An Experimental Study of Momentum-Preserving Shaped Holes for Film Cooling Using PSP and PIV

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An experimental investigation was conducted to study the performance of film cooling injection from a cylindrical hole and a “W-shaped” hole. A Pressure Sensitive Paint (PSP) technique was used to map the spatial distribution of film cooling effectiveness on the surface of the test plate based on a mass-transfer analogy. The effects of mass flux ratio and density ratio on film cooling effectiveness are investigated by performing PSP experiments at various mass flux ratio using two coolant gases. In order to investigate the kinematic relationship between film cooling effectiveness and mass flux ratio, a high resolution PIV system was used to conduct flow field measurements. Interactions between the mainstream flow and injection flow are also investigated to elucidate the kinematic mechanisms of film cooling.

Nomenclature

\( M = \) Mass flux ratio of the coolant and mainstream flow, \( \rho_c V_c / \rho_\infty V_\infty \)

\( I = \) Momentum flux ratio of the coolant and mainstream flow, \( \rho_c V_c^2 / \rho_\infty V_\infty^2 \)

\( DR = \) Density ratio of the coolant and mainstream, \( \rho_c / \rho_\infty \)

\( D = \) Diameter of hole

\( \eta_{\text{av}} = \) Adiabatic film cooling effectiveness

\( T_{\text{av}} = \) Adiabatic wall temperature

\( T_c = \) Temperature of coolant

\( T_\infty = \) Temperature of main stream

\( (C_{O_2})_{\text{main}} = \) Oxygen concentration of mainstream

\( (C_{O_2})_{\text{mix}} = \) Oxygen concentration of mainstream-coolant mixture

\( (C_{O_2})_{\text{coolant}} = \) Oxygen concentration of cooling flow

\( (p_{O_2})_{\text{air}} = \) Partial pressure of oxygen with air as the coolant

\( (p_{O_2})_{\text{mix}} = \) Partial pressure of oxygen with \( N_2 \) or \( CO_2 \) as the coolant

\( p_{\text{ref}} = \) Partial pressure of oxygen at reference state

\( I_b = \) Intensity of background noise

\( I_{\text{ref}} = \) Intensity of the excitation light at the reference state without flow

\( I_{\text{air}} = \) Intensity recorded with air as the coolant

\( I_{\text{mix}} = \) Intensity recorded with \( N_2 \) or \( CO_2 \) as the coolant

\( M_c = \) Molecular mass of coolant

\( M_{\text{air}} = \) Molecular mass of mainstream

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I. Introduction

Film cooling technology was born under the necessity to protect turbine blades from damage due to corrosion and melting despite the extremely high temperature of the stage in which they operate, which may be as high as 1700°C for certain gas turbine engines. By releasing a film of coolant gas on the surface of the turbine blades, the solid components can be protected from the high temperature flow through the turbine stage, thus extending their life. Because the greatest potential to improve the performance of gas turbine engines is realized through further increasing the temperature of the combustion process, there is an inherent desire to optimize the film cooling systems for significant economic savings. Thoroughly understanding the mechanism of the coolant film behavior is necessary for such optimization.

Many researchers over the past 40 years have contributed their talents and efforts to investigate the fundamental principle of film cooling to improve gas turbine engine performance. The performance of film cooling elements is significantly affected by several parameters: mass flux ratio (\(M\), commonly referred to as the blowing rate), momentum flux ratio (\(I\)), turbulence intensity of the mainstream, the coolant density ratio (\(DR\)), and the configurations of holes \(\cite{1}\). After summarizing recent research results from other researchers, Bogard and Thole \(\cite{2}\) listed a series of physical factors that affect the film cooling effectiveness. Their research shows that the interaction between the coolant jets and the mainstream flow result in mixing and decay of film cooling effectiveness, which is highly dependent on the mass- and momentum-flux ratios. Baldauf, \textit{et al.} \(\cite{3,4}\) utilized infrared thermography to study the influence of blowing rate, density ratio, and other factors on a flat surface with a row of cylindrical holes. Their results show that the lateral film cooling effectiveness increases with increasing blowing rate (before jet lift-off), and lower density ratios with higher momentum ratios promote the shear layer effects while enhancing interactions with adjacent jets. Pressure sensitive paint (PSP) was used by Johnson and Hu \textit{et al.} \(\cite{5}\) to investigate the relationship between film cooling effectiveness, mass flux ratio, and density ratio for coolant flowing from cylindrical holes, demonstrating results similar to those of Baldauf, \textit{et al.} \(\cite{3,4}\). The results of Wright \textit{et al.} \(\cite{6}\) showed that the turbulent mixing near the edge of film cooling holes increased and lead to reduced film cooling effectiveness. Meanwhile, their PIV results showed the augmentation of mixing between the jet and mainstream, driving coolant away from the test surface.

