INFLUENCE OF IN-HOLE ROUGHNESS AND HIGH FREESTREAM TURBULENCE ON FILM COOLING FROM A SHAPED HOLE

A Dissertation in
Mechanical Engineering

by

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Submitted in Partial Fulfillment of the Requirements for the Degree of

Doctor of Philosophy

December 2015
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Abstract

Gas turbines are heavily used for electricity generation and aircraft propulsion with a strong desire in both uses to maximize thermal efficiency while maintaining reasonable power output. As a consequence, gas turbines run at high turbine inlet temperatures that require sophisticated cooling technologies to ensure survival of turbine components. One such technology is film cooling with shaped holes, where air is withdrawn from latter stages of the compressor, is bypassed around the combustor, and is eventually ejected out holes in turbine component surfaces. Air ejected from these shaped holes helps maintain components at temperatures lower than flow from the combustor. Many studies have investigated different factors that influence shaped hole performance. However, no studies in open literature have investigated how cooling performance is affected by roughness along interior walls of the shaped hole. The effect of in-hole roughness on shaped hole film cooling was the focus of this research.

Investigation of in-hole roughness effects first required the determination of behavior for a shaped hole with smooth walls. A public shaped hole, now used by other investigators as well, was designed with a diffused outlet having 7° expansion angles and an area ratio of 2.5. At low freestream turbulence intensity of 0.5%, film cooling adiabatic effectiveness for this smooth hole was found to peak at a blowing ratio of 1.5. Measurements of flowfields and thermal fields revealed causes of this behavior. Blowing ratio increases above 1.5 caused the jet from the smooth hole to penetrate higher into the surrounding mainstream, exhibit a stronger counter-rotating vortex pair, and have narrower contact with the wall than at lower blowing ratios. Experiments performed at high freestream turbulence intensity of 13% revealed dynamics of how freestream turbulence both diluted and laterally spread coolant. At the high blowing ratio of 3 the dilution and spreading were competing effects, such that elevated freestream turbulence did not cause a decrease in area-averaged effectiveness. At the blowing ratio of 1.5, high freestream turbulence caused area-averaged effectiveness to decrease 17% relative to the low freestream turbulence case.

Film cooling performance was measured for the shaped hole geometry with several different configurations of in-hole roughness. At low freestream turbulence intensity, in-
hole roughness caused decreases in area-averaged adiabatic effectiveness up to 61% relative to the smooth hole performance. These percent decreases in adiabatic effectiveness were more severe with increasing roughness levels and with increasing blowing ratios. Flowfield and thermal field measurements for the configuration with largest roughness size showed that the decrease in adiabatic effectiveness for rough holes as compared to smooth holes was due to thicker boundary layers along the interior walls of the cooling holes. The thicker boundary layers resulted in faster jet core flow, which in turn caused increased penetration of coolant into the mainstream and increased turbulence intensity inside the jet, with both leading to reduced adiabatic effectiveness. Detrimental effects of in-hole roughness persisted at the high freestream turbulence conditions as well.
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<th>Definition</th>
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<tbody>
<tr>
<td>A</td>
<td>hole cross-sectional area</td>
</tr>
<tr>
<td>AR</td>
<td>area ratio, $A_{exit}/A_{inlet}$</td>
</tr>
<tr>
<td>b</td>
<td>diameter of turbulence grid bars</td>
</tr>
<tr>
<td>$c_f$</td>
<td>skin friction coefficient, measured experimentally</td>
</tr>
<tr>
<td>$c_{f,0}$</td>
<td>flat plate correlation $c_f$, $0.036 \cdot \text{Re}<em>{\delta_2}^{-0.3}$ (for $\text{Re}</em>{\delta_2} &lt; 3000$) [1]</td>
</tr>
<tr>
<td>CA</td>
<td>compound angle (zero for experiments of this dissertation)</td>
</tr>
<tr>
<td>CFD</td>
<td>computational fluid dynamics</td>
</tr>
<tr>
<td>CRVP</td>
<td>counter-rotating vortex pair</td>
</tr>
<tr>
<td>$d_p$</td>
<td>diameter of seeding particle</td>
</tr>
<tr>
<td>D</td>
<td>diameter of film cooling holes</td>
</tr>
<tr>
<td>DR</td>
<td>density ratio, $\rho_c/\rho_\infty$</td>
</tr>
<tr>
<td>h</td>
<td>heat transfer coefficient</td>
</tr>
<tr>
<td>H</td>
<td>boundary layer shape factor</td>
</tr>
<tr>
<td>I</td>
<td>momentum flux ratio, $\rho_c U_c^2/\rho_\infty U_\infty^2$</td>
</tr>
<tr>
<td>k</td>
<td>thermal conductivity; ratio of specific heats</td>
</tr>
<tr>
<td>L</td>
<td>hole length</td>
</tr>
<tr>
<td>LDV</td>
<td>laser Doppler velocimetry</td>
</tr>
<tr>
<td>$\dot{m}_c$</td>
<td>coolant mass flow rate</td>
</tr>
<tr>
<td>M</td>
<td>blowing ratio, $\rho_c U_c/\rho_\infty U_\infty = (\dot{m}<em>c/A_c)/\rho</em>\infty U_\infty$</td>
</tr>
<tr>
<td>p</td>
<td>pressure</td>
</tr>
<tr>
<td>P</td>
<td>lateral distance between holes, pitch</td>
</tr>
<tr>
<td>PIV</td>
<td>particle image velocimetry</td>
</tr>
<tr>
<td>$q''$</td>
<td>heat flux</td>
</tr>
<tr>
<td>R</td>
<td>radius of diffused outlet interior edges</td>
</tr>
<tr>
<td>$R_a$</td>
<td>centerline average roughness height, the mean of absolute value deviations from the average surface height</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number ($\text{Re}<em>{\delta_2} = \delta_2 \cdot U</em>\infty/\nu$, $\text{Re}^* = \delta \cdot u_\tau/\nu_c$)</td>
</tr>
<tr>
<td>s</td>
<td>equivalent slot width based on metering area, $A_{inlet}/P$</td>
</tr>
<tr>
<td>Stk</td>
<td>Stokes number, $\rho_p d_p^2 U_c/18 \rho_\nu D = (\rho d_p^2 / 18 \rho_\nu) / (D/U_c)$</td>
</tr>
<tr>
<td>t</td>
<td>hole breakout width, per Figure 1.4</td>
</tr>
<tr>
<td>T</td>
<td>temperature</td>
</tr>
<tr>
<td>$T_{u_\infty}$</td>
<td>freestream turbulence intensity, $\sqrt{(u''_x^2 + v''<em>y^2) / 2}$ / $U</em>\infty$</td>
</tr>
<tr>
<td>$u_t$</td>
<td>friction velocity, $U_\infty \sqrt{c_f/2}$</td>
</tr>
<tr>
<td>$U_c$</td>
<td>coolant area-average velocity in metering section</td>
</tr>
<tr>
<td>$U_\infty$</td>
<td>mainstream mean velocity</td>
</tr>
<tr>
<td>u,v,w</td>
<td>$x$-, $y$-, and $z$-velocities</td>
</tr>
<tr>
<td>$\bar{u}'v'$</td>
<td>streamwise-vertical component of turbulent shear stress</td>
</tr>
<tr>
<td>$\bar{u}'w'$</td>
<td>streamwise-lateral component of turbulent shear stress</td>
</tr>
<tr>
<td>$v'w'$</td>
<td>vertical-lateral component of turbulent shear stress</td>
</tr>
<tr>
<td>VR</td>
<td>velocity ratio, $U_c/U_\infty$</td>
</tr>
<tr>
<td>x,y,z</td>
<td>position measured from origin at hole centerline breakout</td>
</tr>
<tr>
<td><strong>Greek</strong></td>
<td>Description</td>
</tr>
<tr>
<td>-----------</td>
<td>-------------</td>
</tr>
<tr>
<td>α</td>
<td>hole injection angle</td>
</tr>
<tr>
<td>β</td>
<td>expansion angle for diffused outlet</td>
</tr>
<tr>
<td>γ</td>
<td>effective injection angle, $α - β_{fwd}$</td>
</tr>
<tr>
<td>δ</td>
<td>99% boundary layer thickness</td>
</tr>
<tr>
<td>$δ_2$</td>
<td>boundary layer momentum thickness</td>
</tr>
<tr>
<td>η</td>
<td>local adiabatic effectiveness, $(T_{∞} - T_{aw})/(T_{∞} - T_{c})$</td>
</tr>
<tr>
<td>ηₚ</td>
<td>heat engine thermal efficiency</td>
</tr>
<tr>
<td>θ</td>
<td>non-dimensional fluid temperature, $(T_{∞} - T)/(T_{∞} - T_{c})$</td>
</tr>
<tr>
<td>Λₓ</td>
<td>integral length scale of freestream turbulence</td>
</tr>
<tr>
<td>υ</td>
<td>kinematic viscosity</td>
</tr>
<tr>
<td>ρ</td>
<td>density</td>
</tr>
<tr>
<td>φ</td>
<td>mean velocity penetration angle, $\arctan(V/U)$</td>
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<table>
<thead>
<tr>
<th><strong>Subscripts</strong></th>
<th>Description</th>
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<tbody>
<tr>
<td>aw</td>
<td>adiabatic wall</td>
</tr>
<tr>
<td>c</td>
<td>coolant, at hole inlet</td>
</tr>
<tr>
<td>f</td>
<td>with film cooling</td>
</tr>
<tr>
<td>CL</td>
<td>centerline</td>
</tr>
<tr>
<td>eff</td>
<td>effective, based on area at hole exit plane</td>
</tr>
<tr>
<td>exit</td>
<td>exit plane of the film cooling hole, per Figure 1.4</td>
</tr>
<tr>
<td>fwd</td>
<td>forward expansion of shaped hole</td>
</tr>
<tr>
<td>inlet</td>
<td>inlet plane of the film cooling hole, per Figure 1.4</td>
</tr>
<tr>
<td>lat</td>
<td>lateral expansion of shaped hole (half-angle)</td>
</tr>
<tr>
<td>m</td>
<td>metering section</td>
</tr>
<tr>
<td>p</td>
<td>seeding particle (DEHS droplet)</td>
</tr>
<tr>
<td>w</td>
<td>wall</td>
</tr>
<tr>
<td>$δ_2$</td>
<td>based on momentum thickness</td>
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<tr>
<td>∞</td>
<td>mainstream</td>
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<tr>
<th><strong>Superscripts</strong></th>
<th>Description</th>
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<tbody>
<tr>
<td>′</td>
<td>fluctuating/rms value</td>
</tr>
<tr>
<td>−</td>
<td>laterally-averaged (except for $\bar{u}'\bar{v}'$, $\bar{u}'\bar{w}'$, and $\bar{v}'\bar{w}'$)</td>
</tr>
<tr>
<td>=</td>
<td>area-averaged</td>
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Acknowledgements

First thanks go to my advisor Dr. Karen Thole. I am thankful for how she provided exciting industry-oriented research work and cultivated an atmosphere of camaraderie and excellence in the lab. Her close involvement and foresight with our lab’s research has been of immense benefit, and is astounding considering her other job responsibilities. I am also thankful for her understanding and support when issues arose outside work. Karen, you are my role model for professionalism and technical excellence. Thank you so much for everything.

I am thankful to other members of my committee: Dr. Yavuzkurt, Dr. Lynch, Dr. Craven, and Dr. Heidmann. Thank you for your overall thoughts, constructive criticism, and service on my doctoral committee. I am also grateful to the NASA Aeronautics Scholarship Program for funding of my research.

There are many others I am indebted to. Michael Alley, thank you for teaching us to be better writers and presenters. I can already see how those communication skills will serve us through our whole careers. Phil Irwin, thanks for being the best machinist ever and cutting perfect smooth and rough film cooling holes for my tests. Labmates and lab supervisors, past and present—there are many of you now that we have two labs—thanks for providing helpful hands or kind words when it counted. I must acknowledge my fellow researchers in the “Little Blue” tunnel: Molly Eberly, Chris Whitfield, and currently Shane Haydt. Thanks for all the help with setting up liquid nitrogen and executing tests.

I must thank those closest to me. To my beautiful wife Ariel: thanks for moving to State College before I did, agreeing to marry me, and standing steadfast by me during these graduate school years. A lot has happened. It is bittersweet leaving State College but I am excited for the next chapter to begin for our family of four in Wheaton! I love you Ariel.

I must give the most thanks to Jesus Christ, whose atoning death and resurrection is joyful news and the reason my life is worthwhile. I attribute my ability to finish this dissertation to him. I want to grow deeper in relationship with him.

To my extended family and Ariel’s, thank you for all your support and encouragement and frequent visits to State College. I am thankful too for the support from
our church communities at Good Shepherd Lutheran and at Calvary Baptist and from our other friends throughout Happy Valley.

This dissertation is dedicated to my Lord and Savior Jesus Christ who “upholds the universe by the word of his power” (Hebrews 1:3). May this dissertation, and all of my work, be to his glory.
Chapter 1.
Introduction

Due to their superior performance over that of cylindrical holes, shaped holes are frequently incorporated in cooling designs for gas turbines. Many studies have investigated the performance of shaped holes, including parametric studies that examined the influence of different geometry parameters on film cooling performance. However, to date there have been no studies examining the influence of roughness occurring on interior surfaces of shaped holes. Most shaped hole studies are performed with holes having smooth interior walls, which is not always realistic of cooling holes in gas turbine parts. Also, many studies are performed at low freestream turbulence intensity that is not necessarily representative of the engine environment. The focus of this dissertation is the investigation of in-hole roughness effects on shaped hole performance, both at low and at high freestream turbulence intensities.

1.1 Background

Gas turbines are widely used to deliver shaft power for propulsion and for electricity generation. Modern jet engines typically have turbofan or turboprop configurations where the gas turbine provides a high-speed propulsive jet but also provides shaft power that turns a ducted fan or an open rotor. This fan or rotor delivers much of the aircraft thrust. Gas turbines also provide motive shaft power for ships, tanks, and occasionally locomotives (Bombardier JetTrain, 2002) and racecars (Indianapolis 500, 1967-68). Electricity power plants use shaft power from gas turbines to turn electric generators. Due to their adjustable power output, gas turbines are more frequently used in load-following and peaking power plants rather than in base-load electricity power plants.

The three main components of a gas turbine are the compressor, combustor, and turbine. These are illustrated in Figure 1.1 which is a half-axis schematic for a basic turboprop engine. Standard numbering is given for stations within the aero engine [2]. Air enters from left, flows through the compressor, is burned with fuel in the combustor, and is then expanded through the turbine which provides power to the shaft.
The basic thermodynamic cycle for gas turbines is the Brayton cycle where, in the ideal case, heat addition in the combustor is modeled as a constant-pressure process. For turboprop configurations, gases exit the turbine (station 5) near ambient pressure since exhaust gases are not used to provide appreciable thrust. Almost all turboprop thrust instead comes from net power delivered to the shaft which turns the open rotor (propellers). The ideal thermodynamic cycle is illustrated in Figure 1.2 by loop 0-3-4-5-0, following the station numberings of Figure 1.1.

For net power delivered to the shaft, thermodynamic analysis of the Brayton cycle using cold-air-standard assumptions yields an expression for thermal efficiency that is a function of pressure ratio \( (p_3/p_0) \) and specific heat ratio \( (k) \) alone, as shown by Equation 1.1.

\[
\eta_t = 1 - \frac{1}{(p_3/p_0)^{(k-1)/k}}
\]  

Higher thermal efficiencies are obtained with higher pressure ratios. However, power output per unit mass flow is proportional to the enclosed area in the T-s diagram, shaded for cycle 0-3-4-5-0 in Figure 1.2. As engine designers choose higher pressure ratios to achieve higher thermal efficiencies, they must also consider engine size for a given power output. Engine size is roughly proportional to mass flow through the gas turbine. To increase
thermal efficiency without necessitating a larger engine size, higher temperatures are required at the turbine inlet. Figure 1.2 illustrates this tradeoff with a second ideal cycle at higher pressure ratio than the first. Cycle 0-3′-4′-5′-0 has the same enclosed area as the first cycle, thereby maintaining power output per unit mass flow, but turbine inlet temperature is commensurately higher for the second cycle (T_4′) than for the first (T_4).

Modern gas turbines use technologies such as film cooling to enable turbine inlet temperatures far in excess of melting temperature of metal components. For reference, the highest turbine inlet temperature currently acknowledged is 1980 °C for the F135 engine [3]. Film cooling works by withdrawing high pressure air from the compressor, bypassing this air around the combustor, and supplying the air to cool airfoils and other hot turbine components. After this cooler flow circulates through internal passages in the components, it is ejected from discrete holes on vanes, blades, and endwalls exposed to the hot mainstream. Figure 1.3
shows an example of film cooling holes on the first turbine component downstream of the combustor, the nozzle guide vane. In a manner of speaking the coolant forms a protective “film” over component surfaces, shielding components from the hottest flow. In reality, coolant concentrations over each surface are non-uniform and depend on location of the individual cooling holes. Ideally the coolant remains attached to the surface and only slowly mixes with the hot mainstream, thereby helping maintain the surface at a temperature lower than the mainstream. As will be explained, shaped film cooling holes are more successful at achieving these behaviors than cylindrical holes.

Air used for internal and then film cooling enables higher T₄ temperatures, but not without cost. In modern gas turbines, up to 20-30% of compressor air is used for cooling [5]. Since this air could otherwise be burned with fuel to produce power, film cooling flows represent a loss on the gas turbine thermodynamic cycle. To minimize this loss, turbine designers continually seek higher cooling performance with less coolant mass flow.

1.2 Film Cooling Convection Model

Film cooling performance is primarily evaluated from a heat transfer perspective. In addition, aerodynamic losses are increasingly becoming a second consideration for film cooling performance, however aerodynamic losses were outside the scope of this dissertation. The extent to which film cooling is successful from a heat transfer standpoint is determined by two performance variables: the heat transfer coefficient and the adiabatic effectiveness. These two variables determine heat flux into the surface from external convection, given by Equation 1.2.

\[ q'_f = h_f \cdot (T_{aw} - T_w) \]  

Equation 1.2 comes from the mechanistic model for convective heat transfer, where heat flux is proportional to the product of a heat transfer coefficient \( h_f \) and a temperature difference driving the convection \( T_{aw} - T_w \). \( T_{aw} \) is the temperature of fluid immediately above the surface. This variable is called the adiabatic wall temperature because the surface attains this temperature if the convective heat flux is zero, occurring for an insulated wall. Both \( h_f \) and \( T_{aw} \) vary locally over each surface because the discrete nature of the coolant jets causes non-uniformity in the flowfield and distribution of coolant. By defining the driving
temperature as $T_{aw}$ rather than $T_\infty$, mixing of coolant and mainstream is accounted for and Equation 1.2 reflects the proportionality between temperature difference and local heat flux.

The external wall temperature ($T_w$) is not a performance variable, but rather is determined by the balance of heat flux into the surface from external convection (and radiation) and heat flux removed from the underside of the surface by internal cooling. The end purpose of internal cooling and film cooling is to keep $T_w$ within material limits.

The adiabatic wall temperature is presented in non-dimensional form as adiabatic effectiveness, consistent with similitude between experiments and real engine conditions:

$$\eta = \frac{T_\infty - T_{aw}}{T_\infty - T_c}$$  \hspace{1cm} (1.3)

Along with indicating the driving temperature for convection, adiabatic effectiveness reveals the “footprint” of coolant over a surface downstream of film cooling holes. In literature, adiabatic effectiveness is the most common metric used to evaluate film cooling performance. This dissertation quantifies film cooling performance with adiabatic effectiveness.

**1.3 Influences on Film Cooling Performance**

Film cooling behavior is governed by many non-dimensional parameters describing both the flow and the film cooling geometry. Flow parameters include the coolant-to-mainstream ratios of density, velocity, mass flux, and momentum flux as shown below. In gas turbine engines, typical density ratios are between 1.5 and 2.0. The mass flux ratio $M$ is referred to as the “blowing ratio.” Note that when any two of the ratios are specified, the other two are fixed:

$$\text{DR} = \frac{\rho_c}{\rho_\infty}, \quad \text{VR} = \frac{U_c}{U_\infty}, \quad M = \frac{\rho_c U_c}{\rho_\infty U_\infty}, \quad I = \frac{\rho_c U_c^2}{\rho_\infty U_\infty^2}$$  \hspace{1cm} (1.4)

These parameters have primary influence on the behavior of coolant jets from film cooling holes. With increasing coolant flowrates, jets have more thermal capacity to cool but also have more upward momentum that tends to carry coolant away from the surface. The detrimental effect of high coolant momentum has been observed by Sinha et al. [6] and Thole et al. [7] for cylindrical holes and by Heneka et al. [8] for shaped holes. Thole et al. [7] found that, for cylindrical-hole jets, behavior falls into different regimes with
momentum flux ratio, including a regime of I > 0.8 where jets fully detach from the surface.

Attributes of the mainstream flow and coolant flow approaching the holes also influences film cooling. Non-dimensional parameters describing the approach flows include, for example, the freestream turbulence intensity and integral length scale ($Tu_\infty$ and $\Lambda_x/D$) and the mainstream boundary layer momentum thickness ($\delta_2/D$). Note that lengths here are appropriately non-dimensionalized by hole metering diameter, D.

