THE EFFECT OF DIFFUSER SHAPE FOR FILM COOLING HOLES WITH CONSTANT EXPANSION ANGLES AND AREA RATIO

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ABSTRACT
Shaped film cooling holes are used in gas turbine components to deliver coolant to the high temperature surfaces of turbine blades and vanes to improve their durability. In general, shaped holes are created by expanding the outlet of the hole, resulting in a large area at the outlet of the hole that diffuses the flow. It has been shown in past studies that increasing the diffuser outlet to meter inlet area ratio causes a lower average momentum of the coolant jet at the hole exit, thereby producing better cooling performance. Instead of increasing the size of the diffuser section by increasing the area ratio, the present study focuses on changing the cross-section shape of the diffuser. This is done to mimic changes observed in the diffuser shape of conventionally manufactured film cooling holes. The present study utilizes 10-10-10 diffuser expansion angles and maintains a constant diffuser to meter area ratio. However, the diffuser shape is varied by changing the diffuser edge angle, \( \kappa \), located between the diffuser sidewall and the diffuser downstream wall. Three film cooling hole shapes were tested using three different diffuser edge angles, resulting in a narrow outlet, a wide outlet, and a standard outlet film cooling hole. Each hole shape was tested in a large wind tunnel with coolant supplied to the film cooling holes at three different blowing ratios by a co-flow and counterflow delivery channel, similar to the delivery method in a turbine vane with an internal baffle. In addition, the film cooling holes were tested with simulated diffuser roughness. Adiabatic effectiveness measurements indicate that film cooling hole performance is most impacted by diffuser roughness. The film cooling hole shape arising from the diffuser edge angle directly impacts the sensitivity to blowing ratio and coolant feed direction. Therefore, it is recommended that manufacturing of film cooling holes focus on reducing roughness in the diffuser for the highest performance. It is also recommended that the tolerance of the film cooling hole shape be biased towards wider film cooling holes to minimize sensitivity to the blowing ratio and coolant feed direction.

NOMENCLATURE

- A: area
- AR: area ratio, \( A_{\text{exit}}/A_{\text{inlet}} \)
- \( C_p \): specific heat capacity
- D: diameter of film cooling holes
- \( DR \): density ratio, \( \rho_c/\rho_\infty \)
- h: convective heat transfer coefficient
- k: thermal conductivity
- \( L_\infty \): length of mainstream
- L: hole length
- \( \dot{m}_c \): coolant mass flow rate
- M: blowing ratio, \( \rho_c U_{\text{jet}}/\rho_\infty U_\infty \)
- P: pitch, lateral distance between holes
- Pr: Prandtl number
- q: heat flow rate
- r: diffuser interior edge radius
- R: thermal resistance
- Ra: arithmetic mean roughness height along a line
- Re: Reynolds number, \( Re=U_\infty L/\nu \)
- t: hole breakout width
- \( \tau/\Pi \): coverage ratio
- \( Sa \): arithmetic mean roughness height across the surface
- T: temperature
- VR: velocity ratio, \( U_{\text{jet}}/U_\infty \)
- U: mean velocity
- \( U_{\text{jet}} \): coolant jet velocity in the direction of the metering section centerline
- x: streamwise distance measured from hole trailing edge
- y: vertical distance from surface
- z: spanwise distance measured from center hole

Greek Symbols

- \( \alpha \): hole injection angle
- \( \beta \): expansion angle for diffuser
- \( \eta \): local adiabatic effectiveness, \( (T_\infty-T_{\text{surf}})/(T_\infty-T_c) \)
- \( \kappa \): diffuser edge angle
- \( \nu \): kinematic viscosity
\[ \rho \text{ fluid density} \]

\[ \Omega \text{ vorticity} \]

**Subscripts**
- o flat plate conditions with no film cooling
- b bulk mean
- c coolant channel
- conv convection
- corr corrected
- exit exit plane of the film cooling hole
- fwd forward expansion of diffuser
- inlet inlet plane of the film cooling hole
- lat lateral expansion of the diffuser
- m metering section
- plate foam and particle board
- surf cooled surface
- tot total
- \( \infty \) mainstream

**Superscripts**
- \( \overline{\cdot} \) laterally-averaged
- \( ' \) per unit length

1. **INTRODUCTION**

Film cooling is utilized in gas turbines to reduce heat transfer to turbine blades and vanes with a goal of improving the durability and life of a gas turbine. Film cooling uses cool air from the compressor that bypasses the combustor section, and is sent to the surface of the airfoil through film cooling holes. To quantify the performance of film cooling on the surface, studies use the adiabatic effectiveness:

\[ \eta = \frac{T_{in} - T(x, z)}{T_{in} - T_c} \]  

(1)

The adiabatic effectiveness compares the temperature difference of the hot mainstream \( T_c \) and the cooled surface \( T(x, z) \) to the maximum achievable temperature difference of the coolant \( T_{in} \) and the mainstream \( T_{in} \).

Using conventional manufacturing, film cooling holes can be cylindrical, or have a shaped diffuser at the outlet. Shaped film cooling holes with diffusers are typically fed through a cylindrical meter section, with many diffusers defined by their lateral and forward angles. Shaped film cooling holes provide significant performance benefits as compared to cylindrical holes. Thus, research in shaped film cooling holes has improved effectiveness by experimenting on new film cooling hole geometry, determining the effect of individual parameters, and producing correlations for optimizing film cooling hole geometry. However, most research is focused on improving effectiveness through optimized geometry, and not on the potential for missed performance due to manufacturing.

Due to the small size of film cooling holes, turbine blades and vanes require a significant number of holes to provide adequate cooling performance. Simplifying geometry and increasing acceptable tolerances of film cooling hole parameters can significantly reduce the cost and time required in the manufacturing of holes.

