number, followed by the second order sensitivity index of the interaction between Reynolds number and D_2/D_{in} . While the sensitivity indices of other single parameters or the sensitivity indices of the interactions between other two parameters can be almost ignored. Figure 10(b) presents that for the C_p of converging-hole impinging cooling, the first order sensitivity index of D_2/D_{in} is also the highest, while the first order and second order sensitivity indices of other terms are very low. Figure 10(c) displays that for the G of converging-hole impinging cooling, the first order sensitivity index of Reynolds number is the highest, and the first order and second order sensitivity indices of other terms are almost negligible. Comprehensive analysis of figs. 10(a)-10(c) shows that the first order sensitivity index of each performance variable to each input variable is significantly greater than the second order sensitivity index. This shows that the direct influence of the input variables on the heat transfer performance, flow performance, and comprehensive thermal performance of converging-hole impinging cooling is absolutely dominant, while the interaction of the two input variables has little influence on the heat transfer performance, flow performance, and comprehensive thermal performance of converging-hole



Figure 10. Distribution of parameter sensitivity indices; (a) local sensitivity index of Nu_{ave} , (b) local sensitivity index of f, (c) local sensitivity index of G, and (d) total sensitivity index

impinging cooling. The results of fig. 10(d) illustrates that the Nu_{ave} of converging-hole impinging cooling has the highest total sensitivity index to D_2/D_{in} , has the higher total sensitivity index to Reynolds number, and has low total sensitivity index to the H/D_{in} . This shows that the $D_2/D_{\rm in}$ has the greatest effect on the heat transfer performance of converging-hole impinging cooling, followed by Reynolds number, while the H/D_{in} has little effect on the heat transfer performance of converging-hole impinging cooling. The C_p of converging-hole impinging cooling is very sensitive to the change in D_2/D_{in} , and its total sensitivity index is more than 0.99. However, the C_p of converging-hole impinging cooling is not sensitive to the changes in Reynolds number and H/D_{in} . Therefore, the D_2/D_{in} has the greatest effect on the flow performance of converging-hole impinging cooling, while the Reynolds number and H/D_{in} have little influence on the flow performance of converging-hole impinging cooling. The G of converging-hole impinging cooling is very sensitive to the change in Reynolds number, and its total sensitivity index is more than 0.99, but it is not sensitive to the changes in $D_2/D_{\rm in}$ and $H/D_{\rm in}$. This shows that Reynolds number has the greatest influence on the comprehensive thermal performance of converging-hole impinging cooling, while the $D_2/D_{\rm in}$ and $H/D_{\rm in}$ exhibit little effect on the comprehensive thermal performance of the converging-hole impinging cooling. The results of the parameter sensitivity analysis guide us in designing converging-hole impingement cooling for turbine blades, indicating that when considering exclusively on enhancing heat transfer or minimizing flow resistance, the selection of suitable hole outlet-to-inlet diameter ratio should be primarily focused. When considering both heat transfer and flow performance (*i.e.* comprehensive thermal performance), it is essential to focus primarily on choosing an appropriate Reynolds number.

Conclusions

The heat transfer and flow performance of jet impinging cooling with three different hole configurations were studied numerically in this paper. The interesting findings have been gained.

- Among the jet impinging cooling of the three types of holes, the converging hole exhibits the superlative heat transfer performance but the poorest flow performance. The straight hole shows moderate levels of both heat transfer and flow performance. The expanded hole exhibits the poorest heat transfer performance but the superlative flow performance.
- When Re < 24000, the expanded-hole impinging cooling shows the superlative comprehensive thermal performance, next is the straight-hole impinging cooling, and the converging-hole impinging cooling has the poorest comprehensive thermal performance. When Re > 24,000, the comprehensive thermal performance of the three types of hole configurations is similar. This may be because when Re > 24000, the influence of Reynolds number on the comprehensive thermal performance is dominant, rather than the shape of the holes.
- The increase in Reynolds number has little influence on the flow performance of the converging-hole impinging cooling. Increasing hole outlet-to-inlet diameter ratio effectively improves the flow performance of the converging-hole impinging cooling. When $H/D_{in} = 2.0$, the converging-hole impinging cooling has better flow performance. Therefore, a lager hole outlet-to-inlet diameter ratio and $H/D_{in} = 2$ are suitable for situations where the mass-flow rate of cold air is insufficient and flow performance needs to be improved, for the converging-hole impinging cooling.
- An increase in the Reynolds number effectively boosts the heat transfer performance of the converging-hole impingement cooling, whereas an increase in the hole outlet-to-inlet diameter ratio notably diminishes the heat transfer performance of the converging-hole im-