Film cooling performance can vary significantly between conditions of low and high freestream turbulence intensity. High freestream turbulence is present in leading stages of the turbine, due to the strong mixing flow in the combustor. Turbulence intensities routinely reach $Tu_\infty = 20\%$ and turbulence integral length scales can approach $\Lambda_x/D = 10$-$20$, due to large dilution jet holes in the combustor [9]. This elevated freestream turbulence causes greater dilution and dispersion of coolant jets from shaped holes, which leads to lower but more-uniform values of adiabatic effectiveness downstream of holes. Elevated freestream turbulence also generally increases heat transfer coefficients in film cooling and non-film cooling flows alike. Per Equation 1.2 the decrease in adiabatic effectiveness and the increase in heat transfer coefficient both contribute to worse film cooling performance, since they lead to higher heat flux into turbine components.

Geometric parameters relevant to the scope of this study are those of the film cooling hole itself. The simplest configuration is a row of cylindrical holes: constant hole diameter, inclined at an angle with respect to the surface ($\alpha$), having a hole length (L/D), a pitchwise spacing (P/D), and an orientation with respect to the mainstream direction (compound angle = CA). These parameters, along with a few additional parameters, are used to describe shaped holes as illustrated in Figure 1.4. Shaped holes have a diffused outlet with expansion angles ($\beta$) in the forward and/or lateral directions, beginning after a metering section of specified length ($L_m/D$). These geometric parameters for shaped holes lead to specific values of the exit-to-inlet area ratio (AR), coverage ratio ($t/P$), and effective injection angle from the floor of the diffused outlet ($\gamma = \alpha - \beta_{fwd}$).

The diffused outlet of shaped holes allows them to provide higher adiabatic effectiveness than cylindrical holes at high blowing ratios. With a diffused outlet, shaped holes have an exit-to-inlet area ratio greater than one, which causes jet velocity at the exit to be lower than at the hole inlet. Diffused outlets lower the momentum of coolant jets
relative to cylindrical holes, thereby reducing jet penetration into the mainstream and nearly precluding jet detachment [11]. Diffused outlets having lateral expansion exhibit a beneficial spreading of coolant over the surface. Forward expansion is also advantageous because it lowers the effective angle of coolant injection [12].

Multiple studies have investigated the influence of these “macro-geometry” parameters in shaped holes. Gritsch et al. [11] tested multiple parameters and found that large values of the P/D spacing had significant effect, with laterally-averaged effectiveness being inversely proportional to P/D for wide spacings. Heneka et al. [8] tested variations of shaped holes with sharp edges in the diffused outlet and found that shallower injection angles and higher area ratios gave higher effectiveness, especially at high blowing ratios. Saumweber and Schulz [12] tested fanshaped holes with different coolant approach flows and found similar trends to the above studies when a plenum condition existed at the hole inlet. By contrast, no studies in open literature have examined shaped hole performance as influenced by the “micro-geometry” parameter of in-hole roughness.
1.4 Flow Phenomena with Shaped Holes

Several flow phenomena have been identified with film cooling from shaped holes and are discussed here, beginning at the hole inlet. Pietrzyk et al. [13] and Thole et al. [14] found that when coolant entering the hole encounters a large turning angle a separation region develops. This separation region occurs on the leeward side of the hole entrance when film cooling holes are inclined in the direction of mainstream flow and coolant approaches from a plenum condition. The separation region is accompanied by higher coolant velocities ("jetting") on the opposite side of the hole entrance. Together, the separation region and jetting cause mean velocity gradients and turbulence generation inside the hole [15, 16]. Longer metering sections have been found to decrease the size of the separation region as noted in a shaped hole CFD study by Kohli and Thole [17].

The large turning angle imposed by the inclined hole entrance also causes a counter-rotating vortex pair (CRVP) inside the hole. For plenum entrance flow into inclined holes as noted above, orientation of the CRVP is such that secondary flow along the hole circumference brings fluid from the windward end to the leeward end of the hole circumference. Completing the circuits, secondary flow in the middle of the hole therefore brings fluid from the leeward end back to the windward end. Such in-hole CRVPs have been observed in computational studies with inclined cylindrical holes by Leylek and Zerkle [15] and Lemmon et al. [18]. Experimental work by Peterson and Plesniak [19] showed that CRVPs can also occur in cylindrical holes normal to the wall having crossflow at the entrance. In fact, entrance crossflow has strong influence on the presence and orientation of CRVPs for all hole inclinations [17, 19, 20]. The present dissertation focuses on shaped hole performance with a plenum inlet condition (no entrance crossflow).

After the metering section, coolant passes through the diffused outlet and exits the hole. The intended behavior in the diffused outlet is gradual and uniform slowing of the coolant. However, separation can occur inside the diffused outlet if the expansion angles are too large, as has been observed by Thole et al. [16], Saumweber and Schulz [12], and Kohli and Bogard [21]. This separation is usually detrimental to adiabatic effectiveness because it inhibits the reduction in jet velocity and jet momentum. The separation region often occurs at the floor of the diffused outlet, as has been observed computationally by Kohli and Thole [20] and by Saumweber and Schulz [22]. Multiple researchers have
observed bimodal patterns of adiabatic effectiveness due to this separation bubble [12, 23]. Saumweber and Schulz [12] found that, for fanshaped holes (no forward expansion), larger expansion angles created a larger separation bubble. In rare cases, separation in the diffused outlet can conversely lead to increased adiabatic effectiveness by causing windward-side jetting that shields the hole from ingestion of the mainstream [17].

The coolant jet outside the film cooling hole also contains a CRVP, distinct from the CRVP inside the hole. This external CRVP has the same sense of rotation as noted above for the in-hole CRVP of plenum-fed inclined film cooling holes. However, the external CRVP is not necessarily a result of the in-hole CRVP. In a computational case with slip walls and no in-hole CRVP, Lemmon et al. [18] saw that the external CRVP still formed downstream of the hole exit due to the shear layer between the coolant jet and the mainstream. Moreover, Peterson and Plesniak [19] observed formation of an external CRVP of the above-stated orientation even when the in-hole CRVP had opposite orientation. These evaluations of in-hole and external CRVPs were for cylindrical holes, but as noted by Heidmann and Ekkad [24] shaped holes can also have an external CRVP, albeit one with lesser strength due to the diffused outlet.

Coolant exiting from shaped holes typically stays attached to the surface [25], so the primary influence on adiabatic effectiveness performance is the external CRVP. The external CRVP is detrimental to film cooling because it promotes mixing between the coolant and mainstream, and because its rotation draws the hot mainstream down to the surface between coolant jets. The external CRVP deforms the perimeter of the coolant jet into a kidney shape and thus this CRVP is sometimes called the “kidney vortices.” Diffused outlets of shaped holes help diminish the external CRVP by reducing the coolant velocity [24] and spreading apart the vortex pair [26]. Haven et al. [26] noted that the external CRVP is an always-present time-mean vortex pair, but other unsteady vortex pairs are periodically shed from the hole outlet. The unsteady “anti-kidney” vortex pair is an example; having rotation opposite to the external CRVP it can actually reduce the strength of the external CRVP [26].

The velocity profile of coolant exiting the shaped hole also influences adiabatic effectiveness. At low blowing ratios the mainstream can push coolant so that it exits at the leeward side of the hole. This leads to mainstream ingestion into the windward side of the
hole, which dilutes coolant before it exits the hole and correspondingly reduces adiabatic effectiveness [16]. In addition, non-uniform mean velocity profiles are generally detrimental because they promote generation of turbulence that mixes out coolant with the mainstream.

1.5 Objectives and Uniqueness of Research

The primary objective of this dissertation is to quantify and understand how, at engine-relevant conditions, the performance of shaped holes is altered by roughness occurring along hole interior walls. Adiabatic effectiveness measurements are used to quantify film cooling performance, while flowfield and thermal field measurements are used to identify the physical mechanisms by which in-hole roughness influences the film cooling flow. To maximize applicability of this dissertation, results are presented for a public baseline shaped hole that is representative of both proprietary and open literature hole designs. Both smooth and several rough versions of the hole are tested and the roughness size is quantified for each case. Results are presented for multiple turbulence levels: from low freestream turbulence intensity, $T_u = 0.5\%$, characteristic of the majority of previous film cooling studies; up to high freestream turbulence intensity, $T_u = 13\%$, characteristic of turbulence occurring at the turbine inlet in real engines.

A secondary objective of this dissertation is to provide comprehensive documentation of performance of the baseline shaped hole, through measurements of adiabatic effectiveness, flowfield, and thermal field. All results presented are for a row of baseline shaped holes on a flat plate, providing higher spatial resolution than typically possible with film cooling studies on vanes, blades, or endwalls. The baseline shaped hole data of this dissertation establishes a standard of performance to which other novel cooling geometries may be compared. The data may also be used for validation of CFD and benchmarking of experimental facilities for film cooling. The data at high freestream turbulence intensity is especially useful since few studies report film cooling performance at intensities above $T_u = 10\%$. Thermal field data at high freestream turbulence intensity is especially rare; this dissertation contributes the first such measurements in open literature for shaped holes.
While new benchmarking data is indeed introduced, the primary contribution this dissertation makes to the state-of-the-art is its investigation of in-hole roughness effects. Past film cooling studies with surface roughness have focused on the influence of roughness outside the film cooling hole, such as roughness on the external film-cooled surface [27-29]. Currently, open literature contains no studies examining the influence of roughness occurring inside shaped cooling holes.

1.6 Outline of Dissertation

This dissertation is organized in manuscript format. First, performance of the baseline shaped hole with smooth in-hole surfaces is characterized in Chapters 2 through 4. Chapter 2, presented at the IGTI Turbo Expo 2014 conference, establishes the public shaped hole geometry as a baseline hole design and reports performance as measured by adiabatic effectiveness. Chapter 3, in review for publication in the Journal of Turbomachinery, uses measured flowfields to illuminate phenomena driving the shaped hole performance. Flowfields are presented for both low and high freestream turbulence intensity and are compared. Chapter 4 further investigates shaped hole performance by comparing measured thermal fields to flowfields, both at low and at high freestream turbulence intensity. Mechanisms causing increased dilution and spreading with the high freestream turbulence are identified. Then, in Chapter 5, results are presented for a variety of in-hole roughness configurations in the baseline shaped hole. The performance change due to in-hole roughness is quantified by comparison of adiabatic effectiveness. Physical mechanisms driving the performance change are identified through flowfields and thermal fields. Measurements with high freestream turbulence intensity show that in-hole roughness is influential even at this condition. Both Chapters 4 and 5 represent work to be submitted to Turbo Expo 2016 for consideration for the Journal of Turbomachinery. Chapter 6 of this dissertation presents the major conclusions and makes recommendations for future research.
Chapter 2.
Adiabatic Effectiveness Measurements for a Baseline Shaped Film Cooling Hole

Abstract

Film cooling on airfoils is a crucial cooling method as the gas turbine industry seeks higher turbine inlet temperatures. Shaped film cooling holes are widely used in many designs given the improved performance over that of cylindrical holes. Although there have been numerous studies of shaped holes, there is no established baseline shaped hole to which new cooling hole designs can be compared. The goal of this study is to offer the community a shaped hole design, representative of proprietary and open literature holes, that serves as a baseline for comparison purposes. The baseline shaped cooling hole design includes the following features: hole inclination angle of 30º with a 7º expansion in the forward and lateral directions; hole length of 6 diameters; hole exit-to-inlet area ratio of 2.5; and lateral hole spacing of 6 diameters. Adiabatic effectiveness was measured with this new shaped hole and was found to peak near a blowing ratio of 1.5 at density ratios of 1.2 and 1.5 as well as at both low and moderate freestream turbulence of 5%. Reductions in area-averaged effectiveness due to freestream turbulence at low blowing ratios were as high as 10%.

2.1 Introduction

Gas turbines utilize film cooling on hot gas path components to increase life by lowering metal temperatures. Cylindrical film cooling holes are the most economical to manufacture, but shaped holes have become widely used in military and commercial engines resulting from better cooling performance than cylindrical holes [25]. New novel cooling geometries are continually introduced as designers and researchers pursue better cooling performance. Cooling geometries that achieve higher cooling effectiveness with

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the same (or less) coolant are desirable because the compressor bleed air used for cooling can otherwise be used to produce power.

The impetus for the present study is the challenge in evaluating performance of any given cooling hole design. While there are many different shaped hole designs in literature, there are few geometries common between publications of different researchers. Most published film cooling studies compare novel cooling hole designs to the performance of a cylindrical hole, which is not necessarily the most helpful standard given the well-known jet detachment that occurs at high momentum flux ratios. The lack of one, single consensus baseline shaped hole currently limits the interpretation of cooling performance for new holes.

Designers and researchers would benefit from defining a baseline shaped hole geometry to be used instead of cylindrical holes for comparison purposes. This study proposes a baseline shaped hole and presents adiabatic effectiveness data for the hole geometry at low and high density ratios and low and moderate freestream turbulence intensities. The design of the proposed baseline shape was guided by shaped holes in public literature while being representative of proprietary shaped hole designs. Data for this baseline shaped hole will be useful for evaluation of novel cooling hole geometries, benchmarking other investigations, and validation of CFD studies. The hole design and adiabatic effectiveness data in this study are placed on a public website to share with the community (http://www.mne.psu.edu/psuturbine).

2.2 Review of Relevant Literature

A review of shaped hole geometries found in the literature identified 130 different hole designs. Geometries included conical, laidback, and fanshaped holes [8,11,12,21,30], and many novel designs: bean-shaped [31], cusp-shaped [32], crescent-shaped [33], cratered [34], “Console” [35], double-jet [36], Nekomimi [37], transonic wall jet [38], waist-shaped [39], arrowhead-shaped [40], and anti-vortex holes [24]. The review included not only flat-plate studies, but also endwall, airfoil, and test coupon studies that featured shaped holes [41-43]. Far more studies were examined than can be reported in this chapter, so a spreadsheet listing the studies and details of hole shapes is made available at the authors’ website [44] and is included in the Appendix. The hole shape most frequent in
literature was the laidback fanshaped, with over 50 variations identified for these laidback fanshaped holes.

Many of the shaped holes only differ from cylindrical holes by having a diffused outlet. The primary advantage of a diffused outlet is to decrease the momentum of the cooling jet at high flow rates relative to that of a cylindrical hole. By decreasing the momentum it reduces the jet penetration into the mainstream and reduces the likelihood of jet detachment [11].

Data for shaped film cooling holes were most recently tabulated and reviewed in the correlation developed by Colban et al. [45]. They found that effectiveness data collapsed better with blowing ratio (M) than with momentum flux ratio (I). Their correlation also accounted for pitchwise spacing (P/D), coverage ratio (t/P), and area ratio (AR). This correlation was successful in correlating laterally-averaged adiabatic effectiveness for shaped holes over a wide parameter range.

Shaped holes exhibit new flow phenomena in addition to that seen with cylindrical holes. Cooling jets from cylindrical holes detach from the surface at high momentum-flux ratios [7]. Jets from cylindrical holes also develop a counter-rotating vortex pair (CRVP, also called “kidney” vortices) that draws hot mainstream gas towards the surface [25]. With shaped holes, undesirable effects of the CRVP can be lessened through spreading apart the vortex pair and by formation of “anti-kidney” vortices [26]. Haven et al. [26] found that appearance of anti-kidney vortices depended on how the mainstream altered the windward side of the jet interface. Other flow phenomena includes a separation bubble that can form inside of the diffused outlet as observed by Saumweber and Schulz [12] causing a bimodal effectiveness pattern, which was also observed by auf dem Kampe et al. [23]. The separation inside the hole was result of expansion angles larger than 10° [12]. Large expansion angles in general cause jet separation inside the hole and thereby mainstream ingestion, as seen by Thole et al. [16], Kohli and Bogard [21], Saumweber and Schulz [22], and Lutum et al. [46].

At high coolant flow rates, different shaped holes can show different trends in cooling. For many shaped holes the effectiveness monotonically increases with blowing ratio and plateaus [45]. However, in other shaped holes effectiveness decreases as blowing ratio increases. Decreases have been observed to occur with holes having small expansion
angles from 0° to 10° [12] and ironically with holes having large expansion angles thought to cause in-hole jet separation [21, 47].

Freestream turbulence is known to significantly affect film cooling in gas turbines, where turbulence intensity can be $T_{u_{\infty}} = 20\%$ exiting the combustor [9]. Bons et al. [48] measured performance of closely-spaced cylindrical holes at four turbulence intensities up to $u'/U_{\infty} = 17\%$ and found that freestream turbulence was very effective in spreading coolant laterally. High freestream turbulence was observed to cause up to 70% reduction in centerline effectiveness at low momentum flux ratios. However, Bons et al. also found that at high momentum flux ratios with a detached jet, the change in centerline effectiveness from high freestream turbulence was negligible and effectiveness increased at the midpitch due to turbulence transporting coolant back onto the surface. Adiabatic effectiveness of shaped holes at moderate turbulence intensities up to 11% were measured by Saumweber et al. [49] and Saumweber and Schulz [22]. For shaped holes they observed no signs of jet detachment from the surface, and therefore freestream turbulence only acted to dilute the cooling and reduce adiabatic effectiveness. They saw a maximum decrease of 30% in laterally-averaged effectiveness at low blowing ratios, due to freestream turbulence intensity increasing from 3.6% to 11% [49]. Colban et al. [41] observed smaller reductions in effectiveness for shaped holes on a turbine endwall. When incoming freestream turbulence intensity was increased from 1.2% to 8.9%, area-averaged adiabatic effectiveness with shaped holes decreased an average of 6%.

In reviewing the film cooling literature, it was found that there was no standard shaped cooling hole for new holes to be compared with. Moreover, there are conflicting trends in the results for the various film cooling hole shapes. The hole proposed here is presented as a standard shaped hole for the community to compare with. This study is the first in a series that will be presented in which a full documentation of the surface cooling, flow and thermal fields, and various effects such as curvature, pressure gradients, and hole roughness will be presented.

### 2.3 Design of the Baseline Shaped Hole

The authors chose the laidback fanshaped hole as the geometry for the baseline shaped hole, due its predominance in the literature and use in industry. Table 2.1
summarizes the ranges for geometric parameters commonly found in the literature for laidback fanshaped holes. These geometric features served as a guide for designing the baseline shaped hole. In the literature there was a large variety in hole shapes. Interestingly many geometries had short metering sections that take up half or less of the total length of the shaped hole, which contradicts the suggested design [25] that the diffused outlet is confined to the “outer 20-50% of the wall thickness.”

The design chosen for the baseline shaped hole is described by the geometric parameters in the rightmost column of Table 2.1. This exact hole shape is illustrated in Figure 2.1 and is available for download from the authors’ website [44]. As illustrated at the bottom of Figure 2.1, the diffused outlet shape is driven by guidelines from the circular end of the metering section (plane A-A) to the filleted rectangle of plane B-B. The expansion angle in each direction was 7°, hence the name “7-7-7 shaped hole.” Lateral expansion angle is defined from the metering section axis as in Figure 2.1, with the half-angle being 7° and therefore the full-angle being 14°. The expansion angle of 7° is seen in literature with the research by Fawcett et al. [50] who had a 7° forward expansion, and in Gritsch et al. [11] who tested multiple shaped holes with 7° lateral expansion. The length of the metering section was chosen to be \( \frac{L_{mf}}{D} = 2.5 \) or 42% the total length of the hole, which is representative of shaped holes reviewed. The hole length and 7° expansion angles resulted in an area ratio \( AR = 2.5 \).
Overall the design choices differentiated the 7-7-7 baseline shaped hole from the geometry investigated extensively by Wittig and coworkers [16, 49, 51-54]. They investigated the performance of fanshaped and laidback fanshaped holes having 15° forward expansion and 14° lateral expansion (half-angle; verified to have been defined as in Figure 2.1). Expansion angles for the 7-7-7 shaped hole are about half those of Wittig, which helps ensure the baseline shaped hole performance will not be complicated by jet separation inside the hole [16, 22, 51] or bimodal effectiveness patterns [12]. Also different is that the forward expansion of the 7-7-7 shaped hole begins well towards the entrance of the hole, which allows forward diffusion prior to interaction with the mainstream. One commonality between the 7-7-7 shaped hole and those studied by Wittig is that in-hole edges of the diffused outlet are rounded to R/D = 0.5. For brevity, the 7-7-7 baseline shaped hole will henceforth in this chapter be referred to as the “shaped hole.”

2.4 Experimental Facility and Test Conditions

Adiabatic effectiveness measurements were acquired in a closed-loop wind tunnel shown in Figure 2.2 that was previously described by Eberly and Thole [55]. Mainstream
For the present study, the mainstream temperature was maintained at 295 K with a bank of electrical heating elements and a chilled water heat exchanger. As shown in Figure 2.2, the incoming boundary layer was removed with a suction loop at the entrance of the test section. A new boundary layer originated at the leading edge of the test plate and a trip wire at x/D = -33 initiated transition of the boundary layer to a turbulent state.

Coolant air for the film cooling injection was diverted from the mainstream using a variable frequency blower that was hermetically sealed. To avoid frost formation that can result from cryogenically cooling the coolant, the coolant was routed through a vent dryer containing solid desiccant. Downstream of the heat exchanger and prior to entering the plenum, the coolant flowrate was measured with a Venturi flow meter. Three flow conditioning screens inside the plenum were used to ensure uniformity of flow to the film cooling holes.