Besides cylindrical holes, researchers have designed a variety of novel hole geometries that can dramatically increase the performance of the coolant delivery: shaped holes. Rather than using simple cylindrical holes to inject coolant onto a turbine blade, a scooped outlet of some particular geometry is often employed to reduce the effluent velocity at the blade surface, reducing the likelihood of the coolant to separate and improving the spanwise distribution of the film. Goldstein \textit{et al.} \(\cite{7}\) demonstrated and quantified the remarkable enhancement of film cooling effectiveness using fan-shaped holes, an initial round cross section widened to 10° diffusion angle on each side. Both flow visualization and film cooling measurements showed that the secondary coolant jet would stay close to the wall as the mass flux ratio increases, whereas the jet of the cylindrical hole would detach from the surface at higher blowing rates. Other researchers, like Schmidt \textit{et al.} \(\cite{8}\) studied compound angle holes, Lu and Ekkad \textit{et al.} \(\cite{9}\) presented a crescent-shaped hole, Sargison \textit{et al.} \(\cite{10}\) demonstrated a console-shaped hole, Dhungel \textit{et al.} \(\cite{11}\) studied anti-vortex holes, and so on. All of these holes, compared with cylindrical holes, have their own aerodynamic characteristics and exhibit higher film cooling effectiveness within certain conditions. In fact, Bunker \(\cite{12}\) reviewed various kinds of shaped holes and made comprehensive comparisons of film cooling performance among them, showing advantages of each type of shaped hole.

In the present study, the film cooling effectiveness and associated flow field of a moment-preserving shaped hole are investigated experimentally using the PSP and PIV methods. The moment-preserving shaped hole (also called “w-shaped hole”) was designed by Shih \textit{et al.} \(\cite{13,14}\), and their CFD results show higher lateral film cooling effectiveness than the cylindrical hole at \(M=0.49\). With a nearly constant cross-sectional area for the injection flow, the momentum of the coolant is supposed to be held roughly constant despite the widening of the scoop from its leading to trailing edge. PIV measurements are also used to reveal the relationship between film cooling effectiveness and various other flow factors, such as \(M, I,\) and hole configuration. Interactions between the mainstream flow and injection flow and how the stream-wise vortical structures influence the film cooling efficiency are also investigated to elucidate the kinematic mechanisms of film cooling.
II. Experimental Setup and Test Model

A. Experimental rig and test model

The experimental studies were conducted in a low-speed, open-circuit wind tunnel located in the Wind Simulation and Testing (WiST) laboratory at the Department of Aerospace Engineering of Iowa State University. The tunnel has an optically-transparent test section with a 200 mm×125 mm cross section and is driven by an upstream blower. A honeycomb and screen structures are installed upstream of the contraction section to create a uniform low-turbulence incoming flow. The turbulence intensity in the test section was found to be less than 1.0%, as measured by a hotwire anemometer. Figure 1 shows the experimental setup of the PSP measurement. A 390 nm UV light source is projected on the PSP-coated test model. A 14-bit digital 2048×2048 pixel CCD camera fitted with a 610 nm long-pass filter is utilized to capture the emission light.

The cylindrical hole test model, shown in Figure 1, is made of hard plastic material and manufactured by a rapid prototyping 3D printer that builds the model layer by layer with a resolution of about 25 microns. A row of five coolant holes, with a diameter $D=4$mm, have a span-wise distance 3$D$ on-center between adjacent holes. The holes have a 35° inclination angle to the breakout plane of the test plate and no additional compound angle. The test model is mounted on a plenum chamber (details of the plenum can be found in Yang & Hu [15]) and sealed by a thin latex rubber gasket. Figure 2 shows the w-shaped hole test plate that has the same inclination angle and entry length 4$D$. In order to verify the CFD results of Shih [13], the geometry of the w-shaped hole is replicated according to their patent [14].