pingement cooling. The H/D_{in} has trivial influence on the heat transfer performance of the converging-hole impinging cooling.

- Increasing the Reynolds number leads to a significant improvement in the comprehensive thermal performance of the converging-hole impingement cooling, whereas increasing the hole outlet-to-inlet diameter ratio and the impinging height ratio has a marginal effect on the comprehensive thermal performance of the converging-hole impingement cooling.
- The sensitivity analysis results indicate that when considering exclusively on enhancing heat transfer or minimizing flow resistance, the selection of suitable hole outlet-to-inlet diameter ratio should be primarily focused. When considering both heat transfer and flow performance, it is essential to focus primarily on choosing an appropriate Reynolds number.

Acknowledgment

This research was funded by the Xi'an Jiaotong University Basic Research Business Fee Free Exploration Project (xzy012023071) and National Key Research and Development Program of China (2021YFF0602301).

References

- Xi, L., et al., Numerical Investigation and Parameter Sensitivity Analysis on Flow and Heat Transfer Performance of Jet Array Impingement Cooling in a Quasi-Leading-Edge Channel, Aerospace, 9 (2022), 2, 87
- [2] Wu, W., et al., Leading Edge Impingement Cooling Analysis with Separators of a Real Gas Turbine Blade, Appl. Therm. Eng., 208 (2022), 118275
- [3] Fan, X., et al., Cooling Methods for Gas Turbine Blade Leading Edge: Comparative Study on Impingement Cooling, Vortex Cooling and Double Vortex Cooling, Int. Commun. Heat Mass, 100 (2019), Jan., pp. 133-145
- [4] Forster, M., *et al.*, Experimental and Numerical Investigation of Jet Impingement Cooling on a Concave Leading Edge of a Generic Gas Turbine Blade, *Int. J. Therm. Sci.*, *164* (2021), 106862
- [5] Ekkad, S. V., et al., A Modern Review on Jet Impingement Heat Transfer Methods, J. Heat Trans-T ASME, 143 (2021), 064001
- [6] Liu, K., et al., A Novel Multi-Stage Impingement Cooling Scheme –Part II: Design Optimization, J. Turbomach, 142 (2020), 121009
- [7] Xu, L., et al., Numerical Simulation of Swirling Impinging Jet Issuing from a Threaded Hole under Inclined Condition, Entropy, 22 (2019), 1, 15
- [8] Oliveira, A. V. S., et al., Experimental Study of the Heat Transfer of Single-Jet Impingement Cooling on a Large Heated Plate Near Industrial Conditions, Int. J. Heat Mass Trans., 184 (2022), 121998
- [9] Salem, A. R., et al., Experimental and Numerical Study of Jet Impingement Cooling for Improved Gas Turbine Blade Internal Cooling With in-Line and Staggered Nozzle Arrays, J. Energ. Resour-ASME, 143 (2021), 012103
- [10] Plant, R. D., et al., A Review of Jet Impingement Cooling, International Journal of Thermofluids, 17 (2023), 100312
- [11] Pawar, S., et al., The Impingement Heat Transfer Data Of Inclined Jet in Cooling Applications: A Review, J. Thermal Science, 29 (2019), Dec., pp. 1-12
- [12] Singh, P., et al., Study of Flow Field and Heat Transfer Characteristics for an Interacting Pair of Counter-Rotating Dual-Swirling Impinging Flames, Int. J. Therm. Sci., 144 (2019), Oct., pp. 191-211
- [13] Yang, T., et al., Comparative Study on Flow and Heat Transfer Characteristics of Swirling Impingement Jet Issuing From Different Nozzles, Int. J. Therm. Sci., 184 (2023), 107914
- [14] Nourin, F. N., et al., Review of Gas Turbine Internal Cooling Improvement Technology, J. Energ. Resour-ASME, 143 (2021), 080801
- [15] Tepe, A. U., et al., Experimental and Numerical Investigation of Jet Impingement Cooling Using Extended Jet Holes, Int. J. Heat Mass Trans., 158 (2020), 119945
- [16] Zhou, J., et al., Numerical Investigation on Flow and Heat Transfer Characteristics of Single Row Jet Impingement Cooling with Varying Jet Diameter, Int. J. Therm. Sci., 179 (2022), 107710
- [17] Marzec, K., et al., Heat Transfer Characteristic of an Impingement Cooling System with Different Nozzle Geometry, Journal of Physics: Conference Series, 530 (2014), 012038

