Experimental and Numerical Studies of the Film Cooling Effectiveness Downstream of a Curved Diffusion Film Cooling Hole

1. Introduction

Gas turbines are widely used in industry and military fields. To satisfy the increasing demand for turbine efficiency and power output, the temperature of gases in the combustion chamber is being increased continuously. Film cooling is an external method used in gas turbines to protect the components that are exposed to hot gases. The main idea is to introduce a secondary fluid from discrete film holes to the surface of components. The coolant forms a thin layer that covers the downstream region to prevent it from contacting hot gases. The goal is to obtain the maximum coolant coverage with the minimum aerodynamic losses.

Film cooling studies began as early as 1960 [1]. Hartnett et al. [2] studied the tangential injection and collected the heat transfer data at the conditions of both adiabatic and constant heat flux walls. Goldstein [3] presented a review that summarized the early film cooling geometries and research. Basic geometries like tangential injection layers, reverse coolant injection, and angled slots can be found in his study. Over the years, researchers [4–14] have improved the effectiveness by continuously developing different geometries. Bunker [15] summarized the literature on shaped hole film cooling over 30 years. He pointed out that the shaped holes have low sensitivity to mainstream turbulence intensity variations. And the greatest advantage of most shaped holes is that the effectiveness is increasing with the blowing ratio. However, the formation of kidney vortices is always a problem that prevents the coolant from attaching to the target surface.

Anti-kidney-vortex geometries have been under investigation as early as 1997 by Haven et al. [16]. Double-jet and sister-hole geometries have been investigated during the past decade [17-22]. Recently, Zhou et al. [23], Yao et al. [24], and Zhu et al. [25] have studied the effects of the diameter ratio of the side hole and main hole on film cooling effectiveness in double-jet and tripod holes. The interaction between
the upstream and downstream film cooling rows was the subject of investigation by Wang et al. [26] who concluded that such interaction weakens the effects of the kidney vortices. Some related US patents have also been registered by researchers in this area [27–29]. In this study, a diffusion hole is proposed that can be envisioned as a passage that can be created by the rotation of a circular hole about the axis z in Figures 1 and 2, normal to the film-cooled surface, and passes through the center of the metering hole cross-section. With today’s laser drilling technology, these holes are easily producible. With rotation angle \( \theta > 15^\circ \), these film hole geometries are believed to diminish the effects of kidney vortices without the help of any side holes. Each film hole starts with a circular metering hole, extending to about half of the wall thickness, and then expands laterally as shown in Figure 1. Due to the film hole curved exit area, the coolant velocity profile has its maximum along the axis of the hole exit area and decreases to a minimum close to the spanwise edges; thus, there is not a strong interaction between the coolant and the main (hot) flow around those edges resulting in no kidney vortices beyond a threshold value of the rotation angle, \( \theta \). Four cases, all with an inclination angle of \( \alpha = 30^\circ \) and four rotation angles, \( \theta = 7^\circ \), \( 14^\circ \), \( 17.5^\circ \), and \( 21^\circ \), shown in Figures 2 and 3, are studied at the blowing ratios of 0.5, 1, and 2. The conventional \( 7^\circ-7^\circ-7^\circ \) diffusion hole is used as the baseline case to which the results of these proposed film hole geometries are compared.

In the experimental part, the pressure-sensitive paint method, a proven technique, was used to measure film effectiveness. Using pure nitrogen gas as the coolant and air as the main (gas) flow, both at ambient conditions, the density ratio was close to 0.97.