The present study is unique as it investigates the effect of varying the diffuser shape for shaped film cooling holes with constant expansion angles and area ratio. This is accomplished by modifying the diffuser cross section shape from a standard rectangular shape to trapezoidal shapes that influence the width of the cooling hole footprint. In addition, the film cooling holes are fed by a channel that enables co-flow and counterflow fed film cooling. The sensitivity of the adiabatic effectiveness to roughness, diffuser shape, blowing ratio, coolant flow direction, and coolant flow speed is investigated.

2. **BACKGROUND**

The impact of manufacturing techniques for cylindrical holes and their effect on adiabatic effectiveness has been studied by Johnson et al. [1]. The manufacturing methods in the study introduced distortions to the intended cylindrical film cooling geometry with shape irregularity and higher surface roughness. These distortions, as well as other manufacturing defects, have been tested individually in literature, with a summary of manufacturing effects presented by Bunker [2]. With regards to the film cooling hole shape, a variation in the diffuser side angles caused the adiabatic effectiveness to vary between +16%/-13% [2]. To combat distortions in manufacturing, previous studies have optimized hole shape, to produce higher performance or insensitivity to defects, by adjusting the expansion angles [3, 4], area ratio [5], and proposing new film cooling hole geometries [6, 7].

Finding an optimum shaped film cooling hole typically involves increasing the diffuser expansion angles to produce wider areas of cooling. For example, multiple studies that optimized the expansion angles concluded that shaped film cooling holes should have a smaller forward expansion angle compared to the lateral expansion angles [3, 4]. The highest area averaged performance normalized to the coverage ratio (\( \overline{U}P \)) occurred when shaped film cooling holes lacked a forward expansion angle [4, 8]. Increasing the expansion angles only improves performance up to a point, as it has also been found that larger expansion angles will result in an increased chance of separation within the hole. Separation in the hole causes a decrease in performance due to the diffuser ingesting the mainstream flow as shown by Thole et al. [9] and increasing sidewall separation for the cooled surface as shown by Kohli and Bogard [10]. However, increased lateral expansion of film cooling holes widens the coolant jet coverage and can improve performance [3, 4, 5], especially when combined with the small forward expansion angle to decrease sidewall separation.

Another method of optimizing performance is to adjust the area ratio, which compares the meter cross sectional area to the diffuser cross sectional area at the outlet of the film cooling hole. One method of optimizing area ratio is to adjust the diffuser expansion angles. Thus, it was also found by Jones et al. [4] that the optimum wide 15-15-1 and narrower 12-12-6 holes both have an AR=3, while the results of Park et al. [3] show an optimum area ratio of 3.5. In order to adjust the area ratio without changing the expansion angles, Haydt et al. [5] increased the diffuser length to create larger diffuser cross sectional areas. Haydt et al.
[5] found that the adiabatic effectiveness improved with increasing area ratio for 7-7-7 and 12-12-12 film cooling hole geometries. An optimum cooling performance was then found based on a blowing ratio normalized by the area ratio.

Two other parameters used for shaped film cooling hole optimization of interest to this present study is the ratio of the breakout width of the hole footprint to the hole spacing (t/P), and the pitch-to-diameter ratio (P/D). Both parameters have been shown to have an effect on performance [11], especially near the film cooling hole (x/D<10), where jet-to-jet interaction is the highest, thereby causing the contours of effectiveness for different holes merge together [12]. For x/D>10, the lateral averaged adiabatic effectiveness for different P/D cases of the same conditions and geometry begin to converge to a similar value due to jet detachment or diffusion.

The optimization of film cooling hole shape does not guarantee performance in actual turbine components. In a study on in-hole roughness, Schroeder and Thole [13] found that roughness decreased adiabatic effectiveness relative to smooth holes. In addition, increasing blowing ratio further decreased the adiabatic effectiveness. Using LES, Zamiri et al. [14] determined that the presence of roughness induced more turbulent mixing and larger vortical structures.

Film cooling hole performance inside a turbine can also be significantly impacted by the coolant feed direction. Plenum fed film cooling holes have been useful for studying individual parameter effects, but channel fed film cooling has become increasingly popular due to the similarities to the actual coolant delivery in turbine blades and vanes. As opposed to plenum fed holes, which force all coolant into the film cooling holes, channel fed holes enable some coolant to flow past the film coolant holes. The amount of flow through a channel can increase while maintaining a constant blowing ratio as long as the channel outlet mass flowrate also increases. The channel can be mounted perpendicular to the mainstream for crossflow, or mounted parallel to the mainstream for coolant moving in the same direction as the mainstream (co-flow), or in the opposite direction (counterflow).

An increasing number of studies are indicating notable differences between plenum fed and channel fed film cooling holes. Inside the film cooling hole, Gunady et al. [15] showed that plenum fed shaped film cooling holes created a region of separation on the leeward side of the diffuser (the downstream side relative to the mainstream), near the inlet of the diffuser using LES and MRV simulations. The region of separation also caused the flow to accelerate on the windward side of the film cooling hole meter section. Thole et al. [16] used co-flow channel fed cylindrical holes, which caused the jet inside the film cooling hole to exit from the windward side to the leeward side when velocity in the coolant feed is increased. From the results in Fraas et al. [17], the coolant flow orientation severely impacted film cooling effectiveness for shaped 10-10-10 holes. Film cooling holes fed in co-flow improved their performance with increasing blowing ratio up to an optimum at M=2.5 [17]. McClinton et al. utilized a crossflow fed film cooling hole at a compound angle [18], where crossflow means that the coolant feed direction is perpendicular to the mainstream direction. Due to the compound angle rotating the film cooling hole relative to the mainstream, the hole could be fed with in-line crossflow or counter crossflow. In-line crossflow represented a film cooling hole feed where the coolant had to minimally turn to enter the hole (as opposed to counter crossflow, where the coolant might have to turn nearly 180 degrees to enter the hole). It was found that counter crossflow fed holes produced higher maximum adiabatic effectiveness than in-line fed crossflow or plenum fed cases [18]. The cause of this performance was suspected to be a lower averaged momentum at the exit of the film cooling hole due to a larger separation region inside the hole.