In the present study, the main-stream flow from the wind tunnel is provided to simulate the hot gas flow in typical turbine stage, while the coolant jet flow is supplied by a pressurized gas bottle. Depending on the desired density ratio and concentration of oxygen, various gases (air, N$_2$, CO$_2$, SF$_6$, and others) can be utilized as coolant. For the present experiment, N$_2$ and CO$_2$ are chosen to investigate effects of the density difference caused by temperature difference in real engine. Therefore, the corresponding density ratios ( $DR=\rho_c/\rho_\infty$ ) are 0.97 and 1.53 for N$_2$ and CO$_2$, respectively. The velocity of the main-stream flow is held constant at 25 m/s, whereas the flow rate of the coolant is adjusted to create mass flux ratios ( $M=\rho_cV_c/\rho_\infty V_\infty$ ) in the range of 0.2 to 1.8.
Figure 2. Cylindrical hole test plate used in the present study.

Figure 3. W-shaped hole test plate used in the present study.

B. Adiabatic film cooling effectiveness measurement using the PSP technique

The adiabatic film cooling effectiveness $\eta_{\text{av}}$ has been investigated for many years. It is defined based on temperature difference ratio between the mainstream and wall condition at the wall by:

$$\eta_{\text{av}} = \frac{T_m - T_{\text{av}}}{T_w - T_c},$$

where $T_m$ is the temperature of the main stream, $T_{\text{av}}$ is the adiabatic wall temperature of the studied surface, and $T_c$ is the initial temperature of the coolant flow.

In general, experimental measurements of film cooling effectiveness can be classified into two categories: “hot” methods and the “cold” methods. Hot methods, are based on conventional thermal measurement techniques, such as thermocouples [16], liquid crystal thermometry [17], infrared thermography [3, 4], and Temperature Sensitive Paint
(TSP)\(^{[18]}\), to measure the surface temperatures and then compute the adiabatic film cooling effectiveness directly. Though it is a straightforward way to reveal useful information of film cooling effectiveness, there always exists a primary concern and difficulty about the error of the adiabatic wall temperature measurement due to effects of heat conduction within the test plate. However, cold methods, such as the Pressure Sensitive Paint (PSP) technique\(^{[19]}\), can eliminate errors due to conduction successfully since it is based on a mass transfer analog to the temperature ratio of equation (1), freeing the measurement of concern about heat conduction. It is reasonable to use the mass transfer to analogy rather than heat transfer in the present experiment because the Lewis number \(Le \approx 1\) at atmospheric pressure.

In PSP measurements, the surface of the test model is painted with PSP which is sensitive to diatomic oxygen. The paint consists of light-sensitive molecules called luminophores mixed into a gas-permeable polymeric binder. When the luminophores are exposed to UV light, the luminophores are excited and fluoresce light at a longer wavelength, 650 nm for the particular paint used herein (ISSI UniFIB). However, in the presence of oxygen, the excited electrons can return to their ground state via a radiationless transition, a process called oxygen quenching\(^{[19]}\). Consequently, the intensity of the photoluminescence is inversely proportional to the concentration of oxygen, which means a reduced concentration of oxygen would cause higher intensity of emission light.

During the “cold” film cooling measurements, some coolant gas is chosen that contains no oxygen, whereas the main stream of the wind tunnel consists of atmospheric air, which is approximately 21% oxygen. Where the coolant gas is in contact with the PSP, the partial pressure of atmospheric oxygen is reduced, causing a subsequent reduction in oxygen quenching such that the photoluminescence may increase. The mass transfer analogy states that the film cooling effectiveness can be expressed according to the distribution of oxygen concentration along the surface of interest. Replacing the temperature in equation (1) with the concentrations of oxygen, the film cooling effectiveness can be written as following equation (2).

\[
\eta = \frac{(C_{o_2})_{main} - (C_{o_2})_{mix}}{(C_{o_2})_{main} - (C_{o_2})_{cooler}} = \frac{(C_{o_2})_{main} - (C_{o_2})_{mix}}{(C_{o_2})_{main}}
\]

Because the PSP method provides a means of measuring the partial pressure of the oxygen rather than its concentration, Charbonnier et al (2009)\(^{[20]}\) derived an expression (3) that accounts for the effect of the difference in molecular mass between the coolant gas and the main stream flow. By relating the concentration of oxygen to the partial pressure of oxygen ratio, they found the following expression for the film cooling effectiveness:

\[
\eta = 1 - \left( \frac{1}{\beta} \right) \left( \frac{M_c}{M_{air}} + 1 \right)
\]

where \(M_c\) and \(M_{air}\) are the molecular masses of the coolant gas and mainstream air, respectively.