2. Experimental Setup

Figure 4 shows the schematics of the test rig representing the air and nitrogen plenums as well as the removable test piece on which different film hole geometries were machined using a five-axis numerically controlled milling machine. Air and nitrogen plenums were fabricated with 0.127 cm thick clear acrylic plastic slabs. Each plenum was equipped with a honeycomb flow straightener to send a uniform flow to the main channel and the film holes. Air entered the main channel from the big plenum through a bellmouth opening. As shown in Figure 5, a 100 psi rotary-vane air compressor supplied the main flow corresponding to a Reynolds number of 87,200. An air dryer was connected to the compressor to remove the water vapor. Two cartridge filters were used to remove any residual water vapor from the air flow. The coolant was supplied by a commercially available nitrogen tank. Critical Venturi meters, choked at all inlet flow conditions, and precision pressure dials were used to control and measure the mass flow rates of air and nitrogen. K-type thermocouples were inserted through small holes into the test section and pipes to measure the flow temperatures using a temperature acquisition system. The removable test plate with three film holes was placed on the nitrogen plenum, flushed with the main channel bottom surface during each test. The reported data in all geometries are for the middle hole to eliminate any side effects. A 400 nm UV LED light (Thorlabs, M405L3 Mounted LED) and a CCD camera of 1600 by 1200-pixel resolution (ImageSource, DMK 23U274) with 610 nm light filter were installed above the test plate for data collection.

Test plates were made of 17.78 cm by 7.62 cm rectangular clear acrylic plastic of 1.143 cm thickness. Three equally spaced parallel film holes were drilled at a pitch to diameter ratio of \( p/d = 6 \) for each case. The machined test piece out of clear acrylic plastic, for \( \theta = 21^\circ \), before the pressure-sensitive paint was applied is shown in Figure 6. The baseline geometry, the commonly used \( 7^\circ-7^\circ-7^\circ \) diffusion hole, with which all new proposed film hole geometries were compared is shown in the top section of Figures 2 and 3. All film holes made an angle of \( \alpha = 30^\circ \) with the main flow direction (inclination angle), and they all had a circular inlet section of \( \ell = 2.5d \) long (Figure 2).

3. Pressure-Sensitive Paint and Calibration

Navarra [30] presented the pressure-sensitive paint (PSP) technique for the measurement of film cooling effectiveness.
Remarkable improvement was achieved over the Temperature-Sensitive Paint (TSP) and Infrared (IR) techniques. The PSP technique is conducted under isothermal conditions; thus, there are little to no heat losses during the tests. This is a great advantage compared to the conventional methods in which one must account for any possible heat losses when data are reduced. Therefore, there is one less uncertainty in the extracted data. Additionally, earlier film effectiveness measurement techniques called for multiple thermocouples due to their ease of installation. However, the limited number of thermocouples at discrete points on the target areas could not produce a continuous thermal image of the surface accurately. This shortcoming, compounded by the number of leads and thermal losses...
Associated with them increased the overall experimental uncertainty. In contrast, the PSP method generates a continuous image of the film cooling effectiveness (illuminance intensity) with high resolution over the entire surface under investigation. The UniFIB® pressure-sensitive paint from ISSI Innovative Solutions Inc was used in this study. This paint is a single component PSP optimized for a maximum luminescent signal while maintaining high pressure sensitivity and low temperature sensitivity. FIB Basecoat™ was used as the base coat before applying the PSP. This compound is sensitive to the oxygen partial pressure in the air-coolant mixture. When a UV light source (a 400 nm ultraviolet LED through a convex lens with 15 cm focal length, in this study) is cast on the painted surface, the paint illuminates with varying intensity depending on the partial pressure of the surrounding oxygen. This PSP property is exploited to our advantage to measure the film cooling effectiveness. Analogy between the mass and energy transport equations under identical boundary conditions (impermeable wall versus adiabatic wall) reveals that the film cooling effectiveness can be recast into

\[
\eta = \frac{C_{aw} - C_{co}}{C_{c} - C_{co}} = 1 - \frac{1}{1 + \left(\frac{P_{O_2,air}}{P_{O_2,ref}} - 1\right)M_{N_2}/M_{air}}. \tag{1}
\]

The intensity ratio which is a function of the oxygen partial pressure can be modelled as

\[
\text{IR} = \frac{I_{\text{ref}} - I_{\text{black}}}{I_{\text{ref}}} = f\left(\frac{P}{P_{\text{ref}}}\right) = f(P_{O_2}). \tag{2}
\]