The goal of this study is to evaluate the effect of diffuser shape and manufacturing roughness on film cooling effectiveness, based on observations from actual parts. This is done specifically in the context of film cooling hole manufacturing tolerances and diffuser shape, not in the context of designing a new film cooling hole. Thus, the area ratio, meter length, and expansion angles were unchanged from the baseline geometry. Instead, the diffuser cross section shape is modified from the standard rectangular shape to trapezoidal shapes that influence the width of the cooling hole footprint. To our knowledge, this is the first study to look at this effect for a fixed area ratio and expansion angle. We attempt to hold these constant to avoid confounding effects. The effect of roughness is also considered for the geometry created in this study, due to the prevalence of roughness effects for novel hole manufacturing technologies.

Due to the sensitivity of diffuser shape to sidewall separation [9, 10], as well as the changes that can occur in the in-hole flow field due to channel fed holes [16, 17, 18], a co-flow channel was used to evaluate the geometries in the current study. The channel also enabled counterflow fed film cooling holes to test if the sensitivity to parameters changes similarly to the study by McClinton et al. [18].

3. EXPERIMENTAL METHODS
Adiabatic effectiveness measurements were conducted in the same closed-loop wind tunnel validated and described by Eberly and Thole [19] and shown in Figure 1. The mainstream air flow was circulated by an in-line fan, which kept the flow at 10m/s measured by a pitot probe in the test section. The mainstream was kept at a temperature of 323K using the heater bank. At the start of the mainstream test section, a suction loop removed the incoming boundary layer. Before the heater bank, some of the mainstream flow was redirected to supply the coolant loop. To prevent frost, the flow in the coolant loop was kept at a low dewpoint using a vent dryer filled with a solid desiccant. This was necessary due to the cryogenic temperatures experienced by the flow in the coolant loop. Inside the coolant loop, a heat exchanger fed by liquid nitrogen cooled the flow, with the gaseous evaporated nitrogen fed back into the flow. This enabled the coolant to reach 263K at the inlet of the film cooling holes.
The tunnel was modified from the previous plenum fed coolant loop system, to a new co-flow and counterflow coolant channel. The channel utilized a valve system to enable co-flow and counterflow coolant feeds from the same supply. In co-flow, the coolant and mainstream flowed in the same direction. While in counterflow the coolant flow was reversed relative to the mainstream. Two gate valves were mounted on opposite sides of the channel to control the channel inlet and outlet mass flow rates. The blowing ratio was adjusted by increasing the channel inlet mass flow rate from the coolant blower, and by reducing the channel outlet mass flow rate using the gate valve. The channel velocity ratio is the coolant average velocity divided by the mainstream velocity:

$$VR_c = \frac{U_c}{U_\infty}$$  \hspace{1cm} (2)

As shown in Figure 1, the inlet and outlet of the channel each had a venturi flowmeter. The difference of mass flow rate through the inlet and outlet venturi was divided by the total film cooling flow area (the number of holes times the metering diameter) to determine the mass flux of the cooling flow. This mass flux is equivalent to the mass averaged jet velocity times the coolant density, shown in Equation 3. This was then converted to a blowing ratio by using the mass flux of the mainstream, which was calculated using the mainstream velocity and density.

$$M = \frac{\rho_c U_{jet}}{\rho_\infty U_\infty}$$  \hspace{1cm} (3)

Figure 2 shows the co-flow and counterflow fed channel in detail. To encourage flow uniformity in the coolant channel, the inlet and outlet of the channel utilized plenums which fed into a 20 pore per inch (ppi) foam located 33D upstream and downstream of the film cooling hole inlets. The flow speed inside the coolant channel was measured by two pitot probes 12D upstream and downstream of the film cooling hole inlets. Type E thermocouples were used to measure the incoming and outgoing fluid temperature at the same locations as the pitot probes, as well as the temperature immediately below the inlet of the film cooling holes. The temperature below the inlet of the film cooling holes was used as the coolant temperature in the definition of adiabatic effectiveness. Static pressure taps were placed at various locations in the channel lower wall to ensure the static pressure was uniform laterally and to calculate the density of the coolant.

A FLIR T650sc infrared camera was used to measure the surface temperature of the foam test models at different locations. Sets of 10 IR images at 1 frame every 10 seconds were taken at each location and averaged. The IR camera was calibrated in situ by utilizing thermocouples attached to small copper plates (to generate a locally isothermal reference) which were embedded in the surface at x/D=3.4, x/D=19.6 and x/D=31.2 along the centerline of the outermost film cooling holes.

In order to account for conduction losses through the flat plate surface and into the coolant channel, the variation in bulk temperature of the coolant was first estimated using an overall thermal resistance between the hot mainstream and cold channel flow [20]:

$$\frac{T_\infty - T_b(x)}{T_\infty - T_c} = \exp\left(\frac{-x}{m_c C_p R'_\text{conv}(x)}\right)$$  \hspace{1cm} (4)

$$\frac{T_\infty - T_c}{T_\infty - T_b(x)} = \exp\left(\frac{-x}{m_c C_p R'_\text{conv}(x)}\right)$$  \hspace{1cm} (5)

with Equation 4 for co-flow and Equation 5 for counterflow. The freestream temperature $T_\infty$ and the film cooling hole inlet temperature $T_c$ were measured in the experiment for a given coolant flow condition. The total thermal resistance $R'_\text{conv}$ was determined as a function of channel length by using a fully developed turbulent coolant flow channel convection resistance, the conduction resistance of the foam plate plus its structural support material, and the external gas path convection resistance determined by:

$$R'_{\text{conv}}(x) = \frac{1}{h(x) A_{surf}}$$  \hspace{1cm} (6)

$$h(x) = \frac{x}{0.0296 Re_x^{\frac{4}{5}} Pr^{\frac{1}{3}}}$$  \hspace{1cm} (7)

where Eq. 6 is assuming a fully turbulent boundary layer. Once the local coolant bulk temperature variation was determined, the local conduction heat transfer rate through the plate was determined:
\[ q(x) = \frac{x}{R'_{tot}(x)} \times \left( \frac{T_\infty - T_b(x)}{\ln \left( \frac{T_\infty - T_c(x)}{T_\infty - T_c} \right)} - \frac{T_\infty - T_c}{T_\infty - T_c} \right) \]  (8)