The pressures listed in equation (3) can be calculated based on the normalized intensity of the emission light, which is directly related to the partial pressure of oxygen. As for the relationship between the normalized intensity ratio and normalized pressure ratio, it can be determined by calibrating the PSP. Through the calibration curve, the following relations between the normalized intensity ratios and partial pressure ratios can be found:

\[
\frac{I_{ref} - I_b}{I_{air} - I_b} = f \left( \frac{P_{o_2}}{P_{o_2}} \right)_{ref}
\]

\[
\frac{I_{ref} - I_b}{I_{mix} - I_b} = f \left( \frac{P_{o_2}}{P_{o_2}} \right)_{ref}
\]

where \(I_{ref}\) is the reference intensity, recorded at atmosphere pressure with no flow on, \(I_{air}\) is the emission intensity with the wind tunnel on and pure air injected as the coolant such that the oxygen concentration is uniform throughout the flow field, \(I_{mix}\) is the intensity the flow on and \(O_2\)-free coolant gas flowing. Finally, \(I_b\) is the background image, taken with the flow and illumination off to isolate effects of dark current and ambient.
illumination. Detailed information of how to conduct PSP experiment can be found in Johnson & Hu\cite{5} and Yang & Hu\cite{15}.

In the present study, a pressure and temperature-controlled test cell was used to calibrate the PSP, as shown in Figure 4. To conduct the PSP calibration, a copper test surface was painted with ISSI UniFIB pressure sensitive paint, which has low stated temperature sensitivity (about 0.5%/°C) and single-coat application. The painted surface of the test plate was set in a pressurized chamber and made visible through a quartz window while the other side of the test plate was exposed to a reservoir through which a thermally-regulated fluid was circulated. A 390 nm LED light source was used for the excitation light. The pressure within the chamber was measured with a digital sensor array (DSA 3217 Module, Scanivalve Corp, accuracy 0.05%). Air was removed from the test chamber by a vacuum pump. A thermostatic water bath was used to adjust the temperature of the test plate, and the temperature was monitored by a K-type thermocouple. The PSP calibration curve, which was based on the averaged intensity ratio and the averaged pressure ratio, is shown in Figure 5.

![Figure 4 Schematic of ISSI UniFIB paint Calibration.](image1)

![Figure 5 PSP calibration curve.](image2)

C. PIV Measurements

Figure 6 shows the experimental setup for PIV experiments. The mainstream airflow and the cooling jet were seeded with ~1 µm oil droplets generated from a theatrical fog machine and an oil droplet generator, respectively. A Nd:YAG laser was utilized to emit two pulses of 200 mJ at wavelength of 532 nm. Using a set of high-energy mirrors and optical lenses, the laser beam was shaped into a thin light sheet with a thickness in the measurement interest of about 1 mm. In order to reveal the interaction between the coolant flow and the mainstream flow, the illuminating laser sheet was aligned with the streamwise flow direction, bisecting the coolant hole. The experimental conditions, they remain the same with PSP experiments where the oxygen-free coolant gas is used.

The PIV system was used to measure the incoming flow boundary layer thickness at the leading edge of the coolant hole. Without coolant flow, the boundary layer thickness $\delta_{99}$ is $0.7D$ while the momentum thickness $\delta^{*}$ is $0.08D$. 


III. Result and Discussion

The PSP experiment was validated by comparing the centerline film cooling effectiveness of cylindrical hole with published results: Schmidt et al [22], Sinha et al [23], and Pedersen et al [24], which is shown in Figure 7. For $M=0.6$ in Figure 7(a), the centerline effectiveness for a cylindrical hole from the present study matches previous research results quite well on the whole. Though there is a slight difference in the range of $1<\alpha/D<10$, the trend is consistent with their results where $x/D$ is greater than 10. The discrepancy could be caused by differences of surface roughness within the coolant hole or other experimental conditions. For $M=1.0$ in Figure 7(b), the cases of all four results differ with each other in first eight diameters, beyond which they become similar.