Before the start of the film effectiveness tests, the PSP was calibrated to determine a functional relationship between the luminescence and the oxygen partial pressure. A small test coupon sprayed with the same paint was placed in a sealed chamber connected to a vacuum pump. The chamber pressure was reduced from the atmospheric pressure (101.3 kPa) to about 3 kPa while a precision vacuum gauge (Pirani gauge, 275 series, by Kurt J. Lesker Company) measured the pressure inside the chamber, and five thermocouples at different locations measured the temperature. During the calibration process, the same lighting and camera arrangements as in the film effectiveness tests were used to excite and record the light intensity. The test coupon was calibrated for different surface temperatures of 22.3, 25, 30, and 35°C. Details of the calibration setup are given in Baldino and Taslim [14]. Figure 7 shows the calibration results indicating that the functional dependence of luminescence with the oxygen partial pressure remained the same in that range of surface temperature. The 5th-degree polynomial representing the best fit to the data was used throughout this study to determine film effectiveness.

**4. Film Effectiveness Tests**

A typical test started with setting the targeted main flow (air) without any coolant flow. Proper lighting was then set on the target surface, and the camera was focused. Once the system reached equilibrium, a baseline photo was captured to serve as the reference light intensity to which all captured photos during a film cooling effectiveness test were compared. Next, the coolant (nitrogen) was turned on and was set to a predetermined mass flow rate corresponding to the desired blowing ratio. Two critical-flow Venturis (choked at all conditions) measured the main and cooling flow mass flow rates. These mass flow rates, divided by the main channel and the film hole inlet cross-sectional areas, respectively, gave the \((\rho U)_N\) and the blowing ratio:

\[
M = \frac{(\rho U)_N}{(\rho U)_{air}} = \frac{m_{N}/\pi d^2}{m_{air}/A_{passage}}. \tag{3}
\]

The freestream turbulence intensity in the main channel was measured to be 3.86%.

The vacuum pressure measurement sensor was found to be the main source of uncertainty. Since each test takes only

Figure 6: Machined test piece of \(\theta = 21^\circ\) before testing.

Figure 7: Calibration curve at different temperatures: 22.3, 25, 30, and 35°C.
a few minutes, the decay of PSP becomes insignificant [31]. Errors associated with the LED lighting and CCD camera as well as the slight temperature variations (Figure 7) are insignificant. Therefore, the film cooling effectiveness uncertainty, following the method of Kline [32], can be expressed as

\[ \delta \eta = \left( \frac{\partial \eta}{\partial P_{O_2}} \right) \delta P_{O_2}. \]  

The derivative of the film effectiveness with respect to the oxygen partial pressure from Equation (1) is

\[ \frac{\delta \eta}{\delta P_{O_2}} = - \frac{M_{N_2}/M_{air}21.21}{(M_{N_2}/M_{air}21.21/P_{O_2} + 1 - M_{N_2}/M_{air})^2 P_{O_2}}, \]  

where 21.21 is the mole fraction of oxygen in air. The vacuum pressure sensor accuracy \( \delta P \) depends on the measured pressure:

\[ \begin{align*} 
\text{if } P < 50 \text{kPa} & \rightarrow \delta P = \pm 10\% \text{ of reading}, \\
\text{if } P > 50 \text{kPa} & \rightarrow \delta P = \pm 2.5\% \text{ of reading}. \end{align*} \]  

Therefore, the uncertainty of the oxygen partial pressure (\( \delta P_{O_2} \)) is given by the uncertainty of the absolute
pressure times the oxygen concentration percentage within the air:

\[
\delta P_{O_2} = 0.21 \delta P - 0.10 P_{O_2}, \quad \text{if } P < 50 \text{kPa}, \\
\delta P_{O_2} = 0.21 \delta P - 0.025 P_{O_2}, \quad \text{if } P > 50 \text{kPa}. 
\]  

(7)

In conclusion, the maximum effectiveness uncertainty was calculated to be about ±5%. More details of the uncertainty analysis are discussed in Baldino and Taslim [14].