The local surface temperature of the plate with this heat loss mechanism was then estimated:

\[ T_{surf}(x) - T_b(x) = q(x) \times (R_{plate} + R_{conv,c}) \]  (9)

This apparent reduction in plate top surface temperature due to conduction can be converted to a “no film cooling” effectiveness:

\[ \eta_o = \frac{T_\infty - T_{surf}}{T_\infty - T_c} \]  (10)

For this study, the “no film cooling” effectiveness varies from 0.09 to 0.16, depending on the mass flowrate through the channel. Finally, the adiabatic effectiveness is corrected for this conduction effect:

\[ \eta_{corr} = \frac{\eta - \eta_o}{1 - \eta_o} \]  (11)

For all experiments, the density ratio was kept constant at 1.3. Each geometry was tested at a blowing ratio of 1, 2, and 3, with the channel velocity ratio varying depending on the blowing ratio.

**Experimental Geometry**

The film cooling holes were machined in a low conductivity Dow Styrofoam \( (k=0.0288 \text{ W/m K}) \) hatch that was removable from the overall test plate. Five shaped film cooling holes with lateral expansion angles of 10 degrees and a forward expansion angle of 10 degrees, where the angles are defined at the meter centerline, were machined with a metering section diameter of \( D=0.843 \text{ cm} \) and a lateral spacing of \( P/D=4; \) see Figure 3(d).

Unique to this study, the diffuser cross section shape was adjusted from the common rectangular shape, but the area ratio (defined as the ratio of the area of the diffuser at the hole breakout perpendicular to the meter axis, to the area of the meter) was fixed. This was done to mimic the effects observed from different manufacturing techniques and their tolerance controls on film cooling holes. Some methods would result in wider trailing edge cross-sections akin to Figure 4(a), while others would cause wider leading-edge cross-sections as seen in Figure 4(c). To adjust the cross-section shape, two geometry parameters were defined at the diffuser exit area cross section as shown in Figure 3(a) and (c). The diffuser edge angle, denoted as \( \kappa \), is defined as the angle located between the diffuser sidewalls and the diffuser trailing edge. The diffuser interior edge radius, denoted as \( r \), is the radius between the diffuser sidewalls and the diffuser trailing edge. Typically, film cooling holes are designed such that the diffuser edge angle is perpendicular to the sidewalls as shown in Figure 4(b). Varying the diffuser edge angle results in trapezoidal cross-sections of the diffuser, and a distinct change in the footprint of the film cooling holes as shown in Figure 4(a) and (c). Adjusting the radius of the diffuser allows for the diffuser exit area cross-section to remain constant. Thus, these two parameters enable the diffuser shape to vary while ensuring the area ratio and centerline expansion angles are maintained.

Three diffuser edge angle film cooling hole geometries were used in this study: \( \kappa=75^\circ \), \( \kappa=90^\circ \), and \( \kappa=100^\circ \). The \( \kappa=90^\circ \) (the baseline nominal geometry) film cooling hole parameters shown in Figure 3 are also defined in Table 1. As shown in Table 2, to maintain an area ratio of three, the diffuser interior edge radius was increased for the \( \kappa=75^\circ \) film cooling holes and reduced for the \( \kappa=100^\circ \) film cooling holes. Both the \( \kappa=75^\circ \) and \( \kappa=100^\circ \) geometries were chosen as they represented the minimum and maximum diffuser edge angles achievable while maintaining an AR=3, equal interior edge radii, and 10-10-10 expansion angles. Due to changes in the cross-section shape, the breakout width varies with the diffuser edge angle. The lateral expansion angles were defined at the meter centerline, as indicated by the dotted line in Figures 3(c) and 4. Note that the diffuser edge angle impacts the lengths of the leading and trailing edges shown in Figure 3(c). At \( \kappa=75^\circ \), the diffuser has a longer trailing edge, as shown in the diffuser exit cross section geometry in Figure 4(a), whereas for \( \kappa=100^\circ \) the diffuser has a longer leading edge.
The uncertainty was checked with only co.

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levels of surface roughness, optical profilometry

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through some manufacturing methods to

create imperfect surface

create imperfect surface

Table 2. Variation of geometric parameters for $\kappa=75^\circ$, $\kappa=90^\circ$, and $\kappa=100^\circ$ shaped holes

<table>
<thead>
<tr>
<th>Feature</th>
<th>$\kappa=75^\circ$</th>
<th>$\kappa=90^\circ$</th>
<th>$\kappa=100^\circ$</th>
</tr>
</thead>
<tbody>
<tr>
<td>AR</td>
<td>3.00</td>
<td>3.00</td>
<td>3.00</td>
</tr>
<tr>
<td>t/D</td>
<td>2.8</td>
<td>2.6</td>
<td>2.4</td>
</tr>
<tr>
<td>r/D</td>
<td>0.525</td>
<td>0.4</td>
<td>0.225</td>
</tr>
<tr>
<td>t/P</td>
<td>0.7</td>
<td>0.65</td>
<td>0.6</td>
</tr>
</tbody>
</table>

In addition to the smooth geometries, the impact of slightly rough and rough diffuser surfaces was investigated as is possible through some manufacturing methods to create imperfect surface conditions, or contaminants in the cooling stream can make their way into the holes and deposit in the diffuser. To obtain relevant levels of surface roughness, optical profilometry scans of as-manufactured smooth and rough film cooling holes were performed. The surface contours from the optical profilometry can be seen in Figure 5, scaled to the film cooling hole diameter. The statistics of the roughness are listed in Table 3. To simulate the roughness measured in the manufactured film cooling holes at the scale of the wind tunnel experiment, a 180-grit aluminum oxide and glass bead sandblasting sand was adhered uniformly to the diffuser of a $\kappa=90^\circ$ geometry. A higher roughness 80-grit aluminum oxide and glass bead sandblasting sand was also adhered to the diffusers of the $\kappa=75^\circ$, $\kappa=90^\circ$, and $\kappa=100^\circ$ geometries in order to compare film cooling performance with high roughness. The levels indicated in Table 3 are higher than those observed from as-manufactured hardware, but serve as an upper bound of roughness and are in a similar range to Schroeder and Thole [13], who also studied diffuser roughness for the 7-7-7 shaped film cooling hole.