![Figure 7. Comparison of Cylindrical hole centerline effectiveness to published data.](image)

A. The film cooling effectiveness distribution of cylindrical holes

The spatial distribution of film cooling effectiveness for varying mass flux ratios of different coolants is shown in Figure 8. On the whole, the length of the film cooling footprint along the streamwise direction first increases with increasing $M$ before then decreasing. The optimal point of mass flux ratio is about $M=0.6$ for CO$_2$ coolant flow, and $M=0.4$ for N$_2$ coolant flow. The spanwise distribution of coolant follows a similar trend versus $M$ as the streamwise...
distribution. When mass flux ratios of $M$ is smaller than the optimal point, the coolant flow covers the surface widely. However, when mass flux ratio is greater than the optimal value, the red contours (roughly $\eta=0.50$, as shown in legend) becomes smaller and narrower compared with other cases. Note that this point is the mass flux ratio where the coolant jet separates from the wall of test plate. When the momentum of the jet flow was sufficiently low, the jet emitting from the coolant hole would simply attach to the plate. However, as the velocity of the coolant gas increases, the coolant stream eventually detaches from the surface. One thing that needs to be pointed out is that, according to the definition of the mass flux ratio, for a fixed $M$, the coolant jet with a lower density ratio would have higher momentum ratio, $I$. For this reason, the N$_2$ coolant separates at a lower mass flux ratio than the CO$_2$ coolant flow.

![Comparison of spatial distribution of film cooling effectiveness for varying mass flux ratios.](image)

**Figure 8.** Comparison of spatial distribution of film cooling effectiveness for varying mass flux ratios.

Figure 9 shows the effects of mass flux ratios upon the film cooling effectiveness of a row of five cylindrical holes for CO$_2$, which has $DR=1.53$. The centerline and laterally-averaged effectiveness are shown, where the lateral averaged effectiveness here is performed over the domain of -3$<z/D<$3, or two full periods of the hole spacing pattern. Comparing the centerline effectiveness for various mass blowing ratios [Figure 9(a)], the film cooling effectiveness for all cases decreases along the stream-wise direction. For low-$M$ cases $M=0.2$, 0.4, and 0.6, the centerline effectiveness changes inversely with the mass flux ratio in the near-field for $x/D<4.5$. Beyond $x/D=5$, the performance for these three cases switch in that the centerline effectiveness increases with increasing $M$. This phenomenon can be explained by the fact that before the coolant separates from the surface, the increased momentum of the coolant jet has an increased likelihood of flowing further downstream before reattaching on the surface of interest. For a smaller mass flux ratio, the reattachment position is closer to the coolant hole and the trajectory of the coolant flow stays closer to the surface of interest in. Therefore, the centerline film cooling effectiveness is higher during the reattachment process for smaller $M$. Coolant injection for higher mass flux ratio delivers more coolant to cool the surface than a coolant flow with lower mass flux ratio, thus leading to higher film cooling effectiveness. Something that should be pointed out is that the above explanation is based on the assumption that the injection flow doesn’t completely separate from the surface of test plate. When $M$ is greater than 0.6, such as $M=0.85$ in Figure 9(a), the measurement results indicate that the jet flow has separated from the surface because of the low centerline effectiveness compared to the $M=0.2, 0.4$, and 0.6 cases. As $M$ increases greater than 0.85, the centerline effectiveness gradually decreases for as $x/D>5$. Figure 9(b) presents the laterally-averaged film cooling effectiveness for the cylindrical holes. It obeys a similar trend as the centerline effectiveness. For $M<0.85$, laterally-averaged effectiveness increases with $M$. But, after the coolant flow separates from the surface, the lateral averaged effectiveness decreases with $M$. 

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In order to gain more insight into the physics of film cooling effectiveness, quantitative comparison is made in Figure 10, which shows the effects of density ratio \( DR \) upon the film cooling effectiveness. According to the definition of mass flux ratio, for a fixed mass flux ratio, the momentum flux of the coolant increases as density ratio decreases. For example, the momentum flux of coolant with \( DR=0.97 \) (\( \text{N}_2 \)) is greater than that of coolant with \( DR=1.53 \) for constant \( M \). The behavior shown in Figure 10 (a) can be explained by this reality. For mass flux ratio of \( M=0.4 \), the centerline effectiveness of the \( \text{N}_2 \) coolant jet is higher than for the corresponding value of \( \text{CO}_2 \). As mentioned above, for \( M=0.4 \) the momentum flux of the \( \text{N}_2 \) jet flow is higher than that of the \( \text{CO}_2 \) jet flow. Thus, based on the results of Figure 9, the corresponding centerline effectiveness of \( \text{N}_2 \) coolant should be higher than \( \text{CO}_2 \) beyond \( x/D>5 \). Note that the laterally-averaged effectiveness appears to have little dependence on the density ratio at \( M=0.4 \), suggesting that the coolant remains well-attached to the surface and spreads nicely at lower \( M \) values, regardless of \( DR \). However, the centerline effectiveness greatly depends on \( DR \) for larger mass flux ratio, such as \( M=0.6 \) and 1.0, and thus coolant flows of higher \( DR \) have better centerline and laterally-averaged effectiveness.