Film cooling holes were machined in Dow Styrofoam brand residential sheathing (k = 0.029 W/m·K) to ensure a nearly adiabatic surface for effectiveness measurement. The film cooling array was a row of five shaped holes with a metering diameter of 7.75 mm.
Adiabatic effectiveness measurements were determined from surface temperature measurements made with a FLIR SC620 infrared camera. The camera output was calibrated to accurately detect the entire range of temperatures measured, similar to as done by Eberly and Thole [55]. Coolant and freestream temperatures were each measured using multiple thermocouples. Adiabatic effectiveness measurements made by Eberly and Thole for cylindrical holes in this facility showed good agreement with the literature [55].

Uncertainty Analysis

An uncertainty analysis was performed for variables of density ratio, blowing ratio, and adiabatic effectiveness by propagating uncertainties using partial derivatives as outlined by Figliola and Beasley [56]. All calculations were done for a 95% confidence interval.

Uncertainty in density ratio was found to be low, being less than ±0.01 and ±0.02 for DR = 1.2 and 1.5 respectively. Blowing ratio uncertainty was found to be highest at the lowest blowing ratio of M = 0.5, for the worst case being ±10% (DR = 1.5, M = 0.5). The uncertainty in M = 0.5 was result of bias uncertainty in the Venturi flowmeter itself (±0.25% of full-scale flow and verified by separate tests with a laminar flow element in series). As coolant flow rate increased, percent uncertainty in M decreased because the inherent flowmeter uncertainty became less dominant. Maximum uncertainty for blowing ratios M = 2 and above was ±2.9%.

The uncertainty in adiabatic effectiveness was found to be greater at DR = 1.2 than at DR = 1.5 because of the smaller ΔT between the coolant and mainstream. Uncertainty in surface temperature (±0.9 °C for DR = 1.2, ±1.8 °C for DR = 1.5) was based on scatter in the infrared camera calibration data and bias uncertainty of the thermocouples used in the calibration, both of which became larger at lower temperatures. Uncertainty in coolant temperature accounted for variation between plenum thermocouples near the cooling holes. This variation was greatest at blowing ratio M = 0.5. Adiabatic effectiveness uncertainty was calculated to be δη = ±0.031 for DR = 1.2 and δη = ±0.024 for DR = 1.5. Repeatability was confirmed to within the uncertainty by repeating the measurements over a four month period with a separate foam specimen giving a maximum difference in laterally-averaged effectiveness of Δη̅ = 0.025, which was within the uncertainty range.
Test Matrix and Approach Flow Conditions

Table 2.2 summarizes the test conditions used for the adiabatic effectiveness measurements, which included low and moderate freestream turbulence. Freestream turbulence intensity ($Tu_\infty$) is reported 1.3D upstream of the hole, and blowing ratio and jet Reynolds number ($Re_D$) are evaluated at the metering section of the hole. The mainstream velocity was 10 m/s for all tests.

<table>
<thead>
<tr>
<th>$Tu_\infty$</th>
<th>DR</th>
<th>Blowing Ratios</th>
<th>$Re_D$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5%</td>
<td>1.2</td>
<td>$M = 0.5, 1, 2, 3$</td>
<td>2800 – 16900</td>
</tr>
<tr>
<td>0.5%</td>
<td>1.5</td>
<td>$M = 0.5, 1, 1.5, 2, 2.5, 3$</td>
<td>3400 – 20600</td>
</tr>
<tr>
<td>5.6%</td>
<td>1.5</td>
<td>$M = 0.5, 1, 2, 3$</td>
<td>3400 – 20600</td>
</tr>
</tbody>
</table>

Moderate freestream turbulence of $Tu_\infty = 5.6\%$ was obtained by installing a vertical-bar turbulence grid upstream of the flat plate leading edge at $x/D = -67$. The grid was composed of vertical round bars of diameter $b = 9.5$ mm, spaced 25.4 mm apart center-to-center. Laser Doppler Velocimetry (LDV) was used to characterize streamwise velocity in the turbulent freestream. Streamwise rms velocity fluctuations were $u'/U_\infty = 0.056$ at slightly upstream of the holes ($x/D = -2.3$) and decayed to $u'/U_\infty = 0.044$ by $x/D = 24$. This fluctuation level and its decay with streamwise distance agreed well with the parallel-rod correlation of Roach [57]. At the hole trailing edge for $Tu_\infty = 5.6\%$, the integral length scale was measured with single-component hot wire to be $\Lambda_x = 4.0D$.

The approach turbulent boundary layer was measured at $x/D = -2.3$ for both low and moderate freestream turbulence. Boundary layers were measured for at least five pitchwise locations and average values are presented in Table 2.3. Profiles of mean and fluctuating velocity at $z/D = 0$ are given in Figures 2.3 and 2.4, respectively.

<table>
<thead>
<tr>
<th>$Tu_\infty$</th>
<th>$\delta_2/D$</th>
<th>$H$</th>
<th>$Re_{62}$</th>
<th>$Re^*$</th>
<th>$u_t$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5%</td>
<td>0.14</td>
<td>1.45</td>
<td>670</td>
<td>315</td>
<td>0.5 m/s</td>
</tr>
<tr>
<td>5.6%</td>
<td>0.28</td>
<td>1.3</td>
<td>1380</td>
<td>420</td>
<td>0.5 m/s</td>
</tr>
</tbody>
</table>
Adiabatic Effectiveness at Low Freestream Turbulence Intensity

Contours of adiabatic effectiveness for DR = 1.2 at low freestream turbulence are shown in Figures 2.5a-d for three cooling holes in the center of the array. The contours show good periodicity between the three holes and, although it is not shown here, good periodicity was achieved with all five holes. Note that the values inside the holes are not given due to the camera focal location being on the flat plate surface.

At low blowing ratios, increases in the coolant flowrate from M = 0.5 to M = 1 resulted in increases in the adiabatic effectiveness. At flowrates above M = 1, Figures 2.5a-d show that coolant patterns on the surface began to narrow and continued to do so as the blowing ratio increased. Figure 2.6 gives the laterally-averaged effectiveness for the DR = 1.2 cases using the averaged values for the three holes shown in the contours. The highest effectiveness was at M = 1. Notice that far from the holes the decay rate was slower for the higher blowing ratios.

Figures 2.7a-d show adiabatic effectiveness contours for the same four blowing ratios at the higher density ratio of DR = 1.5. As was seen at DR = 1.2, effectiveness increased and then began to decrease with increasing blowing ratio resulting from the narrowing of the jet footprint. At blowing ratios above M = 1, the coolant footprint progressively narrowed with increases in blowing ratio. Centerline effectiveness values for
Figure 2.5. Shaped hole η contours for DR=1.2 and low freestream turbulence intensity of Tu∞ = 0.5%: (a) M=0.5, (b) M=1, (c) M=2, and (d) M=3.

Figure 2.6. Shaped hole laterally-averaged effectiveness at DR=1.2, low freestream turbulence intensity.
all DR = 1.5 cases are given in Figure 2.8a (note not all the contours were presented in Figures 2.7a-d for brevity). The results show how the behavior changed from low blowing ratio to high blowing ratio. At the trailing edge of the hole, centerline effectiveness increased with blowing ratio until plateauing at M = 1.5. The lower centerline effectiveness at the trailing edge for M = 0.5 and 1 can be explained by mainstream ingestion occurring when coolant had low momentum ($I_{eff} = 0.03$ and 0.11 respectively, calculated from $I_{eff} = I/AR^2$). Thole et al. [16] observed mainstream ingestion into a laidback fanshaped hole at $I_{eff} = 0.25$. Mainstream ingestion caused mixing that diluted the coolant inside the diffused outlet. At higher blowing ratios, coolant had more momentum and resisted mainstream ingestion. Laterally-averaged effectiveness shown in Figure 2.8b also indicates a change in behavior at M = 1.5. For x/D > 5, the highest laterally-averaged effectiveness occurred at M = 1.5 which also exhibited a slower decay in $\bar{\eta}$ than at M = 0.5 or 1. The steeper decay in effectiveness at low blowing ratios was attributed to mainstream ingestion. At blowing ratios greater than M = 1.5 the decay in effectiveness remained slow.

Laterally-averaged effectiveness was found to be higher at the high density ratio for the same blowing ratio, which resulted from an increased lateral spreading of the coolant. Figure 2.9 compares laterally-averaged effectiveness between DR = 1.2 and 1.5. In Figure 2.9 the solid symbols are the high density ratio data. Increased effectiveness at high density

---

**Figure 2.7.** Shaped hole $\eta$ contours for DR=1.5, low freestream turbulence intensity: (a) M=0.5, (b) M=1, (c) M=2, and (d) M=3.
Figure 2.8. Shaped hole adiabatic effectiveness at DR = 1.5, low freestream turbulence intensity: (a) centerline effectiveness and (b) laterally-averaged effectiveness.

Figure 2.9. Comparison of laterally-averaged effectiveness at DR = 1.2 and DR = 1.5, low freestream turbulence intensity.

ratio was associated with better lateral spreading of the coolant jet at high density ratio, consistent with observations by Eberly and Thole for cylindrical holes [55]. Figures 2.10a-b show lateral distributions of coolant for the center pitch at M = 1 and 3. At 5 diameters downstream of the hole trailing edge (x/D = 7.4, Figure 2.10a) the increased spreading at DR = 1.5 was only slightly apparent. Lateral spreading increased with downstream distance, as evidenced by lateral distributions at x/D = 32.4 (Figure 2.10b).
At both blowing ratios, the lateral spread of the coolant was greater at DR = 1.5 than at DR = 1.2. Note the vertical scale is reduced in Figure 2.10b to show detail.

Scaling of adiabatic effectiveness with blowing ratio and momentum flux ratio was investigated. Since effectiveness for cylindrical holes is known to scale with blowing ratio in the regime where jets do not detach from the surface [7], it is plausible that the attached jets from shaped holes would scale similarly. Effectiveness averaged over the three hole pitches over a streamwise distance between $5.4 \leq x/D \leq 37.4$ is shown in Figure 2.11 for both density ratios. (Predictions from the correlation of Colban et al. [45] are also shown and will be discussed in the following section.) At low blowing ratios, the DR = 1.2 and 1.5 results for the shaped hole indicated similar effectiveness. The $\bar{\eta}$ curves did not collapse, however, at blowing ratios $M \geq 2$ because of the better cooling achieved at DR = 1.5 compared with 1.2.

Figure 2.12 shows the same area-averaged effectiveness as a function of momentum flux ratio. The top abscissa in Figure 2.12 is the effective momentum flux ratio occurring at the hole exit, $I_{eff}$. At DR = 1.5 the peak effectiveness occurred at $I_{eff} = 0.2$. Above $I_{eff} = 0.2$, the area-averaged effectiveness did not collapse very well at the two density ratios. In summary, effectiveness for the shaped holes did not scale with blowing ratio or momentum flux ratio over the whole range evaluated.
Figure 2.11. Area-averaged effectiveness for shaped holes plotted as a function of the blowing ratio. Averaged over 3-35 diameters downstream of the hole trailing edge.

Figure 2.12. Area-averaged effectiveness for shaped holes plotted as a function of the regular and effective momentum flux ratios. Averaged over 3-35 diameters downstream of the shaped hole trailing edge.
Performance Relative to Other Geometries

Comparisons of the results were made to the shaped hole correlation of Colban et al. [45] developed for film cooling at density ratios between $1.7 \leq DR \leq 2$. Figure 2.13 compares laterally-averaged effectiveness measured at $DR = 1.5$ in the present study to the correlation. The shaped hole correlation underpredicted effectiveness for blowing ratios below $M = 2$. At $M = 2$, agreement between the correlation and the present study was relatively good. Higher blowing ratios are not plotted because they correspond to parameter values outside those used to develop the correlation. Adiabatic effectiveness predicted from the correlation was area-averaged and is given in Figures 2.11 and 2.12 as mentioned earlier. As seen, the correlation underpredicted the $DR = 1.5 \bar{\eta}$ at blowing ratios up to $M = 2$.

The disagreement between the present study and the correlation may be attributed to the 18 different shaped holes used to develop the correlation as shown in Table 2.4. Table 2.4 compares the shaped hole of the present study to three holes used in developing the correlation (Gritsch et al. [11], Colban et al. [45], and Saumweber et al. [49]). The shaped hole of the present study was at the low extreme of area ratios incorporated into the correlation, as only 3 of the 18 shaped holes had $AR = 2.5$ and none of the 18 had lower area ratios. Additionally, the majority of the shaped holes used to develop the correlation had diffused outlets much longer than in the present study. Most of the shaped holes used to develop the correlation were those presented by Gritsch et al. [11], comprising 13 of the 18 hole shapes used to derive the correlation [45]. All 13 of these shaped holes had diffused outlets at least 5.5D long. Longer diffused outlets lead to different exiting velocity profiles which in turn affects adiabatic effectiveness. Differences in the exact shape of the hole breakout may also have had influence. Haven et al. [26] found that breakout shape affects the counter-rotating vortex pair, since the leading edge of the breakout influences how the jet-mainstream interface is deformed. The schematic in Gritsch et al. [11] shows that their shaped hole breakouts had a bowed leading edge, as compared to the straight leading edge of the shaped hole for the present study (see Figure 2.1).

The shaped hole of the present study has expansion angles and an area ratio less-aggressive than several other shaped holes in literature, so a reasonable question is where
Figure 2.13. Laterally-averaged adiabatic effectiveness in the present study compared to predictions from the shaped hole correlation [45].

Table 2.4. Comparison of Shaped Holes

<table>
<thead>
<tr>
<th>Reference</th>
<th>$L_m/D$</th>
<th>$L_{lat}/D$</th>
<th>$L_{fwd}/D$</th>
<th>$\beta_{lat}$</th>
<th>$\beta_{fwd}$</th>
<th>R/D</th>
<th>$\alpha$</th>
<th>P/D</th>
<th>$t/P$</th>
<th>AR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Present Study</td>
<td>2.5</td>
<td>3.5</td>
<td>3.5</td>
<td>7°</td>
<td>7°</td>
<td>0.5</td>
<td>30°</td>
<td>6</td>
<td>0.35</td>
<td>2.5</td>
</tr>
<tr>
<td>Saumweber et al. [49]</td>
<td>2</td>
<td>4</td>
<td>1</td>
<td>14°</td>
<td>15°</td>
<td>0.5</td>
<td>30°</td>
<td>4</td>
<td>0.75</td>
<td>3.1</td>
</tr>
<tr>
<td>Colban et al. [45]</td>
<td>≥ 2*</td>
<td>4*</td>
<td>4*</td>
<td>10°</td>
<td>10°</td>
<td>0.5</td>
<td>30°</td>
<td>6.5</td>
<td>0.48</td>
<td>3.9</td>
</tr>
<tr>
<td>Gritsch et al. [11]</td>
<td>2</td>
<td>9.5</td>
<td>9.5</td>
<td>2°</td>
<td>4°</td>
<td>&gt;0.5*</td>
<td>30°</td>
<td>6</td>
<td>0.31</td>
<td>2.5</td>
</tr>
<tr>
<td>Heneka et al. [8]</td>
<td>2</td>
<td>3.5*</td>
<td>3.5*</td>
<td>10°</td>
<td>10°</td>
<td>0</td>
<td>35°</td>
<td>8</td>
<td>0.37</td>
<td>3.7</td>
</tr>
</tbody>
</table>

* -- value approximated from information provided in reference

performance falls with respect to other shaped holes. Figure 2.14 compares cooling performance in the present study to that of the shaped holes listed in Table 2.4, all of which are laidback fanshaped holes. The vertical axis gives area-averaged effectiveness taken over the range available from these studies, 5-22 diameters downstream of the shaped hole trailing edges. An exception is that Saumweber et al. [49] reported $\overline{\eta}$ averaged over 2-22 diameters downstream of their hole trailing edge. The geometries had pitchwise spacings varying from $P/D = 4$-8, but were made comparable by plotting as a function of a scaled blowing ratio as massflow per unit pitch. This variable is normalized by mainstream variables and the hole diameter as shown in Equation 2.1.

$$\left(\frac{\dot{m}_c}{P}\right) = \frac{\dot{m}_c/A_{inlet}}{\rho_\infty U_\infty} \cdot \frac{A_{inlet}/P}{D} = M \cdot \frac{s}{D}$$  (2.1)
Figure 2.14. Area-averaged effectiveness for shaped holes at high density ratio and low freestream turbulence intensity, plotted as a function of the coolant flowrate per pitch (Equation 2.1).

At low values of $M\cdot s/D$ Figure 2.14 shows that film cooling holes tend to the same performance regardless of shaped hole geometry differences. At high values, the shaped hole of Heneka et al. [8] performed worst among the shaped holes with their hole having the highest injection angle ($\alpha$). Heneka et al. hypothesized that the low effectiveness may be related to sharp edges of their holes ($R/D = 0$) perhaps initiating detrimental vortices inside the shaped hole.

It is worthwhile to note that higher $\bar{\eta}$ corresponded closely with the effective angle of diffused injection as mentioned by Saumweber and Schulz [12]. The effective angle of diffused injection ($\gamma$) is given by the angle between the diffuser floor of the hole and the external surface. It appears that shallower angles lead to greater cooling as shown in Figure 2.14. The shaped hole of the present study had an effective injection angle of 23º, which was relatively steep among the laidback fanshaped holes. It is also interesting to see that for shallower angles below 20º the cooling performance increases with $M\cdot s/D$ as compared with angles greater where increases in $M\cdot s/D$ result in decreased cooling. The authors note that not all shaped holes scale in this manner with only the effective injection angle.
Adiabatic Effectiveness at Moderate Freestream Turbulence Intensity

With the turbulence grid installed, adiabatic effectiveness was measured for the shaped holes at \( \text{DR} = 1.5 \). Contours of effectiveness measured at moderate freestream turbulence intensity are shown in Figures 2.15a-d. In comparing Figures 2.15a-d with 2.7a-d, it is seen that the freestream turbulence increased mixing between coolant and the mainstream, particularly evident on jet centerlines at low blowing ratios. At each blowing ratio, centerline effectiveness decreased with increased freestream turbulence. Mixing from increased freestream turbulence also manifested itself as increased lateral spreading of coolant relative to \( \text{Tu}_\infty = 0.5\% \), except at \( M = 0.5 \). Negligible coolant spreading occurred at \( M = 0.5 \) when comparing the low and moderate turbulence intensities. At \( M = 1 \) and \( M = 2 \), the \( \eta = 0.05 \) contour level for adjacent jets merged by \( x/D = 27 \). At \( M = 2 \) and 3, narrowing of coolant footprints similar to at \( \text{Tu}_\infty = 0.5\% \) occurred at \( \text{Tu}_\infty = 5.6\% \).

Figure 2.16 compares laterally-averaged effectiveness at low and moderate freestream turbulence intensities. For \( M = 0.5 \) and 1, the increased freestream turbulence decreased \( \bar{\eta} \) relative to the low turbulence case. At \( M = 2 \) and 3 the effect of increased freestream turbulence on \( \bar{\eta} \) was nearly negligible. The effect of increasing freestream turbulence is revealed by lateral effectiveness distributions at \( \text{Tu}_\infty = 0.5\% \) and 5.6\% shown in Figures 2.17a-b. Note the scale was reduced in Figure 2.17b as compared with Figure 2.17a to show the details. At 5D downstream of the shaped hole trailing edge, increased freestream turbulence did not change the lateral spreading of coolant. Far downstream of the holes at \( x/D = 32.4 \), as shown in Figure 2.17b, the coolant footprint at \( M = 3 \) was widened by the freestream turbulence but not so for \( M = 1 \). Freestream turbulence also caused reduction in centerline effectiveness relative to the \( \text{Tu}_\infty = 0.5\% \) performance.

The effect of freestream turbulence on area-averaged effectiveness is shown in Figure 2.18. Area-averaging was done over the range 2-22 diameters downstream of the hole trailing edge for comparison of the present study to shaped hole freestream-turbulence results from Saumweber et al. [49]. Figure 2.18 shows that at blowing ratios \( M = 0.5 \) and 1 in the present study, increased freestream turbulence of \( \text{Tu}_\infty = 5.6\% \) led to reductions in \( \bar{\eta} \) of up to 10\%. At higher blowing ratios, the freestream turbulence caused no reduction in \( \bar{\eta} \). In the study of Saumweber et al. [49], increasing freestream turbulence intensity from 3.6\%
Figure 2.15. Shaped hole $\eta$ contours at DR=1.5 and moderate freestream turbulence intensity of $\text{Tu}_\infty = 5.6\%$: (a) M=0.5, (b) M=1, (c) M=2, and (d) M=3.

Figure 2.16. Comparison of DR = 1.5 laterally-averaged effectiveness at low and moderate freestream turbulence intensity.
Figure 2.17. Lateral distributions of effectiveness compared between $Tu_\infty = 0.5\%$ and $5.6\%$, at $M=1$ and 3, $DR = 1.5$ for (a) $x/D = 7.4$ and (b) $x/D = 32.4$. Center pitch shown.

Figure 2.18. Area-averaged effectiveness for shaped holes plotted as a function of the blowing ratio. Averaged over 2-22 diameters downstream of the hole trailing edge.

...to 7.5% led to similar reductions (11%) in $\bar{\eta}$ at low blowing ratios. However, at high blowing ratios the percent reductions were 5-10%. The results of the present study are consistent with observations by Colban et al. [41] for shaped holes on an endwall. Most of the holes on the endwall were operating at blowing ratios below $M = 2$, and laterally-
averaged effectiveness only decreased slightly or remained unchanged when freestream turbulence intensity was elevated from 1.2% to 8.9%. Area-averaged effectiveness on the endwall decreased an average of 6% from the elevated freestream turbulence.