Computational Setup

In order to understand some of the in-hole differences for the three diffuser shapes, steady Reynolds Averaged Navier-Stokes (RANS) Computational Fluid Dynamics (CFD) solutions were created using STAR-CCM+, with the realizable k-epsilon turbulence model and ideal gas model for air with temperature dependent properties. A single hole pitch of the $\kappa=75^\circ$, $\kappa=90^\circ$, and $\kappa=100^\circ$ geometries including the coolant channel was tested, with periodic side boundaries, inlet velocity boundaries, and exit pressure boundaries for both the mainstream and coolant channels. The channel was modeled 12D upstream and 12D downstream of the meter’s inlet with only co-flow feed to the film cooling holes. The mainstream was modeled 16D upstream and 45D downstream of the film cooling hole exit.

The mesh had inflation layers with $y+<1$; the simulation used the all-$y+$ formulation (two-layer zonal model) which enables the realizable k-epsilon model to be used in this near-wall meshing formulation. The total mesh size was ~5 million cells, which was checked for grid independence. The solution was run for >8000 iterations at which point residuals and monitored quantities were unchanged. Due to the well-known inability of eddy-viscosity based steady RANS to properly predict adiabatic effectiveness, the solutions were not compared to the effectiveness measurements here, but several studies have indicated that the in-hole flow field is fairly well predicted [21, 22]. In this work, the CFD is used only to qualitatively understand the effect of the diffuser shape on the in-hole flow field.

Uncertainty Analysis

The precision uncertainty of the various measured quantities in this study are shown in Table 4. The uncertainty was calculated using 10 repeats of the same experiment geometry at the same parameters in a co-flow channel configuration. The bias uncertainty was calculated using Moffat’s constant odds general form equation [23]. The greatest source of error in the experiments was from the pressure transducers. Overall, the density ratio had the highest total uncertainty at 13.11%. However, this percent error only contributes to a variation of ±0.17 for the target density ratio of 1.3.
Table 4. Total uncertainty for each parameter

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Total Uncertainty</th>
</tr>
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<tbody>
<tr>
<td>M</td>
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</tr>
<tr>
<td>DR</td>
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<tr>
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<tr>
<td>VRc</td>
<td>1.82%</td>
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<td>ε</td>
<td>0.012</td>
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4. RESULTS AND DISCUSSION

The experiments are performed with co-flow and counterflow coolant feed configurations with a fixed mainstream velocity of 10m/s and a density ratio of 1.3. The adiabatic effectiveness from the center three holes of a 5-hole film cooling hole test plate are averaged for the laterally averaged adiabatic effectiveness. The center and left film cooling holes are used in the contour plots to show the jet-to-jet interaction present in all geometries due to the low P/D. In addition, as blowing ratio is increased, differences between length and width between film cooling hole jets of the same flow configuration and geometry can be seen. This effect is primarily attributed to the fluid flow in the co-flow and counterflow channel, as it is consistent in every geometry across blowing ratios.

Performance of the co-flow and counterflow channel is evaluated by comparing the \( \kappa=90^\circ \) geometry of this study to the results of Fraas et al. [17]. This study is chosen due to the similarity between geometry and the direct comparison of co-flow fed 10-10-10 film cooling holes. However, due to Fraas et al. using a higher density ratio of 1.7 the velocity ratio is used to compare performance instead of the blowing ratio.

Figure 6 compares the laterally averaged adiabatic effectiveness normalized to coverage ratio for similar velocity ratio cases from the \( \kappa=90^\circ \) film cooling hole geometry in this study to the geometry tested by Fraas et al. [17] described in Table 1. As shown by the figure, both film cooling holes provide the same performance at low velocity ratios. At higher velocity ratios, the normalized performance of the Fraas et al. geometry outperformed the \( \kappa=90^\circ \) baseline geometry of this study. However, this is expected due to the higher area ratio of their geometry, and for a fixed velocity ratio, their higher density ratio also results in a higher blowing ratio. There could also be some increased jet to jet interaction for the holes studied in this work, due to the low P/D (Table 1).

4.1 \( \kappa=90^\circ \) Geometry

The contours of adiabatic effectiveness for the \( \kappa=90^\circ \) film cooling hole geometry are shown in Figure 7. The contours are shown with increasing blowing ratio, which also increases the effective channel velocity ratio, as well as the two coolant feed directions. Note that the hole footprint is outlined in white in the figures for ease of identification.

In Figure 7, some flow separation can be seen inside the diffuser at all blowing ratios, due to the relatively high expansion angles of 10°. The location of this flow separation inside the diffuser does not appear to change significantly in size with increasing blowing ratio and changing film cooling hole feed direction. Figure 7 shows that increasing the blowing ratio for \( M=1 \) to \( M=2 \) in co-flow and counterflow increases the cooling downstream of the film cooling hole. However, increasing the blowing ratio from \( M=2 \) to \( M=3 \) decreases the spread and magnitude of cooling on the surface. There are no significant differences between co-flow and counterflow fed \( \kappa=90^\circ \) film cooling holes seen in Figure 7 when comparing the same blowing ratio.

The laterally averaged adiabatic effectiveness for the \( \kappa=90^\circ \) geometry at various blowing ratios is shown in Figure 8. Low blowing ratio cases produce the best performance at \( x/D<5 \), while the \( M=2 \) cases produce better cooling further downstream.
The M=3 cases perform worst closest to the film cooling hole exit where the jet has started to detach. Also, this figure shows that the $\kappa=90^\circ$ film cooling holes do not have a significant change in lateral average effectiveness when fed in co-flow or counterflow.