5. Computational Model

The computational domain included the main channel, the coolant plenum, and the five film hole geometries. The solid models were created in Solidworks and were imported to the powerful ICEM-CFD™ mesher by Ansys. Figure 8 shows the computational domain and details of the mesh distribution around a typical middle hole geometry with two symmetric side walls. Cells in all models were entirely hexahedral, a preferred choice for CFD analyses, and were varied in size.
bigeometrically from the boundaries to the center of the computational domain in order to have finer mesh close to the boundaries. The CFD analyses were performed using Fluent/UNS solver by Ansys, Inc., a pressure correction-based, multiblock, multigrid, unstructured/adaptive solver. Coolant and main channel air mass flow rates for the numerical models were identical to those of the experiments. They were at 300 K and 320 K, respectively. The main channel exit had a pressure boundary condition identical to that of the lab. The realizable $k$-$\varepsilon$ turbulence model was employed in combination with the enhanced wall function. Other turbulence models such as $k$-$\omega$ and standard $k$-$\varepsilon$ turbulence model were tried but did not reduce the difference with the experimental data. To study the details at the wall region, the average $y^+$ for the first layer of cells was controlled to be below 5 for all cases. The turbulence intensity of the main flow was set to be identical to that of the tests. The main flow boundary layer thickness was calculated to be around $5d$ so that the coolant was not disturbing its development. Mesh independence was achieved at about 3 million hexahedral elements for a typical model. All models, however, were run with a total number of hexahedral elements of about 4 million. Residual sums for all variables in all models were less than $1 \times 10^{-7}$. Convergence, for most cases, was achieved at around 30,000 iterations.

6. Results and Discussion

Experimental film cooling effectiveness contours downstream of the four new geometries are shown in Figures 9–12. Each figure represents three blowing ratios of $M = 0.5$, 1, and 2. The slight asymmetry in some contours is due to machining imperfections. Visual comparison of these contours shows that the case of $\theta = 21^\circ$ does a much more effective coverage of the target surface, and, in particular, for the larger blowing ratio of 2, the coolant extends farther, both axially and laterally. Apart from the $\theta = 7^\circ$ case, the area covered by the coolant increases as the blowing ratio increases from 0.5 to 2. In $\theta = 7^\circ$ geometry, its small exit area creates a strong jet which results in a blow-off, especially for $M = 2$. However, as $\theta$ increases, the larger hole exit area leads to the diffusion of coolant in the lateral direction that weakens the kidney vortices and consequently increases the film coverage. Similarly, the CFD contours of film cooling
effectiveness downstream of these four geometries are shown in Figures 13–16. These contours are in good agreement with those of the experimental results as $\theta = 21^\circ$ represents the best coverage, and the film cooling effectiveness increases with the blowing ratio. The double streaks in Figures 15 and 16 could be an indication of vortex generation due to higher diffusion of the film hole at the exit, compounded by the sharp edges modelled in the CFD runs. Figures 17–19, to be discussed shortly, show the migration of the main hot flow to the centerline for these cases thus reducing the film effectiveness along the centerline. The experimental geometry, however, had a slight rounding on the exit edge, dictated by the milling machine tolerances that prevented the formation of exit vortices.

The area-averaged effectiveness values between $y/d = (−3, 3)$ in Figures 9–16 are shown in Tables 1 and 2. The experimental film effectiveness results are slightly higher than those of the CFD analyses for $M = 0.5$ and 1.0. At $M = 2$, however, the experimental results fell below the CFD results. Other conventional turbulence models such as standard $k$-$\varepsilon$...
and $k-\omega$ with SST options were turned on in our CFD runs with no significant improvement in the final results. Both experimental and CFD results show that the case of $\theta = 21^\circ$ has the best performance with all blowing ratios. This behaviour is expected as the same amount of coolant emerges from an area with about 1.75 times the baseline hole exit area at an average velocity of about 57% of that in the baseline hole case with less interactions with the main flow. Therefore, it presents a more effective coverage of the target area. The spanwise-averaged film effectiveness variations of experimental as well as numerical results in the flow direction are shown in Figures 20–25. The solid lines represent the test data while the broken lines show the numerical results. It should be noted that $x/d = 0$ corresponding to the farthest exit, especially at higher blowing ratios, are expected. We included this geometry to see the threshold beyond which the proposed geometry performs better than the diffusion holes. The agreement between the test data and numerical results is fair and improves with the blowing ratio. It should be noted that $x/d = 0$ corresponds to the farthest exit point of the cooling hole on the target surface for