2.6 Conclusions

A baseline shaped hole was designed after reviewing shaped holes in public literature and their performance characteristics. Geometric parameters for the shaped hole match those seen with shaped holes in literature. Conservative expansion angles of 7º were selected so this shaped hole would not exhibit in-hole jet separation. The geometric parameters resulted in a shaped hole proposed to the gas turbine community as a new baseline with which other geometries may be compared, other investigations may be benchmarked, and other CFD studies may be validated.

Measurements of adiabatic effectiveness at low and high density ratio showed that the baseline shaped hole had a wider coolant distribution at high density ratio for the same blowing ratio as compared with the low density ratio. Lateral spreading of coolant from the baseline shaped hole was highly dependent on the blowing ratio. At blowing ratios above 1.5, effectiveness distributions narrowed with increased blowing ratio. Narrowing of the coolant distributions coincided with a shallow decay rate of laterally-averaged effectiveness with downstream distance. The peak effectiveness for the low and high density ratio cases occurred near a blowing ratio of 1.5.

Multiple geometric parameters for shaped holes influence their performance. One useful metric for shaped hole performance is area-averaged effectiveness as a function of coolant flowrate per pitch, which scales the cooling performance of most shaped holes at low flowrates. In comparing the baseline shaped hole to other shaped holes in literature at high flowrates, it is clear that cooling differences are quite pronounced. A conservative hole expansion of the baseline shaped hole presented in this study resulted in a cooling performance that was lower than some aggressively-expanded shaped holes.

Increasing the freestream turbulence to a moderate intensity did result in a widening of the jets at the intermediate to high blowing ratios. Area-averaged effectiveness decreased slightly with moderate turbulence at low blowing ratios, but not significantly for high blowing ratios.
Chapter 3.

Effect of High Freestream Turbulence on Flowfields of Shaped Film Cooling Holes†

Abstract

Shaped film cooling holes have become a standard geometry for protecting gas turbine components. Few studies, however, have reported flowfield measurements for moderately-expanded shaped holes and even fewer have reported on the effects of high freestream turbulence intensity relevant to gas turbine airfoils. This study presents detailed flowfield and adiabatic effectiveness measurements for a shaped hole at freestream turbulence intensities of 0.5% and 13%. Test conditions included blowing ratios of 1.5 and 3 at a density ratio of 1.5. Measured flowfields revealed a counter-rotating vortex pair and high jet penetration into the mainstream at the blowing ratio of 3. Elevated freestream turbulence had a minimal effect on mean velocities and rather acted by increasing turbulence intensity around the coolant jet, resulting in increased lateral spreading of coolant.

3.1 Introduction

In modern gas turbines the components immediately downstream of the combustor operate in an environment of high temperature and high turbulence intensity. Temperature of the mainstream gas is far above the melting temperature of metal components. Freestream turbulence intensities can be above 20% [59], which exacerbates convection to the components. Film cooling is used to protect components by ejecting air cooler than the mainstream from discrete holes in component surfaces, providing a layer of cool air over surfaces.

While effects of elevated freestream turbulence on adiabatic effectiveness are fairly well-understood, less is known about how freestream turbulence modifies the flowfields of jets from shaped holes. In the present study the performance of a publicly-available shaped

hole design was characterized at freestream turbulence intensities up to $\text{Tu}_c = 13.2\%$. Both flowfields and adiabatic effectiveness were measured.

Experimental data from this study provide a useful benchmark against which CFD simulations may be compared. Film cooling flows with high freestream turbulence are difficult to model computationally, as they involve interactions between disparate turbulent structures of the freestream and those of shear layers around the coolant jet. The data set presented in this chapter is fully available to the community for benchmarking purposes (http://www.mne.psu.edu/psuturbine).

### 3.2 Previous Studies

Multiple studies have examined the effect of high freestream turbulence on film cooling, mostly through studies of adiabatic effectiveness. Bons et al. [48] and Schmidt and Bogard [60] each measured adiabatic effectiveness for cylindrical holes at turbulence intensities up to 17% and found that freestream turbulence was detrimental due to mixing of the coolant. For detached jets from cylindrical holes, however, Schmidt and Bogard [60] found that increased freestream turbulence intensity actually increased effectiveness by bringing more coolant closer to the surface. Saumweber et al. [49] and Saumweber and Schulz [22] measured effectiveness for shaped holes at turbulence intensities up to 11% and found that elevated freestream turbulence only acted to mix out coolant of the attached jets, decreasing adiabatic effectiveness.

Previous flowfield measurements for shaped holes have focused on performance at low freestream turbulence intensity. Haven et al. [26] found that the breakout edge of shaped holes influenced interaction between the jet and mainstream, sometimes leading to unsteady vortices that partially cancelled the detrimental counter-rotating vortex pair (CRVP) inherent to jets in crossflow. Thole et al. [16] measured flowfields for shaped holes with aggressive expansion angles and found evidence of separation in the diffused outlet but saw no sign of a strong CRVP. These flowfield measurements and others by Fawcett et al. [50], Laveau and Abhari [61], and Jessen et al. [62] were performed at low freestream turbulence intensities.

Only a few studies have measured film cooling flowfields at elevated freestream turbulence intensity. One of the more recent studies was that of Wright et al. [63] who used
stereo PIV to measure flowfields in crossplanes downstream of laidback fanshaped holes at a freestream turbulence intensity of 8%. They observed a weak CRVP in the crossplane at the shaped hole trailing edge.

Only one study was found to report measured flowfields for shaped holes with high density ratio jet injection at elevated freestream turbulence intensity (auf dem Kampe et al. [23]). The present study builds upon those results by contrasting shaped hole performance at low and high freestream turbulence at conditions not previously reported in literature. Both the turbulence intensity of $\text{Tu}_{\infty} = 13.2\%$ and the blowing ratio of $\text{M} = 3$ in the present study are higher than in previous flowfield measurements. Results of the present study add to a suite of data characterizing performance of a publicly-available shaped hole design.

### 3.3 Experimental Facility and Test Conditions

All experiments were performed in the closed-loop wind tunnel shown in Figure 3.1. Film cooling measurements in this tunnel have been previously been reported by Eberly and Thole [55] for cylindrical holes and Schroeder and Thole [10] for shaped holes. As shown in Figure 3.1, mainstream air was circulated by an in-line centrifugal fan. Temperature of the mainstream was controlled by a bank of electric heaters and a chilled

![Figure 3.1. Schematic of the film cooling wind tunnel.](image-url)
water heat exchanger. Downstream of the heater bank were flow straighteners and a 6:1 contraction leading to a film-cooled flat plate in the test section. Mainstream conditions of 10 m/s and 295 K were maintained in the test section for the present study. The incoming boundary layer along the tunnel floor was removed by a suction loop, thereby initiating a new boundary layer at the plate leading edge. Downstream of the leading edge at x/D = -33 a trip wire initiated transition of the boundary layer to a turbulent state.

Freestream turbulence intensity at x/D = -2 was Tu₀ = 0.5% with no grid in place. To obtain the high freestream turbulence of Tu₀ = 13.2%, a grid with large vertical bars was installed at x/b = -14. Bar diameter was b = 38 mm and bars were spaced 76 mm apart center-to-center. At the hole trailing edge the length scale was Λₓ = 5.2D.

The film cooling flow was supplied by a coolant loop shown in the lower section of Figure 3.1. Air was diverted from the mainstream by a variable frequency blower that was hermetically sealed. The air was then cryogenically chilled in a heat exchanger using liquid nitrogen. Experiments for the present study were performed at a density ratio of DR = 1.5, which necessitated use of desiccant to remove moisture that would form frost at low temperatures.

Film cooling holes used in the present study were the shaped holes introduced by Schroeder and Thole [10]. Full specification of the geometry, including CAD models, is openly available for download at the authors’ website (http://www.mne.psu.edu/psuturbine). Geometric parameters for the shaped hole are shown in Table 3.1 and the hole is illustrated in Figure 3.2. The shaped hole featured expansion angles of 7º in the three directions from the metering-section centerline.

The shaped holes were machined in styrofoam residential sheathing (polystyrene) and installed in the flat plate floor of the test section. Polystyrene was chosen because of its low thermal conductivity of k = 0.029 W/m·K, which was suitable for adiabatic effectiveness measurements. Metering diameter of the holes was D = 7.75 mm for all experiments.

This study presents flowfields and adiabatic effectiveness measured at Tu₀ = 0.5% and 13.2% for M = 1.5 and 3.0 at a density ratio of 1.5. Additionally, adiabatic effectiveness is also presented for Tu₀ = 5.6% [10]. Flowfield measurements were made in two planes. The first plane was the centerline x-y plane and the second plane was the
y-z crossplane located at x/D = 4, which was 1.6D downstream of the shaped hole trailing edge. Location of the measurement planes is shown in Figures 3.3a-b.

### Table 3.1. Geometric Parameters of the Shaped Hole

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>P/D</td>
<td>6</td>
</tr>
<tr>
<td>L_m/D</td>
<td>2.5</td>
</tr>
<tr>
<td>α</td>
<td>30°</td>
</tr>
<tr>
<td>β_fwd, β_lat</td>
<td>7°</td>
</tr>
<tr>
<td>L_lat/D, L fwd/D</td>
<td>3.5</td>
</tr>
<tr>
<td>Area Ratio, AR</td>
<td>2.5</td>
</tr>
</tbody>
</table>

![Figure 3.2. Geometry of the shaped hole.](image)

![Figure 3.3. Measurement setups for (a) PIV in the centerline plane and (b) stereo PIV in the x/D = 4 crossplane.](image)
Flowfield Measurements

Film cooling flowfields were measured in the centerline plane and the crossplane using particle image velocimetry (PIV) in two different setups. A dual-head Nd:YLF laser capable of 10 kHz firing rate per head was used to illuminate seed particles in planar sheets of the flow. Images of the particles were captured on high speed CMOS cameras capable of recording images at up to 1024x1024 pixel resolution. For all flowfield measurements the seed used was droplets of di-ethyl-hexyl-sebecat (DEHS) from an aerosol generator providing mean particle diameter of 1 µm [64]. This diameter corresponded to Stokes numbers up to 0.010 for the present study, which, being much less than unity, indicated that seed followed the flow. Stokes number was based on a flow timescale of D/Uc = 0.4 ms for the blowing ratio M = 3. The mainstream and coolant were equally seeded with DEHS.

Flowfield data in the centerline plane were obtained using the setup shown in Figure 3.3a. Images were obtained by a single camera viewing normal to the centerline plane in an arrangement similar to that used by Eberly and Thole [55]. The laser sheet entered the tunnel from above, reflected off a small mirror downstream of the film cooling holes, and then proceeded upstream to illuminate the centerline plane. Laser sheet thickness was estimated to be 0.9 mm (0.12D). For the centerline plane, the camera recorded image pairs at 4 kHz with image size of 1024x256 pixels. Time-mean flowfields were obtained by averaging over at least 8000 instants spread over a time period of 2 seconds corresponding to more than 240 flow crossings of centerline field of view (x/D = -2 to 8.6). Time delay between laser pulses was chosen to provide mainstream particle displacements around 8 pixels. PIV calculations were performed using LaVision’s DaVis 8.2.1 commercial software [65]. A background image of minimum intensity was subtracted from all images and intensity was normalized in each frame to give equal weighting to all particles. Particle displacements, and thereby velocities, were calculated using a multi-pass scheme of interrogation windows ending with final window size of 16x16 pixels and 75% overlap. This final window size corresponded to 0.18D x 0.18D since spatial resolution was 11.6 pixels/mm. DaVis post-processing vector validation was performed using a median-based “universal outlier detection” [65] that removed and replaced spurious vectors, often with vectors based on secondary peaks in the cross-correlation. Vector validation was reasonable: post-processing modified only 5% of computed vectors. All
velocity field statistics reported in this dissertation were calculated from the post-processed velocity fields.

Flowfield measurements in the \( x/D = 4 \) crossplane were made using the setup in Figure 3.3b, which employed stereo PIV to measure all three components of velocity. The laser sheet entered the tunnel flush with the polystyrene surface through a side window. Two CMOS cameras with Scheimpflug lens adapters viewed opposite sides of the laser sheet at 40º from normal. The cameras recorded image pairs at 250 Hz with image size of 1024x512 pixels. Time-mean flowfields were obtained by averaging over at least 4000 instants spread over a time of at least 16 seconds. Time delay between laser pulses was set between 22-26 µs to follow the best practice of limiting out-of-plane displacement to \( \frac{1}{4} \) the sheet thickness. These time delays corresponded with particle displacements in the mainstream of 4 pixels for the dewarped images used to compute vectors (3.1 pixels in raw images). The light sheet was 1.9 mm (0.25D) thick for crossplane measurements. Only intensity normalization was performed on crossplane images. Velocities were calculated with a multi-pass scheme ending in 32x32 pixel interrogation windows with 50% overlap. These windows were 0.19D x 0.19D based on the spatial resolution of 21.9 pixels/mm. Universal outlier detection was used, however few vectors required modification for the crossplane data.

**Adiabatic Effectiveness Measurements**

Adiabatic effectiveness was measured using images of the film-cooled surface taken with a FLIR SC620 infrared camera. The measurement is described in Schroeder and Thole [10] and background is given by Eberly [66]. Briefly, the infrared camera viewed the film-cooled surface through a ZnSe window in the test section ceiling. To ensure that temperatures were accurately detected over the entire range of surface temperatures, camera output was calibrated to temporarily-installed thermocouples whose positions are noted by red squares in Figure 3.3b. The calibrated infrared images provided \( T_{aw} \). Freestream and coolant temperatures both were averages, each taken from multiple thermocouples in the respective locations (mainstream, and in plenum 2.5D below film cooling hole entrances).
Uncertainty Analysis

An uncertainty analysis was performed for both flowfield and adiabatic effectiveness data, as well as for variables describing test conditions. Uncertainties were propagated using the partial derivatives method of Figliola and Beasley [56] and values reported here are for a 95% confidence interval.

In terms of test conditions, uncertainty in density ratio was found to be low at ±0.02. For blowing ratio the uncertainty was dominated by bias uncertainty of the Venturi flowmeter itself with a maximum blowing ratio uncertainty at M = 1.5 of ±4.5%.

For adiabatic effectiveness the uncertainty was driven by uncertainty in temperature of the plate surface and coolant. Uncertainty in surface temperature was ±1.8 °C based on scatter in infrared camera calibration data and bias uncertainty of thermocouples used in the calibration. Adiabatic effectiveness uncertainty was calculated to be $\delta\eta = \pm 0.025$.

Uncertainties in PIV flowfield measurements were based on an instantaneous displacement uncertainty of ±0.15 pixels, a conservatively-high estimate [67]. For particle displacements in the mainstream, streamwise velocity uncertainty was estimated to be ±1.9% in the centerline plane and ±4.8% in the crossplane. Displacements were smaller near the wall, for instance minimum displacement at $y/D = 0.25$ in the centerline plane was 1.9 pixels, corresponding to a worst-case uncertainty of ±8%. Repeatability tests in the centerline plane were used to estimate precision uncertainties, found to be ±4% for U, ±2% for V, ±4% for $u'$, ±4% for $v'$, and ±5% for $\bar{u}'v'$ shear stress. Percentages were based on $U_\infty$ for U and V, and for other variables were based on maximum magnitudes observed for $u'$, $v'$, and $\bar{u}'v'$.

3.4 Results and Discussion

Mainstream Approach Flow

The freestream turbulence field was characterized at low, moderate, and high freestream turbulence intensities of $Tu_\infty = 0.5\%, 5.6\%, \text{ and } 13.2\%$. Reported values of $Tu_\infty$ are based on two-component PIV measurements at $x/D = -2$. Turbulence decayed as it flowed downstream, for instance the $Tu_\infty = 13.2\%$ turbulence intensity decayed to 11.7% by $x/D = 8$. At each freestream turbulence intensity, measurements showed that mean
streamwise velocity of the approach flow was uniform laterally and vertically within ±2.5%. Uniformity of \( u'/U_\infty \) in the same profiles was within ±0.4% for the high freestream turbulence condition.

To quantify anisotropy, freestream values of \( v'/U_\infty \) were examined in PIV measurements at \( Tu_\infty = 13.2\% \). Vertical velocity fluctuations were less than those in the streamwise direction, giving \( v'/u' = 0.8 \). This ratio is the same as that seen in the freestream for boundary layer measurements at \( Tu_\infty = 15\% \) made by Thole and Bogard [68].

Approach boundary layer mean and rms values are shown in Figures 3.4 and 3.5. The streamwise mean velocity profile for \( Tu_\infty = 0.5\% \) in Figure 3.4 agreed with Spalding’s Law. At higher freestream turbulence, mean velocity profiles showed the expected diminishment of the wake region of the boundary layer profile. Figure 3.5 gives profiles of the fluctuating component of streamwise velocity. Good agreement is seen between the \( Tu_\infty = 13.2\% \) condition and the \( Tu_\infty = 15\% \), \( Re_\delta^2 = 620 \) boundary layer studied by Thole and Bogard [68]. At each freestream turbulence condition, boundary layers were measured at multiple pitchwise locations and average parameter values are given in Table 3.2.

![Figure 3.4. Approach boundary layers measured at x/D = -2.3 for low, moderate, and high freestream turbulence intensity.](image)

![Figure 3.5. Profiles of fluctuating streamwise velocity at x/D = -2.3 for low, moderate, and high turbulence intensity.](image)

### Table 3.2. Boundary Layer Characteristics

<table>
<thead>
<tr>
<th>( Tu_\infty )</th>
<th>( \delta_2/D )</th>
<th>( H )</th>
<th>( Re_\delta^2 )</th>
<th>( u_\tau )</th>
<th>( c/c_{f,0} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5%</td>
<td>0.14</td>
<td>1.45</td>
<td>670</td>
<td>0.5 m/s</td>
<td>1.0</td>
</tr>
<tr>
<td>13.2%</td>
<td>0.12</td>
<td>1.38</td>
<td>580</td>
<td>0.57 m/s</td>
<td>1.19</td>
</tr>
</tbody>
</table>
Low Freestream Turbulence Intensity Results

Contours of mean streamwise velocity in the centerline plane are shown in Figures 3.6a-b for the blowing ratios $M = 1.5$ and 3 at low freestream turbulence. Time-mean streamlines are also shown. For the $M = 1.5$ case in Figure 3.6a, streamwise velocity contours show the near-wall mainstream decelerated as it approached the hole breakout. Streamwise velocity then increased as it flowed over the upstream half of the hole breakout. Streamwise velocity decreased over the downstream half of the breakout and the horizontal streamlines downstream indicated attached flow. Figure 3.6b shows the boundary layer was similarly disrupted at $M = 3$, although at this blowing ratio a region of high streamwise velocity exiting the hole penetrated high into the mainstream. A shear layer developed behind (beneath) the coolant jet, as evidenced by how streamwise velocity immediately decreased downstream of the hole trailing edge at $M = 3$.

![Figure 3.6](image.png)

**Figure 3.6.** Contours of time-mean streamwise velocity and streamlines in the centerline plane for $DR = 1.5$, $Tu_{\infty} = 0.5\%$ at (a) $M = 1.5$ and (b) $M = 3$.

Corresponding fields of turbulence intensity in the centerline plane are shown in Figures 3.7a-b. Maximum turbulence intensity in the jet increased with blowing ratio and occurred immediately above the hole breakout, exceeding 30% for the $M = 3$ case. Figure 3.7b shows that at $M = 3$ the maximum turbulence intensity occurred close to the leading edge of the hole breakout. One source of this high turbulence intensity was the strong shear layer developing at the jet-mainstream interface, although turbulence inside the
hole caused by a separation region at the hole inlet is another possible contributor. Turbulence decayed as flow advected downstream.

Contours of $\bar{u}'v'$ turbulent shear stress in the centerline plane are shown in Figures 3.8a-b. Regions of positive and negative turbulent shear stress occurred at both $M = 1.5$ and 3. At $M = 1.5$, Figure 3.8a shows that the region of positive $\bar{u}'v'$ stress was small and occurred above the upstream portion of the hole breakout where high streamwise velocity was observed in Figure 3.6a. Negative $\bar{u}'v'$ stress for $M = 1.5$ began at the leeward

![Figure 3.7. Contours of turbulence intensity and time-mean streamlines in the centerline plane for DR = 1.5, $Tu_{\infty} = 0.5\%$ at (a) $M = 1.5$ and (b) $M = 3$.](image)

![Figure 3.8. Contours of $\bar{u}'v'$ turbulent shear stress in the centerline plane for DR = 1.5, $Tu_{\infty} = 0.5\%$ at (a) $M = 1.5$ and (b) $M = 3$.](image)
portion of the hole breakout and persisted downstream. These \( u'v' \) regions are opposite in sign to those observed by Thole et al. [16] for shaped holes, due to the shaped holes of Thole et al. having a larger expansion angle and a greater area ratio. At the blowing ratio of \( M = 1 \) tested by Thole et al., the coolant exiting shaped holes had lower velocity than the approaching mainstream, ensuring a positive \( \partial U/\partial y \) gradient at the hole outlet. In the present study both positive and negative \( \partial U/\partial y \) were observed, which will be discussed with velocity profiles in a later figure.