Figure 9 compares the changes in lateral average adiabatic effectiveness due to varying the channel velocity ratio and channel feed direction for the $\kappa=90^\circ$ geometry at a fixed blowing ratio of M=3. Increasing the channel velocity ratio causes a decrease in adiabatic effectiveness, due to the more significant redirection of the coolant at the meter inlet at high coolant velocities. Co-flow cases are slightly more sensitive to changes in channel velocity ratio than the counterflow cases, likely because the coolant already has to turn backward from the channel into the hole for a counterflow configuration.

### 4.2 $\kappa=75^\circ$ Geometry

The contours of adiabatic effectiveness for the $\kappa=75^\circ$ geometry can be seen in Figure 10. Low blowing ratios result in wider contours of adiabatic effectiveness. However, the persistence of the high centerline effectiveness does not change significantly as blowing ratio increases, which is unique to the $\kappa=75^\circ$ geometry. The magnitudes of effectiveness also do not change significantly between co-flow and counterflow, with the exception of the low blowing ratio case where the coolant persists further downstream for co-flow versus counterflow.

The $\kappa=75^\circ$ lateral average adiabatic effectiveness for various blowing ratios and feed directions is shown in Figure 11. Overall, increasing the blowing ratio decreases the near hole adiabatic effectiveness, but as was observed in the contours, the effectiveness converges to a relatively similar level for all blowing ratios beyond a x/D of 20. For M=1, the $\kappa=75^\circ$ co-flow and counterflow fed holes have the same performance until an x/D=15, at which the co-flow fed holes begin to outperform the counterflow fed holes due to the jet remaining attached as shown in Figure 10. At M=2, there is no difference between lateral average effectiveness for co-flow or counterflow. As shown in Figure 11, the $\kappa=75^\circ$ geometry is insensitive to changes in the lateral average effectiveness due to channel feed within the experimental uncertainty, except at M=3, where counterflow provides a slight improvement to effectiveness.

Though not shown here, the $\kappa=75^\circ$ geometry is insensitive to the channel velocity ratio at a fixed blowing ratio. Both co-flow and counterflow fed film cooling holes at $\text{VR}_c=0.1$, $\text{VR}_c=0.2$ for M=2, and $\text{VR}_c=0.2$, $\text{VR}_c=0.3$ for M=3 did not have any differences in the adiabatic effectiveness to within the experimental uncertainty.

<table>
<thead>
<tr>
<th></th>
<th>Co-Flow</th>
<th>Counterflow</th>
</tr>
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<tbody>
<tr>
<td>M=1</td>
<td>$\text{VR}_c=0.10$</td>
<td>$\text{VR}_c=0.10$</td>
</tr>
<tr>
<td>M=2</td>
<td>$\text{VR}_c=0.22$</td>
<td>$\text{VR}_c=0.22$</td>
</tr>
<tr>
<td>M=3</td>
<td>$\text{VR}_c=0.34$</td>
<td>$\text{VR}_c=0.34$</td>
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Figure 10. Adiabatic effectiveness contours for the $\kappa=75^\circ$ geometry
The lateral average adiabatic effectiveness for $\kappa=100^\circ$ with varying blowing ratio and channel feed can be seen in Figure 13. At $M=1$, co-flow and counterflow fed holes perform similar to each other, producing the highest near hole performance. However, this high performance decreases rapidly further away from the film cooling hole. The co-flow fed cases all decrease in performance as the blowing ratio increases, indicating that co-flow fed $\kappa=100^\circ$ holes are highly sensitive to changes in blowing ratio. Overall, the $\kappa=100^\circ$ geometry performs differently in co-flow and counterflow, with counterflow providing more coolant over the surface, thereby suggesting better jet attachment. In counterflow, the lowest blowing ratio $\kappa=100^\circ$ case provides the highest near hole performance, while increasing the blowing ratio to $M=2$ provides higher cooling downstream similar to the $\kappa=90^\circ$ geometry in Figure 8.

Figure 14 compares the changes of effectiveness due to the channel velocity ratio at $M=3$ for the $\kappa=100^\circ$ geometry.
As shown in Figure 14, the $\kappa=100^\circ$ hole is very sensitive to changes in the channel velocity ratio in co-flow. Counterflow fed $\kappa=100^\circ$ holes are insensitive to changes in the channel velocity ratio. From the results of Brundage et al. [24], the velocity profile at the exit of the film cooling hole can change due to the direction of the coolant feed. Specifically, counterflow fed holes have a higher mean velocity on the leading edge of the diffuser compared to a co-flow fed hole. In this study, it is expected that counterflow fed holes are insensitive to the inlet channel velocity because the flow must rotate 150° in order to enter the film cooling hole. In contrast to this, co-flow fed holes are more sensitive to channel velocity due to the flow only rotating by 30°.

4.4 Comparison of Hole CFD Results

In order to better understand the effect of the diffuser edge angle, a steady RANS CFD solution was created for each geometry at $M=3$ and $VR_c=0.3$. Figure 15 shows the predicted normalized streamwise velocity of the $\kappa=90^\circ$ geometry at the centerline plane of the hole. The black contour divides between positive velocity and negative (backward flowing) velocity. It is well known that the flowfield is not uniform inside of a shaped hole [15, 25], with high velocities near the leading edge of the hole and a separation region enclosed in the black contour near the trailing edge (downstream face) of the diffuser shown in Figure 15.

Figure 16 shows the normalized streamwise velocity for all geometries at the diffuser inlet, exit plane, $\chi/D=0$, and two intermediate planes. One intermediate plane is located between the diffuser inlet and exit plane and the other is located between the exit plane and $\chi/D=0$. Because each of the simulations had $M=3$ and $VR_c=0.3$, there are no significant differences between the velocity fields of the geometries at the inlet of the diffuser to the diffuser exit area. As shown in Figure 16 (a), the coolant jet from the $\kappa=75^\circ$ geometry does not change size, shape, or magnitude significantly inside the diffuser. This is in direct contrast to both the $\kappa=90^\circ$ and $\kappa=100^\circ$ geometries. Both of these geometries spread the jet wider and create two high velocity cones that dissipate by $\chi/D=0$, with the widest width jet created by the $\kappa=100^\circ$ geometry. These results are expected, as the $\kappa=75^\circ$ has the smallest leading edge, while the $\kappa=100^\circ$ geometry creates the widest leading edge, resulting in the largest contour of high velocity flow on the top of the coolant jet.