B. Comparison of film cooling effectiveness between cylindrical and W-shaped holes

The results of the effectiveness measurements are displayed as color contour maps showed in Figure 11. For both cylindrical holes and W-shaped holes, the length of the effectiveness footprint ‘tails’ increases along the stream-wise direction when \( M \) is less than 0.6, while the lateral spread of effectiveness is rather wide. However, the
film cooling performance decreases sharply for \( M > 0.6 \) for cylindrical holes due to the separation of coolant jet from the plate surface. As for the W-shaped holes, the trends of the film cooling distribution is similar to the cylindrical holes, but the effectiveness footprint maintains a high performance for much higher \( M \). It seems that the coolant streams do not separate for the W-shaped holes.

![Figure 11. Comparison of spatial distribution of film cooling effectiveness for different hole geometries.](image)

In order to more-clearly explain the quantitative trends of the film cooling performance of W-shaped hole, a comparison of both centerline and laterally-averaged film cooling effectiveness was made for the cylindrical and W-shaped holes, shown in Figure 12. For the low-\( M \) cases of the W-shaped hole, such as \( M = 0.4 \) and 0.6 in Figure 12(a), the centerline film cooling effectiveness of \( M = 0.4 \) is slightly higher than the value of \( M = 0.6 \) at first, but the curves cross after about \( x/D = 6 \), beyond which point the \( M = 0.6 \) case shows higher effectiveness than the \( M = 0.40 \) case... From the analysis of Figure 11, it is known that the coolant injection from both types of holes tends to differ when \( M \) is bigger than 0.6, and the cylindrical jet flows seem to detach from the surface at \( M = 0.85 \). Therefore, it is straightforward to arrive at the same conclusion for the higher mass flux ratio cases (\( M = 0.85, 1.0, \) and \( 1.25 \)) shown in Figure 12(b). Since the jet flow separated at \( M = 0.85 \), the other jets with higher mass flux ratios certainly should drive further away from the surface. Therefore, the values of centerline effectiveness decrease dramatically with increasing of mass flux ratio. All in all, the main difference of centerline effectiveness between the non-separating cases (\( M = 0.4 \) and \( 0.6 \)) and the separating cases (\( M = 0.85, 1.0, \) and \( 1.25 \)) lies in the dramatically low effectiveness in the near-field, as shown in Figure 12(b).

Worthy of attention is that the peak location of centerline effectiveness moves downstream as \( M \) increases, from about \( x/D = 3.7 \) (trailing edge of shaped hole) to 6. Also, concerning the laterally-averaged film cooling effectiveness presented in Figure 12(c) & (d), the same trends are observed as for the centerline effectiveness except that the peak locations of laterally-averaged effectiveness do not tend to move downstream, rather appearing at the same streamwise location—the trailing edge of the shaped hole, which can be verified by checking the contour maps of coolant flow in Figure 11 where the jet flows reach their widest spread at the exit of shaped hole.

As described above, both types of hole tend to separate when mass flux ratio is greater than 0.6, so it is convenient to divide all the cases into a non-separation regime (Figure 12 (a)&(c)) and a separation regime [Figure 12(b)&(d)]. During the non-separation regime, there is little difference for the cylindrical and W-shaped holes in the centerline and lateral effectiveness for \( x/D > 7 \), suggesting that the coolant emitting from the two types of holes spreads uniformly and remains attached. But they do have differences for \( x/D < 7 \) in that the film cooling effectiveness of the W-shaped hole is somewhat higher than the cylindrical hole. Due to the laidback and scooped geometry of the shaped hole, the injections after the entrance passage tend to stay closer to the surface than...
cylindrical hole for $x/D<7$ region, while after passing the trailing edge of the shaped hole, all of the jets flow remain well-attached to the surface of the test plate. As for the separation-stage cases shown in Figure 12(b) & (d), the centerline and laterally-averaged film cooling effectiveness of the W-shaped hole is considerably higher than the cylindrical hole, particularly for the laterally-averaged effectiveness. There is more than 100% enhancement of the effectiveness for the $M=0.85$ and 1.25 cases at the trailing edge of the shaped hole compared with the cylindrical hole. The expansion geometry of the shaped holes significantly reduces the jet velocity, postponing the separation of coolant from the test surface. As $x/D$ increases, the film cooling effectiveness of both holes decreases, but the film cooling effectiveness of W-shaped hole remains slightly higher up to the end of our test plate.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig12.pdf}
\caption{Comparison of film cooling effectiveness between cylindrical and W-shaped holes for various mass flux ratios (CO$_2$).}
\end{figure}