At \( M = 3 \) the magnitude of \( u'v' \) stresses was higher than at \( M = 1.5 \), as expected considering the higher turbulence intensity in the jet. Figure 3.8b shows a long region of positive \( u'v' \) stress that extended over the top of the coolant jet, corresponding to a region of negative \( \partial U/\partial y \) producing the stress. Shape of the negative \( u'v' \) region at \( M = 3 \) was similar to that at \( M = 1.5 \).

Contours of mean streamwise velocity, turbulent shear stress, and turbulence intensity in the \( x/D = 4 \) crossplane are shown in Figures 3.9a-b, 3.10a-b, and 3.11a-b. Note that contour level scales are reduced relative to those for the centerline plane to show detail. Arrows on the contours show the in-plane mean velocities, revealing a weak counter-rotating vortex pair (CRVP). The CRVP grew larger and moved higher above the plate with increased blowing ratio. Contours of mean velocity in Figure 3.9b and turbulent shear

![Figure 3.9](image)

**Figure 3.9.** Contours of mean streamwise velocity in the \( x/D = 4 \) crossplane for \( DR = 1.5, Tu_\infty = 0.5\% \) at (a) \( M = 1.5 \) and (b) \( M = 3 \). In-plane mean velocity is shown by white arrows.
Figure 3.10. Contours of turbulent shear stress in the x/D = 4 crossplane for DR = 1.5, $Tu_\infty = 0.5\%$ at (a) $M = 1.5$ and (b) $M = 3$. In-plane mean velocity is shown by arrows.

Figure 3.11. Contours of turbulence intensity in the x/D = 4 crossplane for DR = 1.5, $Tu_\infty = 0.5\%$ at (a) $M = 1.5$ and (b) $M = 3$. In-plane mean velocity is shown by arrows.
stress in Figure 3.10b have a kidney shape caused by the CRVP having been stronger at M = 3 than at M = 1.5.

The contours of mean velocity, turbulent shear stress, and turbulence intensity in the x/D = 4 crossplane indicate extent of the coolant jet. For M = 1.5 the jet extended laterally between z/D = ±1.2 and did not penetrate above a height of y/D = 1.2. The jet was slightly wider and much higher at M = 3, extending between z/D = ±1.3 and up to a height of y/D = 1.7. Although overall extent of mean velocity and turbulence intensity contours match at M = 3, maximum turbulence intensity in Figure 3.11b occurred at a position below the region of maximum velocity in Figure 3.9b. The mismatch indicates turbulence production in the shear layer between the jet and wall.

Contours of adiabatic effectiveness measured at low freestream turbulence, previously reported by Schroeder and Thole [10], are shown in Figures 3.12a-b. The principle behavior seen in the adiabatic effectiveness contours was narrowing of the coolant patterns at high blowing ratios. Figures 3.12a-b show that effectiveness patterns immediately downstream of holes were narrower at M = 3 than at 1.5. Narrower contact between the coolant jet and surface at M = 3 can also be discerned from how effectiveness contours merged at midpitches for M = 1.5 but did not merge by x/D = 40 for M = 3.

Figure 3.12. Contours of adiabatic effectiveness for DR = 1.5, Tu_{∞} = 0.5% at (a) M = 1.5 and (b) M = 3.0 [10]. Gray dashed lines illustrate position of the two flowfield measurement planes.
Narrowing of the coolant patterns was due to high jet penetration at the high blowing ratio relative to the low blowing ratio. Increased jet penetration is evident from comparing the streamwise velocity contours for $M = 1.5$ and 3 in Figures 3.6a-b and 3.9a-b. Turbulence intensity contours in Figures 3.7a-b also indicated much higher vertical extent for the $M = 3$ jet. Jet penetration was accompanied by a decrease in contact between the jet and the wall. Effectiveness contours indicate that contact became more restricted to the centerline region from $M = 1.5$ to 3, even though crossplane contours show the jet itself was not narrower at $M = 3$. The stronger CRVP at $M = 3$ swept mainstream flow inward underneath the jet.

Although less coolant remained on the surface at $M = 3.0$, adiabatic effectiveness along jet centerlines in Figures 3.12a-b was about the same between $M = 1.5$ and 3. Preservation of the centerline effectiveness at high blowing ratio indicated that the base of the $M = 3$ jet remained attached to the surface. While streamlines for $M = 3$ in Figure 3.6b tended upwards, their inclination was much less than that shown by Eberly and Thole [55] for detached jets from cylindrical holes.

**High Freestream Turbulence Intensity Results**

Flowfields were measured at $M = 1.5$ and 3.0 with the high freestream turbulence grid installed, generating turbulence intensity of $Tu_\infty = 13.2\%$. Centerline plane contours are not shown because they appeared similar to the $Tu_\infty = 0.5\%$ contours except for increased turbulence intensity in the freestream. Flowfield variables in the x/D = 4 crossplane are shown in contours of Figures 3.13a-b, 3.14a-b, and 3.15a-b.

Figures 3.13a-b show that freestream turbulence caused little change in mean velocities. By comparing back to Figures 3.9a-b it can be seen that shape of the coolant jet, position of the jet, and magnitude of CRVP in-plane velocities only changed minimally. The increased freestream turbulence only acted to smooth velocity gradients, as can be seen by the slightly-wider region of slow velocity for $M = 1.5$ and the more evenly-spaced velocity contour lines for the $M = 3$ jet.

Crossplane contours of $\bar{u}'\bar{v}'$ shear stress for $Tu_\infty = 13.2\%$ are shown in Figures 3.14a-b. Comparing back to the low freestream turbulence cases in Figures 3.10a-b reveals that increased freestream turbulence spread the region of correlated $\bar{u}'\bar{v}'$ stress,
similar to how mean velocity contours Figures 3.13a-b are spread slightly more than those in Figures 3.9a-b. Magnitude of $u'v'$ was similar to that at $Tu_\infty = 0.5\%$.

Turbulence intensity in the crossplane for $Tu_\infty = 13.2\%$ cases was quite different between the blowing ratios of $M = 1.5$ and 3 as shown by Figures 3.15a-b. The changes due to freestream turbulence are discerned by comparing back to Figures 3.11a-b. For $M = 1.5$ in Figure 3.11a there was a region of high turbulence intensity reaching 8% at a height of $y/D = 0.4$ above the wall. This pattern was no longer discernable at high freestream turbulence intensity. Instead, Figure 3.15a shows turbulence intensity was somewhat uniform across the entire hole pitch, ranging only from 10.0-11.5% at $y/D = 0.4$. Turbulence intensity from the freestream was greater than that associated with the jet. A region of lower turbulence intensity near the wall extended across $z/D = \pm 2$, which was wider and still more turbulent than the $M = 1.5$ jet at $Tu_\infty = 0.5\%$.

The situation was different at $M = 3.0$, where maximum turbulence intensity in the coolant jet reached 15% at low freestream turbulence (Figure 3.11b). For the high freestream turbulence case in Figure 3.15b the jet was still evident from turbulence intensity contours that were of higher magnitude than the surrounding mainstream. Maximum turbulence intensity was 16% and effectively the same as at low freestream turbulence. Turbulence intensity contours slightly changed shape with high freestream turbulence, they

![Figure 3.13. Contours of mean streamwise velocity in the x/D = 4 crossplane for DR = 1.5, Tu_\infty = 13.2\% at (a) M = 1.5 and (b) M = 3. In-plane mean velocity is shown by white arrows.](image)
Figure 3.14. Contours of turbulent shear stress in the x/D = 4 crossplane for DR = 1.5, Tu_\infty = 13.2% at (a) M = 1.5 and (b) M = 3. In-plane mean velocity is shown by arrows.

Figure 3.15. Contours of turbulence intensity in the x/D = 4 crossplane for DR = 1.5, Tu_\infty = 13.2% at (a) M = 1.5 and (b) M = 3. In-plane mean velocity is shown by arrows.
extended out farther laterally at high freestream turbulence. For instance, the 12.5% contours extended past \( z/D = \pm 1.5 \) for the high freestream turbulence case. At low freestream turbulence the jet periphery was of course associated with a lower level of turbulence intensity (~7%), which extended to \( z/D = \pm 1.3 \).

Figures 3.16 through 3.19 compare low and high freestream turbulence cases using vertical profiles of flowfield variables. Profiles are presented for three positions in the centerline plane. Note that the positions \( x/D = 0 \) and 2 are over the diffused outlet of the shaped hole.

Profiles of mean streamwise velocity are shown in Figure 3.16 for both blowing ratios and freestream turbulence intensities. Note that at \( x/D = 0 \) the profiles show negative \( \partial U/\partial y \) near the holes, which was not observed in flowfields measured by Thole et al. [16] for aggressively-expanded shaped holes. In Figure 3.16 the mean velocity gradients near the wall changed sign by \( x/D = 2 \) for both blowing ratios. For \( M = 1.5 \) the profiles monotonically increased with height at the downstream stations of \( x/D = 2 \) and 4. For \( M = 3 \) the vertical profiles showed a peak in streamwise velocity. The peak existed despite the fact that effective velocity ratio, based on hole exit area, was only 0.8 for the blowing ratio of \( M = 3 \). At \( x/D = 4 \) the peak was high above the wall at \( y/D = 0.9 \) and had a

![Figure 3.16](image)

**Figure 3.16.** Profiles of mean streamwise velocity in the centerline plane at three streamwise positions, for both low and high freestream turbulence intensity.
magnitude of \( \frac{U}{U_\infty} = 1.3 \). Figure 3.16 shows that increased freestream turbulence slightly smoothed the velocity gradients, which was particularly evident for the \( M = 1.5 \) case.

For clarity, profiles of rms velocity components are plotted separately for \( M = 1.5 \) and 3 with Figures 3.17 and 3.18. For low freestream turbulence cases at \( M = 1.5 \), Figure 3.17 shows that \( u' \), \( v' \), and \( w' \) were all small in the freestream as expected. The solid lines for the low freestream turbulence case show that closer to the wall, inside the coolant jet, \( u' \) was the greatest contributor to the overall turbulence intensity. Dashed lines for the \( T_{u_\infty} = 13.2\% \) case show that in the freestream \( v' \) was 20\% less than \( u' \) as previously discussed. Inside the jet the \( v' \) component was unchanged at high freestream turbulence relative to low freestream turbulence, whereas the \( u' \) and \( w' \) components were increased by high freestream turbulence and were responsible for the increased turbulence intensity noted in Figure 3.15a contours. The profile at \( x/D = 2 \) shows an interesting deficit in \( u' \) near \( y/D = 0.6 \). Although the rms value was greater than that for \( u' \) in the low freestream turbulence case, \( u' \) was decreased relative to its value at \( x/D = 0 \). The \( u' \) deficit was due to mainstream flow accelerating around the coolant jet, redistributing the turbulent fluctuations to the \( v' \) component.

Figure 3.18 shows that components of rms velocity were higher at \( M = 3 \) than at \( M = 1.5 \). For \( T_{u_\infty} = 0.5\% \) at \( x/D = 4 \) it is seen that \( w' \) was the smallest rms velocity component, except near the wall where \( w' \) had its peak and \( v' \) was restricted by presence of the wall. At \( x/D = 4 \), the peak in \( v' \) occurred at \( y/D = 1.0 \) and the peak in \( u' \) occurred at \( y/D = 0.6 \). The \( u' \) peak was below the \( y/D = 0.9 \) mean velocity peak noted in Figure 3.16, which provides evidence that at least some of the turbulence at \( M = 3 \) was produced by the shear layer below the velocity peak. Similar behavior was observed by Pietrzyk et al. [69] for attached jets from cylindrical holes. At \( DR = 2, M = 0.5 \) they observed a peak in the turbulence intensity profile produced by the shear layer developing in the lower part of the jet. Figure 3.18 also shows that increased turbulence of \( T_{u_\infty} = 13.2\% \) only slightly increased the rms velocity components inside the coolant jet. The \( u' \) component dominated over \( v' \) both in the freestream and in the coolant jet as was seen at \( M = 1.5 \).

Vertical profiles of \( \bar{u}'\bar{v}' \) shear stress in Figure 3.19 show little difference between the low and high freestream turbulence cases. Consistent with the \( M = 3 \) velocity peaks in Figure 3.16, each \( M = 3 \) vertical profile of \( \bar{u}'\bar{v}' \) shear stress transitioned from negative to
Figure 3.17. Profiles of velocity fluctuations in the centerline plane at three streamwise positions, for M = 1.5 at both low and high freestream turbulence intensity.

Figure 3.18. Profiles of velocity fluctuations in the centerline plane at three streamwise positions, for M = 3.0 at both low and high freestream turbulence intensity. Legend is the same as in Figure 3.17.

Figure 3.19. Profiles of turbulent shear stress in the centerline plane at three streamwise positions, for both low and high freestream turbulence intensity. Legend is the same as in Figure 3.16.
positive near the height of the streamwise velocity peak (except for at the injection location of \(x/D = 0\)). In each case the \(\bar{u}'v' = 0\) crossing occurred slightly below the \(y/D\) height of the streamwise velocity peak.

Effectiveness contours at \(Tu_\infty = 13.2\%\) are shown in Figures 3.20a-b. Effectiveness patterns still narrowed from \(M = 1.5\) to 3 as was observed at low freestream turbulence. However, freestream turbulence increased dilution and lateral spreading of coolant over the surface. Dilution is apparent from how centerline effectiveness decreased for the \(Tu_\infty = 13.2\%\) cases as compared to \(Tu_\infty = 0.5\%\) cases in Figures 3.12a-b. Lateral spreading was more apparent far downstream of the holes, as seen by comparing Figures 3.20a-b to their \(Tu_\infty = 0.5\%\) counterparts. Increased lateral spreading of coolant was expected since regions of turbulence intensity associated with the jet were wider in Figures 3.15a-b than in Figures 3.11a-b.

Laterally-averaged adiabatic effectiveness is compared for low, moderate, and high freestream turbulence in Figure 3.21. Area-averaged adiabatic effectiveness was also calculated, averaged over the region 2-22 diameters downstream of the shaped hole trailing edge. Over the majority of the streamwise distance, Figure 3.21 shows that laterally-averaged effectiveness was higher at \(M = 1.5\) as compared to \(M = 3\) due to the previously

![Figure 3.20. Contours of adiabatic effectiveness for DR = 1.5, Tu_\infty = 13.2% at (a) M = 1.5 and (b) M = 3.0. Dashed lines illustrate position of the two flowfield measurement planes.](image)
Figure 3.21. Laterally-averaged adiabatic effectiveness for DR = 1.5, M = 1.5 and 3, at three freestream turbulence intensities.

mentioned jet penetration with high blowing ratio. For the lower blowing ratio of M = 1.5, increased freestream turbulence was detrimental because it diluted coolant that was near the wall. At M = 1.5, area-averaged effectiveness decreased 17% due to freestream turbulence increasing from Tu_∞ = 0.5% to 13.2%. This was consistent with Saumweber et al. [49] who averaged over the same downstream area for shaped holes and observed that area-averaged effectiveness decreased up to 11% as freestream turbulence was increased from 3.6% to 7.5%.

Behavior at M = 3 contrasts with that at M = 1.5. Figure 3.21 shows that increased freestream turbulence was not detrimental to laterally-averaged effectiveness for M = 3. Rather, area-averaged effectiveness over the previously-stated averaging region increased 6% when freestream turbulence was increased from Tu_∞ = 0.5% to 13.2%. Effectiveness for Tu_∞ = 5.6% was also consistent with this trend, however, it must be noted that the increases in laterally-averaged effectiveness between these M = 3 cases were within the experimental uncertainty of δη = ±0.025. Still, even if the trend at M = 3 is one of unchanging area-averaged effectiveness, it contrasts with previous literature which has consistently shown decreased performance of shaped holes with freestream turbulence,
especially for turbulence intensities reaching 13%. Increased effectiveness due to high freestream turbulence has previously been observed for cylindrical holes, for instance Bons et al. [48] and Schmidt and Bogard [60].

3.5 Conclusions

Flowfield measurements for a shaped hole at blowing ratios of 1.5 and 3 showed little change in the mean velocity field with elevation of freestream turbulence intensity from 0.5% to 13.2%. At the blowing ratio of 3 the jet penetrated high above the surface which caused lower adiabatic effectiveness than occurred at the blowing ratio of 1.5. The M = 3 jet also featured high turbulence intensity near the hole breakout and a counter-rotating vortex pair that was clearly visible in the crossplane. Vertical profiles downstream of the hole showed that significant turbulence was generated by a shear layer developing behind blockage caused by the M = 3 jet.

Elevated freestream turbulence increased velocity fluctuations surrounding the coolant jet and thereby increased lateral spreading of coolant. Velocity fluctuations inside the jet increased minimally with elevated freestream turbulence, except in cases where turbulence intensity associated with the jet itself was less than turbulence intensity associated with the freestream. This exception occurred at the blowing ratio of 1.5, where turbulence intensity in the crossplane was fairly uniform across the hole pitch and was everywhere higher than at low freestream turbulence.

The measured flowfields contribute to our understanding of the physics of shaped hole film cooling. At high blowing ratios, coolant jets even from shaped holes can penetrate high above the surface and the effect of high freestream turbulence is not always detrimental. Turbulence in the coolant jet depends on blowing ratio and is anisotropic, but freestream turbulence does not significantly alter anisotropy of this turbulence. The relationship between mean velocity gradient, turbulence production, and the streamwise-vertical component of turbulent shear stress also appears to be unaffected by high freestream turbulence. These insights can be used to improve the turbulence models and correlations used by designers to predict film cooling from shaped holes.
Chapter 4.
Thermal Field Measurements for a Shaped Hole at Low and High Freestream Turbulence Intensity

Abstract

Shaped holes are increasingly selected for airfoil cooling in gas turbines due to their superior performance over that of cylindrical holes, especially at high blowing ratios. The performance of shaped holes is regarded to be result of the diffused outlet which slows and laterally-spreads coolant, causing coolant to remain close to the wall. However, few thermal field measurements exist to verify this behavior at high blowing ratio or to evaluate how high freestream turbulence alters the coolant distribution in jets from shaped holes. The present study reports measured thermal fields, along with measured flowfields, for a shaped hole at blowing ratios up to 3 at both low and high freestream turbulence intensities of 0.5% and 13.2%. Thermal fields at low freestream turbulence intensity showed that the coolant jet was initially attached, but far downstream of the hole the jet lifted away from the surface due to the counter-rotating vortex pair. Elevated freestream turbulence intensity was found to cause strong dilution of the coolant jet and also increased dispersion, almost exclusively in the lateral as opposed to the vertical direction. Dominance of lateral dispersion was due to the influence of the wall on freestream eddies, as indicated from changes in turbulent shear stress between the low and high freestream turbulence cases.

4.1 Introduction

With combustion temperatures far exceeding the melting temperature of metal components in gas turbines, film cooling holes are critical for maintaining parts at acceptable service temperature. Cooling designers often use shaped holes, owing to how shaped holes promote jet attachment and exhibit adiabatic effectiveness that only gradually varies with blowing ratio [25]. At high coolant flowrates, shaped holes can deliver

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satisfactory performance whereas jets from cylindrical holes are fully-detached [7]. Recent studies indicate interest in shaped hole performance at high coolant flowrates: adiabatic effectiveness for shaped holes is reported by Heneka et al. [8] up to $M = 3.0$ ($I = 6.0$) and by Colban et al. [45] up to $M = 4.0$ ($I = 8.0$). However, corresponding measurements of thermal field at high blowing ratio have not been made to investigate performance of these shaped hole jets. Little is known about the distribution of coolant within these jets, especially for cases with high freestream turbulence intensity that is characteristic of flow exiting the gas turbine combustor.

The present study reports thermal fields measured for shaped holes at high blowing ratio. First, thermal fields are presented for a condition of low freestream turbulence intensity, $T_u = 0.5\%$. Comparison is made to flowfield data to gain insight into behavior of the shaped hole jet. Then thermal field and flowfield data is presented for high freestream turbulence intensity of $T_u = 13.2\%$. Jet behavior is again discussed. Differences in how coolant disperses at low and at high freestream turbulence intensity are explained. As well as illuminating behavior of shaped hole jets, data of the present study is useful for validating film-cooling CFD models at quiescent and at highly-turbulent freestream conditions.

### 4.2 Previous Studies

Thermal field studies of film cooling flows provide insights not available from the surface measurements alone. For example, Thole et al. [7] measured thermal fields in the centerline plane of cylindrical holes and observed the different trajectories of jets over a range of density ratios and coolant flowrates. From the thermal fields they identified three regimes of jet-trajectory behavior for cylindrical holes: attached ($I < 0.4$), initially detached but then reattaching downstream ($0.4 < I < 0.8$), and fully detached ($I > 0.8$).

Thermal field measurements with shaped holes are rare. Kohli and Bogard [21] reported thermal fields for shaped holes with steep injection angle ($\alpha = 55^\circ$) at blowing ratios up to $M = 0.8$. They observed mainstream ingestion into the diffused outlets, but performance was still superior to cylindrical holes because coolant was spread laterally by the diffused outlets. Takeishi et al. [70] used PLIF to measure thermal fields for shaped
holes at blowing ratios up to $M = 1.5$ and likewise observed instances of mainstream ingestion. They also observed increased coolant penetration into the mainstream with increased blowing ratio, with the coolant jet detaching at the shaped hole trailing edge in one instance. Funazaki et al. [71] reported thermal fields at $M = 0.5$ and 1.0 for shaped holes with and without protrusions on the film cooled surface upstream of the holes. Protrusions were shown to cause slight coolant dilution but also increased lateral spreading of coolant close to the wall. All these shaped hole studies were performed with low freestream turbulence intensity.