Xi, L., *et al.*: Study on Flow and Heat Transfer Performance of Single ... THERMAL SCIENCE: Year 2024, Vol. 28, No. 6A, pp. 4499-4517

[18] Shuja, S. Z., et al., Jet Impingement on a Tapered Hole: Influence of Jet Velocity and Hole Wall Velocities on Heat Transfer and Skin Friction, Int. J. Numer. Meth. Fluids, 60 (2009), 9, pp. 972-991

4517

- [19] Carcasci, C., et al., Experimental Investigation of a Leading Edge Cooling System with Optimized Inclined Racetrack Holes, Turbo Expo: Power for Land, Sea, and Air, American Society of Mechanical Engineers, 45714 (2014), V05AT12A034
- [20] Ram, C., et al., Computational Study of Leading Edge Jet Impingement Cooling with a Conical Converging Hole for Blade Cooling, ARPN J. Eng. Appl. Sci., 12 (2017), 22, pp. 6397-6406
- [21] Sivamani, S., et al., Numerical Analysis of Jet Impingement Cooling Using Converging Conical Hole for Blade Leading Edge, Gas Turbine India Conference, American Society of Mechanical Engineers, 58509 (2017), V001T03A009
- [22] Sathish, S., et al., Influence of Converging Conical Hole Angles on Jet Impingement Blade Cooling of Gas Turbine Blade Leading Edge, AIP Conference Proceedings, 2385 (2022), 120003
- [23] Zielinski, A. J., et al., Impingement Cooling Using a Variable-Diameter Synthetic Jet, Proceedings, 17th IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm), IEEE, San Diego, Cal., USA, 2018, Vol. 17, pp. 429-435
- [24] Zielinski A. J., et al., Flow Analysis of the Impingement of a Variable-Diameter Synthetic Jet, J. Flow Vis. Image Pro., 26 (2019), 2, pp. 127-148
- [25] Chi, Z., et al., Geometrical Optimization of Non-Uniform Impingement Cooling Structure with Variable-Diameter Jet Holes, Int. J. Heat Mass Trans., 108 (2017), Part A, pp. 549-560
- [26] Xu, L., et al., Flow and Heat Transfer Characteristics of a Swirling Impinging Jet Issuing from a Threaded Nozzle, Case Stud. Therm. Eng., 25 (2021), 100970
- [27] He, J., et al., Heat Transfer Enhancement of Impingement Cooling by Different Crossflow Diverters, J. Heat Trans-T ASME, 144 (2022), 4, 042001

Paper submitted: January 9, 2024 Paper revised: March 20, 2024 Paper accepted: March 29, 2024

2024 Published by the Vinča Institute of Nuclear Sciences, Belgrade, Serbia. This is an open access article distributed under the CC BY-NC-ND 4.0 terms and conditions.