Comparisons of film cooling effectiveness between cylindrical and W-shaped holes are also conducted for the N$_2$ coolant cases. As discussed above, for a fixed $M$, the momentum flux of the coolant jet is higher if a coolant of lower density ratio is used. Higher momentum would cause the jet to separate from the surface in a lower mass flux ratio for low $DR$. For the N$_2$–as coolant cases, the jet starts to lift off when $M$ is greater than 0.4 for cylindrical and W-shaped holes, as shown in Figure 13. It is obvious that the film cooling effectiveness of the N$_2$ cases follows the...
same pattern as for the CO\(_2\) cases in both the non-separation and separation regimes. The capability of film cooling of W-shaped holes is better than for simple cylindrical holes in the flow-separation regime.

![Figure 13. Comparison of film cooling effectiveness between cylindrical and W-shaped holes (N\(_2\)).](image)

C. PIV results

In order to study the interaction between the jet and mainstream flow, PIV was used to measure the velocity and TKE distributions in the near-field close to the injection hole. Figure 14 shows the comparisons of ensemble-averaged velocity fields between the cylindrical and W-shaped holes for mass flux ratios of \(M=0.4\) and \(M=1.0\). For the \(M=0.4\) case, there is little difference in the velocity distribution between the cylindrical and W-shaped holes, which is consistent with the PSP measurements. Because the mass flux ratio is low, the coolant would remain attached to the surface of the test plate for both types of hole. However, for mass flux ratio \(M=1.0\), both of the injections separate from the surface. The contour plot shown in Figure 13(c) indicates that there is strong interaction between the jet and mainstream, which causes the velocity distortion at the exit of the cylindrical hole, while for the W-shaped hole the interaction is comparatively lower due to the layback of the hole. As for the corresponding normalized TKE presented in Figure 15, the distributions of TKE for the cylindrical hole are similar with that of W-shaped hole, except that for the case where \(M=1.0\), the magnitude of the flow fluctuation for the cylindrical hole is much higher at the point of injection, which could be one of the reasons for coolant flow separation.

![Figure 14. Comparison of ensemble-averaged velocity fields between cylindrical and W-shaped holes for mass flux ratios of \(M=0.4\) and \(M=1.0\).](image)

(a) Cylindrical hole at \(M=0.4\) (b) W-shaped hole at \(M=0.4\)
IV. Conclusions

In the present study, a series of experiments were conducted to investigate the effects of blowing ratio and density ratio on the film cooling effectiveness of cylindrical holes and W-shaped holes. The PSP technique was utilized to investigate the spatial distribution of film cooling effectiveness on the surface of test plate. Meanwhile, the associated flow field of the interaction between the main and coolant flows were also studied by the PIV method. The measurement results showed clearly that:

1. Before the coolant jet separation from the surface, higher mass flux ratio would result in better film cooling effectiveness for $x/D$ greater than the 'merge point'. However, worse film cooling performance was observed for when the coolant jets detached from the wall.

2. A coolant with lower density ratio (such as $N_2$ in this study) would cause onset of separation of the coolant by enhancing the momentum ratio of the jet, resulting in reduced film cooling performance.
Because of the laidback and scooped geometry of the W-shaped hole, the velocity from the injection is reduced and, therefore, prevents the jet from separating. By using CO\textsubscript{2} as coolant, the film cooling effectiveness for the W-shaped hole is about twice the film cooling effectiveness for the cases of $M=0.85$, 1.0, and 1.25.

Based on the PIV results, the coolant stream from the W-shaped hole would stay closer to the test surface and has slightly lower TKE than for the cylindrical hole.

**References**

1. Bogard, D. G., *Airfoil Film Cooling*, The Gas Turbine Handbook, National Energy Technology Laboratory, 2006, Section 4.2.2.1