While thermal field measurements at high freestream turbulence intensity have not been previously reported for shaped holes, the authors are aware of two experimental studies on cylindrical holes. Kohli and Bogard [59] measured the fluctuating thermal field for $M = 0.4$ jets from cylindrical holes with freestream turbulence of 20% and found that the high freestream turbulence “obliterated” the jet-mainstream interface. Mainstream fluid intermittently penetrated through the coolant jet to the wall, and conversely some coolant was ejected from the jet into the mainstream. Cutbirth and Bogard [72] measured time-mean thermal fields for compound-angle cylindrical holes on a vane pressure side with turbulence intensity of 20% at the cascade inlet. They observed the elevated freestream turbulence caused unsteady lateral oscillation (displacement) of the coolant jet rather than increased dispersion of the coolant jet. Correspondingly, flowfield measurements showed that turbulence surrounding the coolant jet was dominated by fluctuations in the lateral velocity component.

To add to this thermal field literature, the present study reports thermal fields for a baseline, publicly-available shaped hole design. Measurements were performed at high blowing ratios up to $M = 3$ and were performed at both low and high freestream turbulence intensities. To provide a comprehensive understanding, thermal field data is complemented by previously-measured flowfield data [73] at the same conditions. Besides adding to our understanding of shaped hole performance, data of the present study is useful for benchmarking of CFD simulations and qualification of new film cooling experimental facilities.
4.3 Experimental Facility and Methods

A recirculating wind tunnel, previously shown in Figure 3.1, was used for experiments. Mainstream test section air was driven by an in-line centrifugal fan and air temperature was conditioned by a chilled-water heat exchanger and an electric heater bank. Air flowed through a 6:1 contraction to enter the test section featuring a flat-plate floor with a row of five shaped holes. For the present study, test section conditions were maintained at 295 K and 10 m/s mainstream velocity.

To control the boundary layer over the flat plate, a suction loop was used to remove the incoming boundary layer so that a new boundary layer developed from the plate leading edge. A trip wire at x/D = -33 caused transition to a turbulent boundary layer. Per boundary layer profiles previously reported by Schroeder and Thole [73], the boundary layer was fully-turbulent at x/D = -2. Table 4.1 provides characterization of the mainstream approach boundary layer.

Tests were performed at two freestream turbulence intensities in this study, Tu∞ = 0.5% and 13.2%. Turbulence intensities are reported for the x/D = -2 location. The low intensity of Tu∞ = 0.5% was obtained without a turbulence grid installed in the wind tunnel. High freestream turbulence intensity of Tu∞ = 13.2% was obtained by precisely positioning a passive grid of vertical bars near the test section entrance at x/b = -14, where bar diameter was b = 38 mm. Bars were spaced apart 2b center-to-center. As measured by hot wire, the integral length scale of the Tu∞ = 13.2% turbulence was Λx = 5.2D at the shaped hole trailing edge.

<table>
<thead>
<tr>
<th>Tu∞</th>
<th>δ2/D</th>
<th>H</th>
<th>Reδ2</th>
<th>cf/cf,0</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5%</td>
<td>0.14</td>
<td>1.45</td>
<td>670</td>
<td>1.0</td>
</tr>
<tr>
<td>13.2%</td>
<td>0.12</td>
<td>1.38</td>
<td>580</td>
<td>1.19</td>
</tr>
</tbody>
</table>

The supply loop for the film cooling flow is shown at the bottom of Figure 3.1. Air for coolant was withdrawn from the wind tunnel far upstream of the test section and was driven by a hermetically-sealed blower through a heat exchanger to cool the air using liquid nitrogen. Coolant flowrate was measured by a Venturi flowmeter and then coolant flowed up through a plenum with fine screens to be evenly-distributed to entrances of the five shaped holes. All experiments were performed at the density ratio DR = 1.5, which
necessitated strategies to avoid frost formation on cold surfaces. Prior to experiments, all wind tunnel air was dried by routing it through the desiccant vent dryer branch of the coolant loop. Also, during experiments a separate pipeline was used to keep the wind tunnel positively pressurized with nitrogen gas.

The shaped holes used in this study were those introduced as a baseline geometry by Schroeder and Thole [10]. CAD models and performance data for these holes are publicly available for download at the authors’ website (http://www.mne.psu.edu/psuturbine). Geometry of this hole is shown in Figure 4.1 and geometric parameters are listed in Table 4.2. Expansion angles of the shaped hole were 7° in each of the three directions from the metering-section centerline. Metering diameter was D = 7.75 mm.

The flat plate material in which the shaped holes were machined was styrofoam residential sheathing (polystyrene). Polystyrene was the preferred material for adiabatic effectiveness measurements and thermal field measurements due to its low conductivity of k = 0.029 W/m·K. The test plate material remained the same for flowfield measurements.

![Figure 4.1. Shaped hole geometry.](image)

<table>
<thead>
<tr>
<th>Table 4.2. Geometric Parameters of the Shaped Hole</th>
</tr>
</thead>
<tbody>
<tr>
<td>P/D</td>
</tr>
<tr>
<td>α</td>
</tr>
<tr>
<td>β_fwd, β_lat</td>
</tr>
</tbody>
</table>

**Adiabatic Effectiveness Measurements**

Adiabatic wall temperatures were measured and non-dimensionalized in effectiveness (η) levels using an infrared camera. The infrared camera viewed the film-cooled surface through a ZnSe window in the test section ceiling. To ensure that
temperatures were accurately detected for the entire range of surface temperatures, camera output was calibrated to thermocouples on the test plate surface similar to as done by Eberly and Thole [55]. The calibration was applied to infrared images to obtain the adiabatic wall temperature. Freestream and coolant temperatures were both averages of multiple thermocouples in the respective locations (mainstream, and coolant plenum 2.5D below entrances to film cooling holes).

**Thermal Field Measurements**

The time-mean thermal field ($\theta$) was measured using a specially designed thermocouple rake. The thermal field was measured in three planes: the centerline x-y plane and the y-z crossplanes at x/D = 4 and 10. Thermocouples on the rake had wire diameter of 0.05 mm and junction diameter of approximately 0.12 mm. Heat leak down the wires was minimized based on a conductive analysis of the probes. Error in thermocouple readings due to heat leak along wires was estimated to be less than $|\Delta \theta| = 0.05$, based on numerical simulations of a 1-D thermocouple model. Temperatures were the time-mean of at least 36,000 samples taken over at least 30 seconds for each measurement location, which was verified in each case to provide converged values of $\theta$.

**Flowfield Measurements**

Particle image velocimetry (PIV) was used to measure flowfields in two planes of the flow: the centerline x-y plane and the y-z crossplane at x/D = 4. These planes and their respective PIV setups are illustrated in Figures 4.2a-b. Note that PIV measurements were made in the same manner as previously reported [73]. For both setups in the present study, mainstream and coolant were equally seeded with di-ethyl-hexyl-sebacat (DEHS) droplets that followed the flow due to their low Stokes number (maximum Stk = 0.010). A dual-head Nd:YLF laser illuminated the respective measurement planes and high-speed CMOS cameras captured image pairs of the illuminated particles.

For PIV in the centerline plane, images were obtained using a single camera viewing normal to measurement plane as shown in Figure 4.2a. Image pairs were recorded at 4 kHz with image size of 1024x256 pixels. Time-mean flowfields were calculated over at least 8000 time instants spread over a period of 2 seconds which corresponded to more than 240 flow crossings of the PIV field of view (x/D = -2 to 8.6). Time delay between
laser pulses was chosen to provide particle displacements around 8 pixels in the image pairs. Particle displacements, and thereby velocities, were calculated with commercial software [65] using a multi-pass scheme of interrogation windows ending with final window size of 16x16 pixels and 75% overlap. This final window size corresponded to 0.18D x 0.18D since spatial resolution was 11.6 pixels/mm. Background subtraction, intensity normalization, and universal outlier detection were all implemented.

A stereo PIV setup was used to measure flowfields in the x/D = 4 crossplane. This setup, shown in Figure 4.2b, used perspective views from two cameras to measure all three components of velocity. Scheimpflug lens adapters brought the measurement plane into focus for each camera. Image pairs were recorded at 250 Hz on each camera with image size of 1024x512 pixels. Time-mean flowfields were calculated over at least 4000 time instants spread over a period of 16 seconds. Time delay between laser pulses was set between 22-26 µs to obtain particle displacements around 4 pixels in the dewarped images used to compute vectors. Velocities were calculated [65] using a multi-pass scheme of interrogation windows ending in 32x32 pixel interrogation windows with 50% overlap. Final window size corresponded to 0.19D x 0.19D since the spatial resolution was 21.9 pixels/mm. Intensity normalization and universal outlier detection were used for calculating velocities in the x/D = 4 crossplane.
Uncertainty Analysis

Uncertainty was calculated for all measurements. Propagation of uncertainty was estimated using the partial derivatives method of Figliola and Beasley [56] and values are reported for a 95% confidence interval. Regarding test conditions, maximum uncertainty in density ratio was ±0.04. For blowing ratio, uncertainty was dominated by bias uncertainty of the Venturi flowmeter and variation during thermal field measurements. Uncertainty was higher at lower blowing ratios. Maximum uncertainty in \( M = 1.5 \) was ±6.5%.

For adiabatic effectiveness, uncertainty was driven by uncertainty in the coolant temperature and in the plate surface temperature. Adiabatic effectiveness uncertainty was calculated to be \( \delta \eta = \pm 0.025 \). Analogously, major contributors to thermal field uncertainty were the uncertainties in coolant temperature and in temperature measured by the thermocouple rake, as well as heat leak down the thermocouple wires. Including a representative heat leak error of \( \Delta \theta = -0.035 \), maximum uncertainty was calculated to be \( \delta \theta = \pm 0.048 \). Repeatability checks of thermal field profiles agreed within this uncertainty, as shown in Figure 4.3.

![Figure 4.3](image_url)

Figure 4.3. Thermal field profiles showing repeatability for two streamwise locations in the centerline plane. Data is for shaped holes of a separate study at \( Tu_{x} = 0.5\% \), \( M = 3.0 \).
For PIV flowfield measurements both the bias and precision uncertainties were considered. Bias uncertainties were estimated by assuming a displacement bias uncertainty of ±0.15 pixels, which translated to velocity bias uncertainty of ±1.9% in the centerline plane and ±4.8% in the x/D = 4 crossplane. Precision uncertainties were estimated from repeatability tests in the centerline plane and were combined with bias uncertainty to estimate overall uncertainties in individual components of mean velocity. Overall uncertainty of U was estimated to be ±4.5% and ±6.3% in the centerline plane and crossplane, respectively, with percentages based on U∞. Overall uncertainty of V (and of W) was similarly estimated to be ±2.5% and ±5.1% in those same respective planes. Repeatability tests in the centerline plane were also used to estimate precision uncertainties for rms velocities and turbulent shear stresses. Uncertainties were estimated to be ±4% for u′, ±4% for v′ and w′, and ±5% for turbulent shear stresses such as \( \overline{u'v'} \). For rms velocities and shear stresses, percentages were based on maximum magnitude observed with each respective variable.

### 4.4 Results and Discussion

First presented is thermal field and flowfield data for the low freestream turbulence intensity case of \( T_u = 0.5\% \). Comparison is made between apparent position of the jet, as indicated by flowfields, and time-mean height of the coolant core, as indicated by maximum θ. The comparison is especially of interest at M = 3 where peaks in mean velocity occurred significantly above the wall.

Subsequently, thermal field and then flowfield data is presented for the high freestream turbulence intensity case of \( T_u = 13.2\% \). As will be shown, elevated freestream turbulence increased the dilution and dispersion of coolant. Dispersion was primarily in the lateral direction and not in the vertical direction, a behavior observed in thermal fields by Cutbirth and Bogard [72] but not by Kohli and Bogard [59]. To explain the preferential lateral spreading from high freestream turbulence, flowfield data is examined in the region shortly downstream of the shaped hole.
Low Freestream Turbulence Intensity Results

Thermal fields in the centerline plane at blowing ratios of $M = 1.5$ and $3$ and $Tu_{∞} = 0.5\%$ are presented Figures 4.4a-b. At $M = 1.5$ the time-mean coolant jet remained on the wall throughout the measurement domain, reaching a height of $y/D = 2.1$ at 30 diameters downstream of the hole centerline breakout. At $M = 3$ the coolant jet was also initially attached to the wall but exhibited maximum coolant concentration off the wall starting at $x/D = 14$. At $x/D = 30$ the top of the jet had reached a height of $y/D = 3.1$ and highest $\theta$ occurred at $y/D = 1$, off the wall.

![Thermal field contours](image)

**Figure 4.4.** Thermal field contours in the $z/D = 0$ centerline plane at $Tu_{∞} = 0.5\%$ for blowing ratios of (a) $M = 1.5$ and (b) $M = 3.0$.

Such delayed liftoff has not been previously reported in literature for either shaped holes ([21]) or cylindrical holes ([7]). The diffused outlet of shaped holes generally promotes jet attachment since the outlet slows coolant relative to velocity which would occur at a cylindrical hole exit. Past shaped hole studies, however, have still shown presence of a counter-rotating vortex pair (CRVP) that lifts coolant away from the wall [73]. In Figure 4.4b, coolant liftoff was delayed to $x/D = 14$ because forward expansion of the outlet ($β_{fwd}$) lowered the effective injection angle and because coolant was laterally spread over the flat trailing edge of the hole. The CRVP brought mainstream fluid underneath sides of the coolant jet but downstream distance was required for this mainstream incursion to reach the $z/D = 0$ centerline. Kohli and Bogard [21] did not observe such liftoff in their thermal field measurements with shaped holes, likely due to the low blowing ratio ($M \leq 0.8$) and no measurements being downstream of $x/D = 10$.

Vertical profiles of thermal and flowfield variables at the $M = 1.5$ and $3$ conditions are shown in Figures 4.5a-h. Data is from two stations in the centerline plane, $x/D = 1$ and 8.
Figure 4.5. Thermal field and flowfield profiles at $T_u = 0.5\%$ for $M = 1.5$ and $3$ in the centerline plane at (a-d) $x/D = 1$ and (e-h) $x/D = 8$.

For each instance, dotted horizontal lines across the plots denotes height at which coolant concentration fell below $\theta = 0.05$, an indicator of the coolant jet height. Also, for the $x/D = 8$ station which was downstream of the hole, measured adiabatic effectiveness ($\eta$) is plotted with $\theta$, showing good agreement between the measures.

At $x/D = 1$ in Figures 4.5a-d the jet was exiting the shaped hole diffused outlet. Consequentially, the vertical temperature ($\theta$) profile extended only slightly higher at this position for the $M = 3$ jet as compared to the $M = 1.5$ jet. Clear differences between $M = 1.5$ and $3$ were seen in the profiles of mean streamwise velocity. At $M = 1.5$ the jet exited the hole with velocity lower than that of the freestream, while at $M = 3$ the jet had greater velocity than the freestream. Note the lines of $\theta = 0.05$ intersected mean velocity profiles approximately where the velocities returned to values consistent with the approach boundary layer. Such correspondence between time-mean temperature and velocity was expected, since path of the coolant was dictated by the mean flowfield. Figures 4.5c and 4.5d show profiles of rms velocity fluctuations. It was difficult to judge jet height from these rms profiles due to their smooth, gradual changes with $y/D$ height above the wall.
The profiles at x/D = 8 emphasize the same points and are shown in Figures 4.5e-h. Thermal field profiles show that the M = 3 jet extended higher into the mainstream than the M = 1.5 jet. For M = 3 both the thermal field reached θ = 0.05 and the mean velocity asymptotically reached U/U∞ = 1 at the same height, y/D = 2. This agreement for the top edge of the jet again was consistent with how coolant was transported primarily by mean-flow convection. Similar correspondence was hard to ascertain at M = 1.5 because U/U∞ exhibited a monotonic profile. Profiles of rms velocity varied even more gradually than at x/D = 1 and therefore it was not feasible to relate these rms profiles to the θ = 0.05 heights.

While θ and U/U∞ profiles agreed on vertical extent of the coolant jet, the profiles did not agree on apparent position of the coolant jet “core”. At x/D = 1 and 8 the coolant was most concentrated at y/D = 0 as seen in the thermal field profiles of Figures 4.5a and e. By contrast, peaks in mean streamwise velocity in Figures 4.5b and f occurred off the wall. Therefore, flowfield data indicated vertical extent of the region containing the coolant jet but provided little information on relative coolant concentrations within this region.

High Freestream Turbulence Intensity Results

Thermal fields were measured for the blowing ratio M = 3 at Tu∞ = 13.2%. Contours in the centerline plane are shown in Figure 4.6. For reference, dashed lines show contour levels of θ = 0.05, 0.40, and 0.60 from the corresponding M = 3 low freestream turbulence case which was shown in Figure 4.4b.

Contours of θ were significantly different between the low and high freestream turbulence cases. At high freestream turbulence the jet was more diluted than at low freestream turbulence, apparent from how θ = 0.40 and θ = 0.60 contour levels extended shorter distances downstream for Tu∞ = 13.2% as compared to the Tu∞ = 0.5% case.

Figure 4.6. Thermal field contours in the z/D = 0 centerline plane for M = 3.0 at Tu∞ = 13.2%. Labeled dashed white lines denote contour levels of θ_{LFST} = 0.05, 0.40, and 0.60 for the corresponding case at low freestream turbulence.
However, there was negligible change in location of the $\theta = 0.05$ level representing the top of the coolant jet.

Also apparent in Figure 4.6 is that liftoff did not occur in the high freestream turbulence case. Absence of the delayed liftoff was due to strong dilution of coolant caused by freestream turbulence, rather than due to a significant change in dynamics of the flowfield. Schroeder and Thole [73] previously showed that there were no significant differences in mean velocities between the low and high freestream turbulence cases. The CRVP was of similar size and had similar velocities between the $Tu_\infty = 0.5\%$ and 13.2\% cases. Absence of detachment can be understood as follows. At low freestream turbulence intensity, the jet stayed coherent for long distances downstream and therefore the CRVP eventually brought mainstream fluid beneath the core of the coolant jet. Conversely, the high freestream turbulence intensity caused aggressive mixing between coolant and the mainstream, especially in the top and middle of the coolant jet where turbulent fluctuations were not damped by the wall. By the downstream position of $x/D = 14$ the jet was more dilute and less coherent than at $Tu_\infty = 0.5\%$. The CRVP was not able to bring appreciably-warmer fluid beneath the already-dilute jet, so liftoff was not seen.

Vertical profiles of the thermal field and flowfield are plotted in Figures 4.7a-h, similar to Figures 4.5a-h given for the low freestream turbulence cases. Trends were similar to those seen with low freestream turbulence intensity. The thermal field reached $\theta = 0.05$ and streamwise velocity approached mainstream values at matching $y/D$ heights, indicating that vertical extent of the jet at $Tu_\infty = 13.2\%$ was still driven by the mean flowfield. Shape of the $\theta$ and $U/U_\infty$ profiles again did not match, since highest $\theta$ occurred at the wall. Profiles of rms velocity fluctuations featured elevated values due to the turbulent freestream, and therefore rms velocity profiles did not sharply distinguish the jet/mainstream interface.

Thermal fields in the $x/D = 10$ lateral crossplane are compared between the low and high freestream turbulence cases in Figures 4.8a-b. For both these $M = 3$ cases the thermal field contours were widest above the wall, not at the wall. Therefore, tracing downward in $y/D$ the contour lines bent inward in the region nearest the wall, caused by the CRVP which gradually brought hot mainstream fluid underneath sides of the coolant jet. Just as Kohli and Bogard [21] did not observe delayed liftoff with their shaped holes, they also did not
Figure 4.7. Thermal field and flowfield profiles at $T_{u_\infty} = 13.2\%$ for $M = 3$ in the centerline plane at (a-d) $x/D = 1$ and (e-h) $x/D = 8$. Legend is the same as in Figures 4.5a-h.

Observe $\theta$ contour lines bending inward at the wall at the $x/D = 10$ crossplane or elsewhere. The difference between their study and the present study is again attributed to their lower flow rate, $M \leq 0.8$, accompanied by a weaker CRVP.

Figures 4.8a-b show that elevated freestream turbulence caused increased dilution but also caused increased lateral dispersion of the coolant. For the low freestream turbulence case shown in Figure 4.8a, the highest coolant concentration was at the wall with maximum $\theta = 0.61$. The $\theta = 0.05$ contour level extended between $z/D = \pm 1.9$. Figure 4.8b shows that elevated freestream turbulence diluted the coolant such that maximum $\theta = 0.51$. Elevated freestream turbulence increased the lateral extent of $\theta = 0.05$ to $z/D = \pm 2.2$. In agreement with Figure 4.6, the high freestream turbulence caused little increase in how high the $\theta = 0.05$ contour level extended above the wall.

The prevalence of dilution and of lateral dispersion with elevated freestream turbulence is illustrated by select adiabatic effectiveness contours in Figure 4.9. Dashed lines represent $\eta$ contour levels for the low freestream turbulence case, while solid lines represent $\eta$ at high freestream turbulence. High freestream turbulence quickly penetrated coolant jets and diluted the jet core, as shown by how $\eta = 0.60$ levels extended shorter
Figure 4.8. Thermal field contours in the x/D = 10 crossplane at M = 3.0 with freestream turbulence intensities of (a) $T_u = 0.5\%$ and (b) $T_u = 13.2\%$.