| $M$ | $\theta = 7^\circ$ | $\theta = 14^\circ$ | $\theta = 17.5^\circ$ | $\theta = 21^\circ$ | $7^\circ$–$7^\circ$ |
|-----|-------------------|-------------------|-------------------|-------------------|-------------------|
| Test | 0.5 0.1603 0.1936 0.1888 0.1830 0.1844 |
| CFD  | 1.0 0.1656 0.2538 0.2626 0.2734 0.2369 |
|      | 2.0 0.0893 0.2257 0.2654 0.3015 0.1973 |
| Test | 0.5 0.1296 0.1499 0.1562 0.1600 0.1127 |
| CFD  | 1.0 0.1703 0.2377 0.2583 0.2674 0.1585 |
|      | 2.0 0.1181 0.2867 0.3221 0.3417 0.1794 |

Table 2: CFD area-averaged film effectiveness from $x/d = 0$ to 40.

| $M$ | $\theta = 7^\circ$ | $\theta = 14^\circ$ | $\theta = 17.5^\circ$ | $\theta = 21^\circ$ | $7^\circ$–$7^\circ$ |
|-----|-------------------|-------------------|-------------------|-------------------|-------------------|
| CFD  | 0.5 0.1083 0.1237 0.1277 0.1250 0.1103 |
|      | 1.0 0.1328 0.1877 0.2046 0.2090 0.1636 |
|      | 2.0 0.1141 0.2474 0.2944 0.3073 0.1853 |

Figure 21: Streamwise variation of measured film effectiveness (spanwise-averaged at each point) for the blowing ratio of 1.

Figure 22: Streamwise variation of measured film effectiveness (spanwise-averaged at each point) for the blowing ratio of 2.
each geometry as is shown in Figure 3. The results are compared with the result of 7°-7°-7° diffusion hole, our baseline geometry. Several observations are made: For the blowing ratios of 1 and 2, the new proposed file hole geometries with \( \theta = 14°, 17.5°, \) and \( 21° \) perform much better than the 7°-7°-7° diffusion hole.

The inferior performance of \( \theta = 7° \) compared with the performance of the diffusion baseline hole is understandable as the \( \theta = 7° \) hole is very close to a straight-through cylindrical hole and the occurrence of jet lift-off and presence of kidney vortices.

Figure 26 represents a visual comparison of the film effectiveness performance for the five film hole geometries. With the same scales in all graphs, it can be seen that, as the rotation angle \( \theta \) increases, the film effectiveness in the stream-wise direction increases. It also shows that \( \theta = 17.5° \) and \( 21° \) geometries present superior film coverage, compared to the conventional 7°-7°-7° diffusion hole.

Figures 17–19 show the CFD vorticity magnitude contours along with the streamlines for all cases at all blowing ratios. These contours are generated on a plane normal to the target surface at \( x/d = 5 \). The baseline geometry, on the bottom section of these figures, and the \( \theta = 7° \) geometry, on the top section of these figures, are affected by the formation of kidney vortices that push the coolant away from the target surface, evidenced by the velocity direction on the streamlines. These two geometries, as we noticed in the previous sections, produced lower film effectiveness values. Figure 27 shows the bird’s eye view of the vorticity magnitude contours and velocity vectors on a plane at a distance of 0.05\( d \) from the target surface. The strong main flow and coolant interactions, represented by the large intersecting vectors, create kidney vortices that adversely affect the film coverage. Examples of those cases are the first and second cases in the third column. Table 3 gives a general performance evaluation of these five geometries based on the vorticity contour observations. The top and bottom rows correspond to these two geometries. As \( \theta \) increases, we see...
\[ \theta = 7^\circ \]
\[ \theta = 14^\circ \]
\[ \theta = 17.5^\circ \]
\[ \theta = 21^\circ \]

Test results

Figure 26: A comparison between the measured film effectiveness performance of the five film hole geometries.
Figure 27: CFD vorticity magnitude contours and velocity vectors at $z/d = 0.05$ (looking down on the target plate).