Figure 17 shows the normalized streamwise vorticity at $M=3$ and $VR_c=0.3$ for each geometry at the same locations as Figure 16. In all geometries, the diffuser inlet has a pair of counter rotating vortex structures that originate in the meter.
As the flow progresses to the diffuser exit, the counter rotating structures migrate toward the leading edge of the hole, and structures with opposite vorticity start to develop in the upper corners of the diffuser. At \( x/D = 0 \), or the hole breakout, the vortex structures that were developing in the upper corners of the diffuser dominate the flow and become part of the counter rotating vortex pair (CRVP), which has been studied widely [26, 27, 28, 29].

The insensitivity of the \( \kappa = 75^\circ \) geometry to feed direction is supported by Figure 17 (a). When compared to the vorticity contours of the \( \kappa = 90^\circ \) geometry in Figure 17 (b), both holes have vortex structures of similar size and magnitude at the diffuser inlet and \( x/D = 0 \) planes. At the diffuser exit plane, the vortical structures in the upper corners of the \( \kappa = 75^\circ \) geometry are closer together and larger, which is suspected to be the cause of the higher mixing in the diffuser shown in the contours of Figure 10 compared to the \( \kappa = 90^\circ \) geometry in Figure 7.

Figure 17 (c) shows the normalized vorticity for the \( \kappa = 100^\circ \) hole. Relative to the two other hole geometries, the vortical structures in the upper corners of the diffuser are spread apart and thus interact less with each other. We conjecture that the enhanced flow attachment of the \( \kappa = 100^\circ \) geometry is because it Diffuses the high velocity flow located on the leading-edge side of the diffuser more than the low velocity flow on the trailing edge of the diffuser shown in Figure 16 (c). Note that the \( \kappa = 100^\circ \) cross section shown in Figure 4(c) has a larger effective expansion near the leading-edge side of the diffuser compared to the nominal 10° expansion at the meter centerline. From the results seen in Figure 12, it is expected that this bias toward the leading edge of the diffuser from the \( \kappa = 100^\circ \) geometry created the high sensitivity to blowing ratio in co-flow.

4.5 Comparison of Hole Experimental Results

The lateral averages of the \( \kappa = 75^\circ \), \( \kappa = 90^\circ \), and \( \kappa = 100^\circ \) geometries in co-flow and counterflow at \( M = 1 \) is shown in Figure 18. The performance of the \( \kappa = 75^\circ \) and \( \kappa = 90^\circ \) film cooling holes are comparable to each other, producing similar lateral average effectiveness. The narrow footprint \( \kappa = 100^\circ \) geometry does slightly outperform wider \( \kappa = 75^\circ \) and \( \kappa = 90^\circ \) geometries in co-flow and counterflow due to the increased length of cooling. However, the shape of the diffuser does not significantly impact the performance at low blowing ratios.

As shown in the contours of the \( \kappa = 75^\circ \), \( \kappa = 90^\circ \), and \( \kappa = 100^\circ \) film cooling holes in Figure 19, sidewall separation inside the diffuser decreases with increasing diffuser edge angle. However, differences in performance due to the decrease in separation are not significant at a blowing ratio of \( M = 1 \). At the exit of the diffuser, the co-flow geometries appear to have higher areas of jet-to-jet interaction between film cooling holes compared to their counterflow counterparts. However, as shown by the lateral average effectiveness in Figure 18, the difference in performance due to the jet-to-jet interaction at the exit is small.

The laterally averaged adiabatic effectiveness at a constant blowing ratio of \( M = 3 \) is shown in Figure 20. The \( \kappa = 90^\circ \) geometry has the best performance overall at high blowing ratios, with little change in performance in either co-flow or counterflow. Both narrow and wide film cooling holes provide worse performance than the \( \kappa = 90^\circ \) in co-flow. In counterflow, the narrow and wide film cooling holes provide similar lateral average performance. This performance is attributed to the \( \kappa = 75^\circ \) geometry wide contours providing similar cooling as the longer \( \kappa = 100^\circ \) contours.
Figure 20. Laterally averaged adiabatic effectiveness for all geometries in co-flow and counterflow, at $M=3$.

Figure 21 shows the contours of all geometries at a blowing ratio of three in co-flow and counterflow. As expected by the lateral average plot in Figure 20, the performance of the $\kappa=75^\circ$ and $\kappa=90^\circ$ geometries are not significantly impacted by changes in coolant feed. As previously discussed, the $\kappa=100^\circ$ is unique in which the counterflow case has a significant improvement towards adiabatic effectiveness compared to the co-flow case. Because the film cooling holes are subject to the same blowing ratio and density ratio, the mass flux through each geometry shown in Figure 21 is constant. Thus, the lower performance of the $\kappa=100^\circ$ in co-flow is attributed to the velocity distribution it creates, but further investigation of the flow field inside the diffuser will be necessary. It is important to remember that the only differences between hole geometry in this study is the diffuser edge angle, diffuser interior edge radius, and breakout width, with two of the features occurring at the diffuser outlet. Thus, from Figure 20, it is shown that 10-10-10 film cooling holes are sensitive to changes in diffuser shape at high blowing ratios in co-flow, but not in counterflow.

4.6 Diffuser Roughness Effects

Figure 22 shows the lateral average adiabatic effectiveness, normalized to the breakout width to pitch ratio, for the $\kappa=90^\circ$ smooth, slightly rough, and rough geometry defined by Table 3 at a blowing ratio of one and three in co-flow. Normalized lateral average data from Schroeder and Thole [13] for smooth and rough plenum fed $\kappa=90^\circ$ 7-7-7 film cooling holes are also included.