Figure 4.9. Comparison of lateral spreading of M = 3.0 jets at freestream turbulence of $T_u = 0.5\%$ (dashed lines) and $T_u = 13.2\%$ (solid lines) through plotting of $\eta = 0.05$ and 0.60 adiabatic effectiveness levels.
distances downstream of holes for the $\text{Tu}_\infty = 13.2\%$ case. At the same time, the high freestream turbulence increased spreading at the sides of the coolant jet. For these shaped holes spaced at $P/D = 6$, the $\eta = 0.05$ levels merged between adjacent jets by $x/D = 15$ for the $\text{Tu}_\infty = 13.2\%$ case.

Lateral Dispersion at High Freestream Turbulence Intensity

Thermal fields presented above showed that increased freestream turbulence caused an increase in lateral dispersion of coolant but negligible increase in vertical dispersion. To illuminate mechanisms causing the difference between lateral and vertical dispersion, flowfields were examined in the $x/D = 4$ crossplane that was $1.6D$ downstream of the shaped hole trailing edge. Figures 4.10a-f show contours of rms velocity fluctuations at $M = 3$, comparing $u'$, $v'$, and $w'$ between the low and high freestream turbulence cases. Gray arrows overlaying the contours show the in-plane mean velocity, revealing the counter-rotating vortex pair that was present. Also overlaid is the $\theta = 0.05$ level representing the perimeter of the coolant jet. These perimeters, and $\theta$ contours overall (not shown for brevity), were similar in this $x/D = 4$ crossplane between the low and high freestream turbulence cases. Similar thermal fields here were expected, due to the crossplane being only a short distance downstream of the hole breakout.

Contours at low freestream turbulence intensity are plotted in Figures 4.10a-c. With low freestream turbulence the jet had higher turbulence intensity than the surrounding

![Figure 4.10. Contours of rms velocity fluctuations at $M = 3.0$ in the $x/D = 4$ crossplane with (a-c) $\text{Tu}_\infty = 0.5\%$ and (d-f) $\text{Tu}_\infty = 13.2\%$. In-plane mean velocity is shown by gray arrows. The thermal field $\theta = 0.05$ contour is shown by the black curve.](image)
mainstream, resulting in regions of high turbulent fluctuations being well-contained within the \( \theta = 0.05 \) contour level. The only exception was that high levels of \( v'/U_\infty \) extended slightly beyond the top of the \( \theta = 0.05 \) contour level, indicating that coolant reached this height but was so dilute that concentration was below \( \theta < 0.05 \).

Contours at high freestream turbulence intensity are plotted in Figures 4.10d-f. For the streamwise and lateral velocity components, fluctuations surrounding the coolant jet increased to \( u'/U_\infty = w'/U_\infty = 0.14 \) with this case of \( T_{u_\infty} = 13.2\% \). Vertical velocity fluctuations also increased, reaching \( v'/U_\infty = 0.10 \) at the top of the coolant jet. Tracing slightly down sides of the coolant jet, \( u'/U_\infty \) and \( w'/U_\infty \) remained large but \( v'/U_\infty \) decreased due to the damping influence of the wall. Tracing down further, to the wall at \( y/D = 0 \), one finds that the minimum values of \( w'/U_\infty \) in Figure 4.10f occurred where the \( \theta = 0.05 \) jet periphery met the wall. Lateral fluctuations here were \( w'/U_\infty = 0.09 \), similar to \( v'/U_\infty \) at the top of the coolant jet. This near-wall region of minimum \( w'/U_\infty \) was due to acceleration of the lateral velocity component, which suppressed \( w' \) fluctuations. Acceleration was caused by the CRVP which brought mainstream fluid (having low-magnitude lateral velocity) into the high-lateral-velocity region comprising the bottom section of the CRVP. While lateral fluctuations were damped by this phenomenon, lateral fluctuations were still greater than the \( w'/U_\infty = 0.05 \) present at the same location with \( T_{u_\infty} = 0.5\% \) (Figure 4.10c).

Comparable magnitude between \( v' \) at the coolant jet top and \( w' \) at coolant jet sides was a significant difference from the study by Cutbirth and Bogard [72], performed with compound-angle cylindrical holes on a vane pressure side. Their thermal fields showed, as in the present study, that high freestream turbulence caused increased lateral dispersion but not vertical dispersion of coolant. Cutbirth and Bogard attributed this to dominance of the lateral fluctuations. Acceleration through the vane passage caused lateral fluctuations surrounding their coolant jet to be \( w'/U_\infty = 0.20 \), measured in a plane analogous to the \( x/D = 4 \) crossplane of the present study. By comparison, other components were \( u'/U_\infty = v'/U_\infty = 0.12 \) around their coolant jet.

In the present study, turbulent shear stresses in the same \( x/D = 4 \) crossplane provide insight into why high freestream turbulence caused primarily lateral spreading of coolant, instead of both lateral and vertical spreading. Figures 4.11a-c show normalized contours of \( \bar{u}'\bar{v}' \), \( \bar{u}'\bar{w}' \), and \( \bar{v}'\bar{w}' \) in the crossplane with the \( M = 3 \) jet at \( T_{u_\infty} = 0.5\% \). Figures 4.11d-f
show the corresponding contours at $T_u = 13.2\%$. Mean in-plane velocities and the $\theta = 0.05$ jet periphery are overlaid as in Figures 4.10a-f. Figures 4.11a,d compare $\bar{u}'v'$ between the low and high freestream turbulence cases. For both freestream turbulence intensities, the upper half of the $\theta > 0.05$ region had positive $\bar{u}'v'$ consistent with momentum transport by turbulent viscosity. Packets of fluid originating at the center of the jet had high $u$-velocities. Such packets which had higher-than-average vertical velocity brought fast-moving $M = 3$ coolant toward the top boundary of the jet, contributing to positive $\bar{u}'v'$. The region of positive $\bar{u}'v'$ decayed toward zero before encountering the $\theta = 0.05$ periphery, indicating that packets of concentrated coolant rarely made excursions far above the jet. The region of positive $\bar{u}'v'$ had similar magnitude and similar vertical extent between the $T_u = 0.5\%$ and $13.2\%$ cases, showing that large eddies of the $13.2\%$ freestream turbulence did little to modify dispersion occurring at the top of the coolant jet.

Freestream turbulence did make a difference in dispersion at sides of the coolant jet, as contours of $\bar{u}'w'$ show. In Figure 4.11b the regions of negative and positive $\bar{u}'w'$ at the respective left and right sides of the jet were contained within the $\theta > 0.05$ region, just as with positive $\bar{u}'v'$ in Figure 4.11a. Such behavior was expected for low freestream turbulence, since turbulent fluctuations associated with the jet would not extend beyond the well-defined jet/mainstream shear layer. The patterns were different for the $T_u = 13.2\%$ case in Figure 4.11e. With high freestream turbulence, magnitudes of $\bar{u}'w'$ were increased.
in the left and right regions relative to the $\text{Tu}_\infty = 0.5\%$ case. The left and right regions also extended laterally beyond the $\theta = 0.05$ periphery, to $z/D = -2.3/2.6$. Note that lateral extent was greatest at $y/D = 0$. Wider $\bar{u}'\bar{w}'$ patterns with $\text{Tu}_\infty = 13.2\%$ were due to instances where the coolant jet was swept by downward-moving eddies from the freestream turbulence. When these eddies started to feel impaction with the wall they deflected laterally, thereby contributing to much-wider regions of non-zero $\bar{u}'\bar{w}'$ than seen at $\text{Tu}_\infty = 0.5\%$. This influence of the wall on freestream eddies was the reason why elevated freestream turbulence preferentially augmented the lateral dispersion of coolant as compared to vertical dispersion. Unlike at the top of the jet, at sides of the jet the eddies from freestream turbulence did indeed modify the dispersion behavior.

The third turbulent shear stress, $\bar{v}'\bar{w}'$, is compared between low and high freestream turbulence cases in Figures 4.11c,f. Non-zero regions of $\bar{v}'\bar{w}'$ occurred beyond lateral sides of the coolant jet at $\text{Tu}_\infty = 13.2\%$ but not at $\text{Tu}_\infty = 0.5\%$, which further confirmed that the wall was preferentially directing turbulent eddies laterally.

Increased lateral dispersion with high freestream turbulence was mainly due to influence of the wall on eddies. However, changes in the distribution of turbulence within the jet caused dilution of coolant that was also a contributing factor to dominance of lateral dispersion. Figures 4.12a-b compare turbulence intensity measured in the centerline plane, overlaid with contours levels of $\theta$, at $M = 3.0$ with freestream turbulence of (a) $\text{Tu}_\infty = 0.5\%$ and (b) $\text{Tu}_\infty = 13.2\%$.

![Figure 4.12. Contours of turbulence intensity in the centerline plane, overlaid with contours levels of $\theta$, at $M = 3.0$ with freestream turbulence of (a) $\text{Tu}_\infty = 0.5\%$ and (b) $\text{Tu}_\infty = 13.2\%$.](image)
plane at $T_{u_{\infty}} = 0.5\%$ and $13.2\%$. Thermal field contours at intervals of $\Delta \theta = 0.10$ are overlaid, with the $\theta = 0.05$ level included as well. Outside the jet, Figures 4.12a-b show increased turbulence with the elevation of freestream turbulence intensity to $T_{u_{\infty}} = 13.2\%$. Within the jet, the region where turbulence intensity increased most was the upper portion of the developed jet at several diameters downstream of the hole (past $x/D = 4$). This increase is evident, for example, from tracing upward at $x/D = 7$: in Figure 4.12a the turbulence intensity decreases as one traces from the lower to the upper portion of the jet, while in Figure 4.12b the turbulence intensity begins one contour level higher than at $T_{u_{\infty}} = 0.5\%$ but then remains high in the upper portion of the jet. The higher turbulence intensity in the upper part of the jet for the $T_{u_{\infty}} = 13.2\%$ case was not accompanied by significant change in low $\theta$ contour levels representing the top edge of the jet. (Note that $\theta$ did decrease in regions closer to the wall with the elevation of freestream turbulence intensity.) Elevated freestream turbulence caused mixing in the jet such that packets of cooler fluid dispersed upward were already dilute and therefore did not contribute to increased height of the $\theta = 0.05$ level from the $T_{u_{\infty}} = 0.5\%$ to the $13.2\%$ case. A similar situation of unchanging-$\theta$ at the jet periphery was observed in the high freestream turbulence study of Kohli and Bogard [59] for a position $3D$ downstream of their cylindrical holes.

With elevation of freestream turbulence intensity from $0.5\%$ to $13.2\%$, the constant $\theta$ at the top of the jet and the constant $\bar{u}'v'$ contained within the jet (Figures 4.11a,d) were related effects. At top of the coolant jet, instantaneous coolant concentrations and instantaneous fluid packets bearing high $u$-velocities representative of the jet were so dilute that they both made little contribution in the time-mean, even when freestream eddies swept the coolant upward out of the jet. Therefore, the high freestream turbulence caused increased lateral spreading of coolant without similar vertical spreading for two reasons: 1) eddies near the wall were preferentially directed laterally and 2) coolant subjected to these lateral fluctuations was more concentrated than coolant dispersed vertically out the top of the jet.
4.5 Conclusions

Thermal fields were measured for jets from shaped holes at high blowing ratios, at low and high freestream turbulence intensities of 0.5% and 13.2%. With low freestream turbulence the jets penetrated farther into the mainstream with increased blowing ratio. At M = 3, the highest blowing ratio tested, the jet exiting the hole was attached to the wall but farther downstream the jet lifted away from the wall due to action of the counter-rotating vortex pair.

The delayed liftoff observed at low freestream turbulence intensity was not observed with high freestream turbulence, due to strong dilution of the coolant jet by freestream turbulence. The main effects of high freestream turbulence were dilution of the coolant jet and increased lateral dispersion of coolant, as seen in thermal field and adiabatic effectiveness measurements. Thermal fields showed that high freestream turbulence caused little increase in vertical dispersion of the coolant jets.

Flowfields measured at the same conditions were compared to thermal field measurements but correlations were not found between flowfield variables and the distribution of coolant within jets. Turbulent shear stresses however did reveal reasons for the preferential lateral dispersion of coolant by elevated freestream turbulence. While the elevated turbulence did impose similar vertical and lateral velocity fluctuations, coolant was dispersed more in the lateral direction because of low coolant concentration at the top of the jet and because turbulent eddies containing the more-concentrated coolant were preferentially directed laterally by the wall.

The present study provides new information on dynamics of film cooling from shaped holes. Experimental data on phenomena such as delayed liftoff of jets and near-wall strengthening of lateral dispersion can lead to improved correlations and turbulence models used by designers for predicting shaped hole performance.
Chapter 5.

Effect of In-Hole Roughness on Film Cooling From a Shaped Hole

Abstract

While much is known about how macro-geometry of shaped holes affects their ability to successfully cool gas turbine components, little is known about the influence of surface roughness on cooling hole interior walls. For this study a baseline shaped hole was tested with various configurations of in-hole roughness. Adiabatic effectiveness measurements at blowing ratios up to three showed that in-hole roughness caused decreased adiabatic effectiveness relative to smooth holes. Decreases in area-averaged effectiveness grew more severe with larger roughness size and with higher blowing ratios for a given roughness. Decreases of more than 60% were measured at a blowing ratio of three for the largest roughness values. Thermal field and flowfield measurements showed that in-hole roughness caused increased velocity of core flow through the hole, which increased the jet penetration height and turbulence intensity resulting in increased mixing between coolant and the mainstream. Effectiveness reductions due to roughness were also observed when roughness was isolated to only the diffused outlet of holes, and when the mainstream was highly turbulent.

5.1 Introduction

Film cooling is a critical technology for maintaining acceptable metal temperatures along the hot gas path in gas turbines. Shaped holes are commonly used instead of cylindrical holes, due to the diffused outlet of shaped holes which slows coolant jets and laterally spreads coolant over the vulnerable surfaces. Many attributes influence the performance of shaped holes, including inclination angle, hole length, expansion angles of the diffused outlet, pitchwise spacing, coverage ratio, area ratio, and breakout shape, not to mention the influence of coolant blowing ratio and flow conditions outside the hole [9].

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Several studies have examined the sensitivity of shaped hole performance to variation of these parameters [8,11,12,21,26,74].

Not yet addressed in literature is the influence of surface roughness along interior walls of film cooling holes, despite the fact that in-hole roughness varies with manufacturing method. Laser drilling, traditionally used for cylindrical holes and increasingly used in trepanning-fashion for shaped holes [75], is regarded to deliver non-smooth surfaces. Details in literature are sparse, although Bunker [75] has described laser drilling as producing in-hole roughness levels severe enough to preclude diameter measurement with pin gages, resulting in a hole diameter tolerance of ±10% which would translate to a roughness of $R_a/D = 0.020$. By contrast, electrode discharge machining (EDM) can produce smoother shaped holes, for instance delivering in-hole surface roughness of $R_a/D = 0.0025$ [75]. Water jet drilling is another method that can similarly produce “very clean and tailored” shaped holes [75].

Roughness is equally a consideration in the laboratory environment. Film cooling geometries are often scaled-up and manufactured in foam for wind tunnel experiments, with little documentation of in-hole surface roughness. As will be shown in this study, in-hole roughness can have significant effect on film cooling performance and therefore is worthy of consideration by experimentalists and designers.

5.2 Previous Studies

Few studies in open literature investigate the influence of roughness inside film cooling holes. Some authors have noted potential benefit from large-scale “rifling” in film cooling holes. Thurman et al. [76] tested cylindrical holes with and without rifled grooves of groove depth 0.2D at blowing ratios ranging from $M = 1.0$ to 2.5. They observed higher laterally-averaged effectiveness with axially-oriented grooved holes than for non-grooved axial and compound-angle holes. Rifling in film cooling holes is also mentioned in patents by Tibbott and Harrogate [77] and Strock and Lutjen [78]. Absent from literature are studies on the effect of irregular in-hole roughness, especially roughness of size typical to industrial and aero engines $0.002 < R_a/D < 0.02$.

The present study provides a foundation for addressing this need in literature. Measurements of adiabatic effectiveness, flowfields, and thermal fields are presented for
shaped holes with a variety of in-hole roughness configurations up to size $R_a/D = 0.02$. The external surface was kept aerodynamically smooth for all tests. In-hole roughness was distributed across hole metering sections and/or diffused outlets.

### 5.3 Experimental Facility and Methods

The closed loop wind tunnel shown previously shown in Figure 3.1 was used for film cooling experiments. Mainstream air was driven by an in-line centrifugal fan such that mainstream velocity in the test section was 10 m/s. Mainstream temperature was maintained at 295 K by a chilled water heat exchanger and a bank of electric heaters.

Film cooling measurements were made for a single-row of five shaped holes located on the flat plate comprising the floor of the test section. Mainstream air entered the test section by flowing through a 6:1 contraction and then passed over the boundary layer suction slot and the plate leading edge. The suction slot removed the boundary layer of the contraction floor, thus starting the boundary layer afresh at the plate leading edge. A boundary layer trip at $x/D = -33$ then initiated transition of the boundary layer to a turbulent state. At the shaped hole location the boundary layer was fully turbulent, as shown in previously-reported flowfield measurements [73].

Without any turbulence grid in place upstream of the holes, freestream turbulence intensity at $x/D = -2$ was $T_u = 0.5\%$. Tests were performed at this low freestream turbulence condition and also at a high freestream turbulence condition of $T_u = 13.2\%$. The high freestream turbulence was obtained using a grid of vertical bars upstream of the holes. Bar diameter was $b = 38$ mm and bars were spaced apart $2b$ center-to-center. This turbulence grid was installed upstream of the holes at $x/b = -14$. Length scale of the high freestream turbulence was measured with hotwire to be $\Lambda_x = 5.2D$ at the shaped hole trailing edge. The mainstream approach boundary layer was characterized at $x/D = -2$ for both low and high freestream turbulence conditions. Characteristics of the boundary layers are given in Table 5.1.

#### Table 5.1. Boundary Layer Characteristics

<table>
<thead>
<tr>
<th>$T_u$</th>
<th>$\delta_2/D$</th>
<th>$H$</th>
<th>$Re_{\delta_2}$</th>
<th>$c_f/c_{f,0}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5%</td>
<td>0.14</td>
<td>1.45</td>
<td>670</td>
<td>1.0</td>
</tr>
<tr>
<td>13.2%</td>
<td>0.12</td>
<td>1.38</td>
<td>580</td>
<td>1.19</td>
</tr>
</tbody>
</table>
An auxiliary closed loop was used to supply coolant flow to the row of shaped holes, as shown at the bottom of Figure 3.1. Air used for coolant was removed from the wind tunnel far upstream of the test section with a hermetically-sealed blower. The coolant air was chilled with liquid nitrogen in a heat exchanger and then flowed through a Venturi flowmeter to the coolant plenum, where fine screens ensured uniform approach flow to entrances of the five shaped holes. The coolant-to-mainstream density ratio used in this study was DR = 1.5. To prevent undesired buildup of frost during experiments, air in the closed loop of the wind tunnel was dried prior to experiments by circulating it through a desiccant vent dryer. Also, during experiments the tunnel was kept positively pressurized with nitrogen gas.

Shaped holes used in this study were those introduced by Schroeder and Thole [10]. The holes had a cylindrical metering section that expanded at 7º angles in the forward and two lateral directions to form the diffused outlet. Exit-to-inlet area ratio was AR = 2.5. Full specification of this geometry, including CAD files, is publicly available for download at the authors’ website (http://www.mne.psu.edu/psuturbine). The shaped hole geometry is illustrated in Figure 5.1 and geometric parameters of the hole are listed in Table 5.2.

All shaped holes for this study were machined in polystyrene that formed the flat plate floor of the test section. The low thermal conductivity of polystyrene, $k = 0.029 \text{ W/m} \cdot \text{K}$, made it ideal for adiabatic effectiveness and thermal field measurements. Nominal metering diameter was $D = 7.75$ mm for all the shaped holes except the “slightly rough hole” specimen which had metering diameter of $D = 8.47$ mm. Normalized values given above for characterization of the mainstream and the approach boundary layers used $D = 7.75$ mm.

![Figure 5.1. Shaped hole geometry.](image-url)
Table 5.2. Geometric Parameters of the Shaped Hole

<p>| | | | | |</p>
<table>
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<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>P/D</td>
<td>6</td>
<td>L_m/D</td>
<td>2.5</td>
<td></td>
</tr>
<tr>
<td>α</td>
<td>30°</td>
<td>L_lat/D, L_fwd/D</td>
<td>3.5</td>
<td></td>
</tr>
<tr>
<td>β_lwd, β_lat</td>
<td>7°</td>
<td>Area Ratio, AR</td>
<td>2.5</td>
<td></td>
</tr>
</tbody>
</table>

Roughness Creation and Characterization

The shaped holes were cut in polystyrene using a three-axis CNC mill. By increasing feed rate and decreasing spindle speed, greater roughness was produced on the in-hole surfaces. The metering section and diffused outlets were cut by separate operations on the CNC mill. The roughness was rigid; visual checks confirmed that roughness elements did not vibrate due to airflow through the hole. It was recognized that roughness elements extending inward from hole walls acted to reduce cross-sectional area, so small offsets were used in the CNC programming with the intent of maintaining equal cross-sectional area between smooth and rough specimens. After machining was completed, hole entrance and breakout dimensions were checked with calipers and found to be within 5% of the design intent. The only exception was the major axis of hole inlets in the “rough diffuser hole” specimen which was up to 9% larger than nominal.