Table 3: Antivortex performance of different geometries.

| Geometry          | $M = 0.5$               | $M = 1.0$               | $M = 2.0$               |
|-------------------|-------------------------|-------------------------|-------------------------|
| $\theta = 7^\circ$| No antivortex action    | No antivortex action    | No antivortex action    |
| $\theta = 14^\circ$| Good antivortex action  | No antivortex action    | No antivortex action    |
| $\theta = 17.5^\circ$| Good antivortex action  | Good antivortex action  | Strong antivortex action|
| $\theta = 21^\circ$| Good antivortex action  | Good antivortex action  | Strong antivortex action|
| $7^\circ$-$7^\circ$ baseline | No antivortex action    | No antivortex action    | No antivortex action    |
the film effectiveness downstream the hole exit increases; (b) the best film effectiveness performance was presented by the hole with $\theta = 21^\circ$; (c) the smallest rotation angle, $\theta = 7^\circ$, exhibited the worst performance due to its smallest exit area that resulted in coolant lift-off; (d) for all tested blowing ratios, the cases of $\theta = 14^\circ$, 17.5°, and 21° showed superior film effectiveness performance, compared with the conventional 7°-7°-7° diffusion hole; and (e) the numerical results are in good agreement with the experimental data.

**Nomenclature**

- $A_{\text{passage}}$: Main flow passage area (38.71 cm$^2$)
- $C$: Chemical concentration (%)
- $d$: Film hole inlet diameter (Figure 3) (3.81 mm)
- $D_h$: Main passage hydraulic diameter (60.96 mm)
- $I$: Light intensity (pixel intensity value)
- $L$: Total axial length of each film hole (Figure 2)
- $\ell$: Feed (entrance) hole length (Figure 2, 9.53 mm)
- $m_{N_2}^\ell$: Nitrogen flow rate in each tripod (kg/s)
- $m_{\text{air}}^\ell$: Main channel air mass flow rate (kg/s)
- $M$: Blowing ratio ($\dot{m}_{N_2}/\pi d^2/4/\dot{m}_{\text{air}}/A_{\text{passage}}$)
- $M_{\text{air}}$: Coolant molecular weight (28 kg/kmol)
- $M_{\text{oxygen}}$: Mainstream (air) molecular weight (28.97 kg/kmol)
- $p$: Hole pitch (22.86 mm)
- $p$: Pressure (Pa)
- $P$: Pressure-sensitive paint
- $P_{O_2,\text{air}}$: Oxygen partial pressure in main channel approach air (Equation (1), 21 kPa)
- $P_{O_2,\text{mix}}$: Oxygen partial pressure at a given point downstream the film holes (Equation (1), kPa)
- $Re$: Reynolds number based on the passage hydraulic diameter ($\rho U D_h/\mu = 87200$)
- $T$: Temperature
- $T^*$: Dimensionless temperature
- $TSP$: Temperature-sensitive paint
- $U$: Main passage air velocity (m/s)
- $\beta_{\text{fl}}$: Shaped hole forward angle (Figure 2)
- $\beta_{\text{lat}}$: Shaped hole lateral angle (Figure 3)
- $\eta$: Film cooling effectiveness
- $\mu$: Air dynamic viscosity (kg/(m·s))
- $\theta$: Hole rotation angle (Figures 1 and 3)
- $\rho$: Air density (kg/m$^3$).

**Subscripts**

- $aw$: Adiabatic wall
- $blk$: Black
- $c$: Coolant
- $O_2$: Diatomic oxygen
- $N_2$: Nitrogen
- $ref$: Reference
- $\infty$: Channel mainstream.

**Data Availability**

Data files are available upon request.
Conflicts of Interest
The authors declare that they have no conflicts of interest.

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