As shown in Figure 22, the performance of the $\kappa=90^\circ$ geometry is not impacted at low blowing ratios by a slightly rough diffuser. The rough diffuser case only decreases the film cooling hole performance for $x/D<14$. The decrease in near hole performance is a result of the increase in sidewall separation inside the diffuser due to the roughness narrowing the cooling jet on the surface. Performance at $x/D\geq14$ is unaffected by roughness. At high blowing ratios, slightly rough and rough film cooling holes provide similar performance. Both rough diffusers induce more sidewall separation. Combining this with the increased momentum from a high blowing ratio, the jet detaches from the surface near the film cooling hole, resulting in poor performance. The increase in adiabatic effectiveness at an $x/D=11$ for the rough 10-10-10 $\kappa=90^\circ$ is a result of the low hole to hole pitch. The low pitch causes the downstream contours of cooling from adjacent holes to merge together, resulting in a higher laterally averaged adiabatic effectiveness.
A comparison of laterally averaged adiabatic effectiveness with changes in roughness and hole geometry in co-flow at \( M=1 \) is shown in Figure 23. As shown in the figure, the \( \kappa=100^\circ \) geometry is insensitive to roughness at low blowing ratios. In addition, the \( \kappa=75^\circ \) geometry follows the same trend as the \( \kappa=90^\circ \) geometry, with both geometries having the same effectiveness with smooth and rough diffusers at an \( x/D \geq 15 \). The 10-10-10 \( \kappa=75^\circ \) and \( \kappa=90^\circ \) holes reduce their effectiveness due to roughness only for \( x/D < 10 \). For \( x/D \geq 15 \), the larger lateral expansion angles promote an insensitivity to roughness, likely due to the higher sidewall separation already present in 10-10-10 holes. Counterflow feed was also studied, however the results match the trends seen in Figure 23.

Figure 24 shows the same co-flow fed cases as Figure 23, but at a blowing ratio of \( M=3 \). At this higher blowing ratio, the \( \kappa=75^\circ \) and \( \kappa=100^\circ \) geometries are severely impacted by roughness, providing a trivial amount of cooling. When normalized by the \( t/P \) values in Table 2, the \( \kappa=75^\circ \) and \( \kappa=100^\circ \) geometries perform similar to each other. Not included in this study is the counterflow results, which follow the exact same roughness trends as the co-flow cases shown in Figure 24. Thus, roughness overshadows any effects that wide or narrow 10-10-10 film cooling holes cause at high blowing ratios. In practice, this means that for manufacturing, the accuracy to the design intent for the diffuser geometry is secondary to the smoothness of the geometry for film cooling performance.

5. CONCLUSIONS

This study presented scaled experimental measurements of adiabatic film cooling effectiveness for 10-10-10 shaped film cooling holes with different diffuser cross sections, as well as varying levels of roughness in the diffuser. The diffuser edge angle and diffuser interior edge radius were modified to produce holes with trapezoidally-shaped diffuser cross sections that had the same area ratio as the baseline rectangular diffuser cross section. The diffuser roughness of as-manufactured holes was measured and simulated in the experiment using sand of appropriate roughness to the meter diameter scale and adhered to the diffuser. The holes were fed from a channel that could be operated with the flow direction aligned with the mainstream (co-flow), or opposite to the mainstream (counterflow), to simulate the method of coolant delivery inside of a turbine vane.

From the results, it has been shown that 10-10-10 film cooling holes can vary in performance, even at constant area ratio, due to changes in the cross-section shape of the diffuser exit. At low blowing ratios, changes in geometry and film cooling hole feed do not impact performance significantly to warrant concerns in film cooling hole performance in both co-flow and counterflow. At high blowing ratios and in co-flow, changes in the diffuser geometry can cause significant changes in performance. Counterflow provides similar performance from \( \kappa=75^\circ \) to \( \kappa=100^\circ \). For manufacturing purposes, variances in film cooling hole shape are not significant to cooling performance except for co-flow fed film cooling holes.

With regards to a film cooling hole design compared to the standard \( \kappa=90^\circ \), it was found that reducing the diffuser edge angle to \( \kappa=75^\circ \) will reduce sensitivity to the film cooling hole feed at the cost of cooling performance at higher blowing ratios. In contrast to this, increasing the diffuser edge angle to \( \kappa=100^\circ \) will increase sensitivity to the film cooling hole feed, with slight increases to performance at low blowing ratios. Overall, the standard \( \kappa=90^\circ \) still provides the best performance with no significant sensitivities. Therefore, if a manufacturing technique is unable to guarantee \( \kappa=90^\circ \) diffuser walls, then the tolerance should be biased towards a \( \kappa=75^\circ \) film cooling hole to minimize sensitivity of parameters on adiabatic effectiveness.

At a low blowing ratio, the 10-10-10 geometries are not significantly impacted by roughness. This indicates that film cooling hole manufacturing does not need to focus on off design
characteristics of film cooling holes at low blowing ratios. For high blowing ratios, changing the diffuser edge angle from the $\kappa=90^\circ$ standard increases overall sensitivity to roughness, with both $\kappa=75^\circ$ and $\kappa=100^\circ$ holes significantly reducing their performance with rough diffusers. Thus, from a film cooling hole manufacturing perspective, it is recommended that for the best performance, the manufacturing of film cooling holes should focus on creating the smoothest diffuser geometry possible.

From all the variables tested in this study, film cooling hole performance is most impacted by diffuser roughness, then by blowing ratio, diffuser edge angle, film cooling hole feed direction, and finally channel velocity ratio. From the results, it can be concluded that the diffuser edge angle primarily affects the performance of manufactured film cooling holes most. Thus, manufacturing methods of film cooling holes should be evaluated primary based on their ability to produce smooth geometry. Finally, geometry variances in the diffuser, besides expansion angle and area ratio, can produce variances in performance, and further study into the diffuser interior edge radius is recommended.

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