After completing wind tunnel experiments, in-hole roughness was quantified by cutting each specimen apart and measuring surface heights with an optical profilometer. The regions scanned included both the metering section and diffused outlets of holes, indicated with red bold lines in Figure 5.1. The profilometer featured a 1024x1024 pixel camera and various objective lenses. For measurements in the present study, lateral resolution was 0.0002D per pixel and vertical height uncertainty was ±0.001D. For occasional pixels that were too dark to report a height (always less than 22% of the scan), surface heights were Laplace interpolated. Between maps before and after interpolation, R_a was found to change less than 12%.

Surface maps were built by stitching together adjacent 1024x1024 pixel scans. To ensure roughness was measured over a large-enough area, each stitched map was divided into four equal sections and R_a of each section was compared to R_a of the overall map. Maximum disagreement was observed with the “rough hole” specimen, where R_a of quadrants disagreed by up to 0.007D from R_a of their overall stitched map.
Adiabatic Effectiveness Measurements

Adiabatic effectiveness ($\eta$) was determined by measuring temperatures of the polystyrene plate surface through use of an infrared camera. The camera viewed the test plate by looking downward through a ZnSe window installed flush with the test section ceiling. Temperatures were accurately determined through direct calibration of camera output to temperatures measured by thermocouples on the test place surface, similar to as done by Eberly and Thole [55]. The calibration was applied to time-average infrared images to determine adiabatic wall temperature, $T_{aw}$. Freestream and coolant temperatures both were averages, each taken from multiple thermocouples in the respective locations (mainstream, and in plenum 2.5D below film cooling hole entrances).

Thermal Field Measurements

Thermal fields ($\theta$) were measured in three planes: the centerline x-y plane, and two y-z crossplanes located at x/D = 4 and 10. Measurements were made using a specially-designed fine-wire thermocouple rake. Diameter of thermocouple wires was 0.05 mm and diameter of sensing junctions was approximately 0.12 mm. Heat leaks were minimized based on a conductive analysis of thermocouple lead wires and the supporting rake structure. Error in thermocouple readings due to heat leak was estimated to be less than $|\Delta \theta| = 0.05$, based on numerical simulations of a 1-D thermocouple model. Thermal field temperatures were the average of at least 36,000 samples taken over at least 30 seconds, ensuring converged values of time-mean $\theta$.

Flowfield Measurements

Flowfields were measured in the centerline x-y plane using particle image velocimetry (PIV). For tracer particles, the mainstream and coolant were seeded equally with di-ethyl-hexyl-sebacat (DEHS) droplets that followed the flow due to their low Stokes number (maximum Stk = 0.010). Particles in the centerline plane were illuminated with a dual-head Nd:YLF laser and image pairs of the particles were captured with a high-speed CMOS camera. Image pairs were taken at 4 kHz at image size of 1024x256 pixels. Time mean flowfields were calculated by averaging over 8000 image pairs acquired over a period of 2 seconds. Based on mainstream velocity, the 2 seconds corresponded to more than 240 flow crossings of the PIV field of view. Velocities were calculated from image pairs using
commercial PIV software [65] with a multi-pass scheme of interrogation windows that ended with final window size of 16x16 pixels and 75% overlap. This final window size corresponded to 0.18D x 0.18D since spatial resolution was 11.6 pixels/mm. In the course of the velocity calculations, background subtraction, intensity normalization, and universal outlier detection were implemented as previously reported by Schroeder and Thole [73].

**Uncertainty Analysis**

Uncertainty analyses were performed for flowfield, thermal field, and adiabatic effectiveness measurements. Uncertainties were propagated with the partial derivative method described by Figliola and Beasley [56]. Values reported are for a 95% confidence interval. Regarding test conditions, the density ratio of 1.5 had a maximum uncertainty of ±0.04. For blowing ratio the uncertainty was dominated by bias uncertainty of the Venturi flowmeter and also variation during thermal field measurements. For M = 1.5 the maximum uncertainty in blowing ratio was ±6.5%.

For adiabatic effectiveness, uncertainty was driven by uncertainty in the coolant temperature and plate surface temperature. Adiabatic effectiveness uncertainty was calculated to be δη = ±0.025. Analogously, major contributors to thermal field uncertainty were the uncertainties in coolant temperature and temperature measured by the thermocouple rake, along with estimated heat leak down the thermocouple wires. For a moderate thermocouple-rake heat leak error of Δθ = -0.035, maximum uncertainty was calculated to be δθ = ±0.048.

Uncertainties for PIV flowfield measurements in the centerline plane were determined through consideration of both bias and precision uncertainty. Bias uncertainty was based on an assumed ±0.15 pixel displacement uncertainty, yielding a velocity bias uncertainty of ±1.9% of U∞. Precision uncertainty was estimated for U and V based on repeatability tests and was combined with the bias uncertainty to determine overall uncertainty. Overall uncertainty was estimated to be ±4.5% for U and ±2.5% for V, given as percentages of U∞. Repeat tests were also used to estimate precision uncertainties of rms velocity components, determined to be ±4% for both u’ and v’, with percentages based on maximum magnitudes respectively observed for u’ and v’.
5.4 Results and Discussion

This paper focuses on how in-hole roughness affects performance of a shaped film cooling hole. First, in-hole roughness of test specimens is characterized and discussed. Then the effect of roughness size is investigated through adiabatic effectiveness results at blowing ratios ranging from $1 \leq M \leq 3$. Thermal and flow field measurements are used to illuminate physical mechanisms behind the roughness effects, especially at the blowing ratio of $M = 3$ where in-hole roughness had greatest effect. Subsequent sections address the effect of roughness isolated to only the hole diffuser and, finally, the combined effect of elevated freestream turbulence and in-hole roughness.

Roughness Characterization

Film cooling tests were performed on four in-hole roughness specimens shown in Table 5.3. The first three specimens represented a progression of increasing roughness size throughout the metering sections and diffused outlets of holes, while the fourth specimen (rough diffuser hole) had smooth metering sections but roughness similar to that of “rough hole” in the diffused outlets of holes.

<table>
<thead>
<tr>
<th>Specimen Name</th>
<th>Range of $R_a/D$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth Hole</td>
<td>0.003 – 0.004</td>
</tr>
<tr>
<td>Slightly Rough Hole</td>
<td>0.006 – 0.009</td>
</tr>
<tr>
<td>Rough Hole</td>
<td>0.017 – 0.020</td>
</tr>
<tr>
<td>Rough Diffuser Hole</td>
<td>0.016 – 0.021</td>
</tr>
</tbody>
</table>

Table 5.3 includes centerline average roughness ($R_a$) measured for each specimen, normalized by the hole metering diameter. Roughness was calculated from optical profilometry surface maps such as those shown in Figure 5.2. For the first three specimens, the stated ranges of roughness include $R_a/D$ measured in both the metering sections and diffused outlets of holes, while $R_a/D$ for the rough diffuser holes represents roughness only in diffused outlets of holes. Roughness in metering sections of the rough diffuser holes was $R_a/D = 0.003$, similar to roughness in the smooth holes.
Figure 5.2. Three-dimensional maps of roughness in metering sections of the (a) smooth hole and (b) rough hole configurations. Note that overall form of the surface has been removed, as done for roughness calculation.

Effect of Increasing Roughness Size

Adiabatic effectiveness was measured and compared among the different in-hole roughness configurations to investigate the effects of in-hole roughness. Except where noted in a later section, all tests were at low freestream turbulence intensity of $T_u\infty = 0.5\%$.

To investigate the effect of increasing size of in-hole roughness, adiabatic effectiveness was compared between the smooth hole, slightly rough hole, and rough hole specimens. For the blowing ratio of $M = 1.0$, Figure 5.3 shows laterally-averaged adiabatic effectiveness of those three roughness configurations. Only in-hole roughness of the largest size, that of the rough holes, caused a change in laterally-averaged effectiveness at this blowing ratio. At low $x/D$, laterally-averaged effectiveness of the rough holes was 20% lower than for the smooth holes. Contours of adiabatic effectiveness for the $M = 1.0$ cases (not shown for brevity) revealed that decreased effectiveness with the rough holes was due to narrower $\eta$ patterns and slightly-lower centerline effectiveness.
Figure 5.3. Laterally-averaged adiabatic effectiveness at $M = 1.0$ for configurations with increasing size of in-hole roughness.

Greater differences between roughness configurations were seen at higher blowing ratios of $M = 1.5$ and $3.0$. Adiabatic effectiveness contours for these cases and others studied are shown in Figures 5.4a-h. While, for brevity, only the middle hole in the row is shown, all roughness configurations tested had good hole-to-hole periodicity except the rough holes. To ensure representative values are presented even for the rough holes, the laterally- and area-averaged values throughout this study are the average taken over the middle three holes in the row of five.

The $M = 1.5$ cases are compared by examining Figures 5.4a-c which correspond respectively to the smooth, slightly rough, and rough holes. Most striking is the large decrease in effectiveness of the rough hole as compared to the two smoother configurations. The coolant pattern downstream of the rough hole was narrower and had lower centerline effectiveness than occurred with the other two holes having lower roughness levels.

Effectiveness differences at $M = 1.5$ are quantified by laterally-averaged effectiveness shown in Figure 5.5a. The slightly rough holes had effectiveness only slightly less than that of the smooth holes. In contrast, the rough holes experienced decreases in laterally-averaged effectiveness exceeding 45%, relative to the smooth holes. It is interesting to note that near the hole the effectiveness values were similar between all
roughness levels, but farther downstream the effectiveness values diverged from that of the smooth hole.

The most severe decreases in effectiveness occurred at $M = 3.0$. As seen in contours of Figures 5.4e-g, centerline effectiveness progressively decreased with increasing roughness size. Coolant patterns also narrowed with increasing roughness size, especially for the rough hole.

Laterally-averaged effectiveness for $M = 3.0$ is shown in Figure 5.5b. Effectiveness for each roughness configuration was lower at this blowing ratio than the corresponding cases at $M = 1.5$ in Figure 5.5a. At $M = 3$, a greater difference occurred between the smooth and slightly rough holes ($\Delta \bar{\eta} \approx 0.03$) than occurred at lower blowing ratio. Between the smooth and rough holes, the absolute decrease in $\bar{\eta}$ was about the same as at $M = 1.5$. However, this decrease on a percentage basis was worse than at $M = 1.5$ ($65\%$ decrease as compared to $45\%$) since effectiveness of smooth holes was lower at $M = 3$ than at $M = 1.5$.

Trends over the whole range of blowing ratios are depicted with area-averaged effectiveness plotted in Figure 5.6. Area-averaging was done over the region $3D$ to $35D$ downstream of the trailing edge of shaped holes. Effectiveness increased with blowing
Figure 5.5. Laterally-averaged adiabatic effectiveness for different in-hole roughness configurations at (a) $M = 1.5$ and (b) $M = 3$.

Figure 5.6. Area-averaged effectiveness for configurations having increasing size of in-hole roughness, plotted as a function of blowing ratio.
ratio up to $M = 1.5$, with further increases in blowing ratio leading to narrower effectiveness contours on the surface, and thereby lower area-averaged effectiveness. Narrower $\eta$ patterns at high blowing ratio for the baseline case of smooth holes were due to a stronger counter-rotating vortex pair (CRVP) at higher blowing ratios [73], accompanied with increased jet penetration. Figure 5.6 shows that slightly rough holes followed the baseline trend at $M = 0.5$ and 1.0, but developed lower effectiveness than smooth holes at higher blowing ratios. Rough holes showed performance significantly worse than even slightly rough holes. Area-averaged effectiveness for rough holes was 17%, 45%, and 61% lower than smooth hole performance at blowing ratios of $M = 1$, 1.5, and 3, respectively.

**Thermal Field and Flowfield Measurements for Smooth and Rough Holes**

Thermal field and flowfield measurements were made to investigate the mechanism by which in-hole roughness caused reduced effectiveness. In Figures 5.7a-b, thermal fields at $M = 3$ are compared in the crossplane at $x/D = 4$, which was only 1.6D downstream of the hole trailing edge. Also shown are surface adiabatic effectiveness contours leading up to the $x/D = 4$ plane. At $x/D = 4$, Figure 5.7b shows that the jet exiting the rough hole extended higher above the wall and had less lateral spreading than the jet from the smooth hole shown in Figure 5.7a. These differences are consistent with the idea that in-hole roughness caused thicker boundary layers along the interior walls of the shaped hole, causing a higher velocity jet core. It is also interesting to note that thermal field contours for the jet exiting the smooth hole are more consistent with a kidney-shaped cross-section as compared to the contours for the jet exiting the rough hole. The shape of $\theta$ contours at $x/D = 4$ is consistent with the fact that there was a stronger jet core for the rough hole as compared to the smooth hole.

Figures 5.8a-b show thermal field contours for the same cases in the crossplane farther downstream at $x/D = 10$, which was 7.6D downstream of the hole trailing edge. Surface effectiveness contours leading up to the crossplane are again shown. Figure 5.8a shows that the jet exiting the smooth hole had spread to a width of nearly 4D out of the 6D pitch between holes. By contrast, the jet exiting the rough hole in Figure 5.8b exhibited less spreading and had less of a kidney shape than the jet from the smooth hole. Also note
Figure 5.7. Contours of $\theta$ in the $x/D = 4$ crossplane and contours of $\eta$ on the wall ($y/D = 0$) at $M = 3.0$, for the (a) smooth hole and (b) rough hole. To show detail, only pitchwise range $z/D = -2$ to 2 is shown.

that thermal field contours indicate the jet core from the rough hole was off the wall, whereas the jet core for the smooth hole was on the wall.

Thermal fields in the centerline plane for the same $M = 3$ cases are given in Figures 5.9a-b. Centerline contours show that the jet core from the rough hole penetrated higher into the mainstream than the jet core from the smooth hole, indicating that maximum cooling potential was not available to cool the wall for the rough hole case. Thermal fields also show that the jet from the rough hole was diluted more quickly than the jet from the smooth hole (compare extent of the $\theta = 0.5$ contour levels). As discussed below with flowfields, this greater dilution for the rough hole case was attributed to higher turbulence intensity in the jet near the hole breakout.

Mean and turbulent flowfields in the centerline plane for the same $M = 3$ cases are shown in Figures 5.10a-b and 5.11a-b. Mean streamwise velocities plotted in Figures
Figure 5.8. Contours of $\theta$ in the $x/D = 10$ crossplane and contours of $\eta$ on the wall ($y/D = 0$) at $M = 3.0$, for the (a) smooth hole and (b) rough hole. To show detail, only pitchwise range $z/D = -2$ to 2 is shown.

Figure 5.9. Thermal field contours at $M = 3.0$ in the $z/D = 0$ centerline plane for the (a) smooth hole and (b) rough hole.

5.10a-b show higher penetration of the jet from the rough hole as compared to the jet from the smooth hole, which is consistent with the thermal fields. Figures 5.11a-b likewise show higher penetration with the rough hole and also show higher turbulence intensity inside the jet, as compared to the smooth hole case. Near the rough hole breakout, turbulence intensities exceeded 44% whereas for the smooth hole the peak turbulence was 35%.
Figure 5.10. Mean velocity vectors and streamwise velocity contours in the centerline plane at DR = 1.5, M = 3.0 for the (a) smooth hole and (b) rough hole.

Figure 5.11. Contours of turbulence intensity in the centerline plane at DR = 1.5, M = 3.0 for the (a) smooth hole and (b) rough hole.

The higher turbulence intensity at the hole breakout for the rough hole case contributed to increased dilution of coolant.

To illuminate the phenomena by which in-hole roughness caused decreased effectiveness, flowfields just outside the hole outlet were examined for both smooth and rough holes at blowing ratios ranging from M = 1 to 3. While not shown for brevity, for rough holes the values of peak mean velocity and peak turbulence intensity in the jet
increased monotonically with blowing ratio, similar to the behavior seen with smooth holes [73]. Greater difference between smooth and rough hole behavior was seen with penetration angle of flow leaving the diffused outlet. Figure 5.12 shows profiles of penetration angle ($\phi$) of mean velocities along a horizontal line just above the wall, at $y/D = 0.4$. For reference, note that flow parallel to the ceiling and floor of the diffused outlet would have penetration angles of 30° and 23°, respectively. Penetration angles along the $y/D = 0.4$ line were smaller than those values of the hole geometry, remaining below $\phi = 15^\circ$ for all cases.

Tracing downstream, all eight cases showed similar position for the windward edge of the jet as indicated by the sharp increase in penetration angle from $\phi \approx 1^\circ$ of the approaching boundary layer shown in Figure 5.12. For the smooth hole cases at blowing ratios up to $M = 2$, the penetration angle increased from $\phi \approx 1^\circ$ up to a local peak, then plateaued for the region directly above the hole breakout, then exhibited a second local peak penetration angle near the hole trailing edge of $x/D = 2.4$, and finally decreased downstream to approach the wall inclination of $0^\circ$. Penetration angles for the smooth hole $M = 3$ jet were similar, except instead of plateauing over the breakout they exhibited a local minimum above the middle of the breakout.

For these smooth hole cases the two peaks in penetration angle, shown in Figure 5.12, corresponded to how flow exiting along windward and leeward surfaces of the diffused outlet followed the inclination of those surfaces. Slightly-lower penetration angles over the middle of the hole breakout corresponded with coolant inclined slightly more in the downstream direction relative to the hole interior walls. The extra downstream inclination was due to secondary flow in the center of the diffused outlet, which circulated such that coolant in the centerline plane moved from the windward wall toward the leeward wall of the outlet. This secondary flow was significantly stronger for the $M = 3$ case, as evidenced by the local minimum in penetration angle over the middle of the breakout. Stronger secondary flow at $M = 3$ was to be expected, since this blowing ratio was 50% higher than the next highest blowing ratio tested, $M = 2$. Note also that penetration angles downstream of the $x/D = 2.4$ trailing edge were higher for the $M = 3$ case than for the other smooth hole cases, due to the external CRVP that developed downstream of the hole and was strongest for the $M = 3$ case [73].
Figure 5.12. Penetration angle of time-mean velocities along y/D = 0.4 in the centerline plane for smooth holes and rough holes, M = 1.0 to 3.0.

For the rough hole, penetration angles shown in Figure 5.12 had the same values as that of the smooth hole at the jet leading edge, but then greatly increased toward the leeward side of the hole breakout, reaching almost $\phi = 15^\circ$ over the latter half of the hole breakout ($0 < x/D < 2.4$). Increases in the peak penetration angle occurred with increases in blowing ratio for the rough hole. Larger in-hole roughness caused thicker boundary layers along hole interior walls than occurred with the smooth hole, resulting in narrower and faster core flow through the hole. This core flow caused higher penetration angle of injection and caused absence of plateaued regions that were seen in $\phi$ profiles for the smooth hole. The constricted core flow also attenuated secondary flow in the hole, although secondary flow was not eliminated. Notice that, for the M = 3 jet from the rough hole, a local minimum penetration angle again occurred over the breakout, albeit less pronounced than in the smooth hole case.

Overall the thermal field and flowfield measurements show that in-hole roughness primarily affected film cooling performance by causing thicker boundary layers along hole walls, which led to faster core flow relative to that of the smooth hole at the same blowing
ratio. Attendant with the faster core flow was higher penetration of coolant into the mainstream where it did not cool the wall, as well as increased turbulence intensity that diluted coolant with the mainstream.

**Effect of Diffused Outlet Roughness**

Besides the effect of roughness size, the influence of roughness location within the hole was investigated. Adiabatic effectiveness was measured for the rough diffuser hole described in Table 5.3, whereby roughness levels were similar to the rough hole geometry.

Contours of adiabatic effectiveness for the rough diffuser hole at M = 1.5 and 3.0 are given above in Figures 5.4d and 5.4h, respectively. For the M = 1.5 case, adiabatic effectiveness performance was between that of slightly rough holes (Figure 5.4b) and rough holes (5.4c), both in terms of centerline effectiveness and lateral spreading of coolant. Figure 5.5a compares M = 1.5 laterally-averaged effectiveness of the rough diffuser holes to the other roughness configurations. Laterally-averaged effectiveness of rough diffuser holes was slightly closer to that of slightly rough holes than to that of rough holes.

At M = 3 the rough diffuser hole had lower performance than at M = 1.5, as shown by contours of adiabatic effectiveness in Figure 5.4h. Adiabatic effectiveness of the rough diffuser hole was close to that of the rough hole (Figure 5.4g). In terms of laterally-averaged effectiveness, Figure 5.5b shows identical performance between the rough diffuser hole and rough hole configurations at M = 3. The identical performance at M = 3 indicates that the shaped hole diffuser plays a major role in determining coolant mixing at high blowing ratios but not necessarily at low blowing ratios.

Area-averaged adiabatic effectiveness is compared among all the roughness configurations in Figure 5.13. At the blowing ratio of M = 1.5, in-hole roughness caused $\overline{\eta}$ decreases as great as 45%, seen for the rough hole configuration. For the rough diffuser hole configuration the $\overline{\eta}$ reduction was half this magnitude at 22%. The situation was different at M = 3 where all roughness cases showed a greater percent decrease from smooth-hole $\overline{\eta}$, as compared to the analogous $\overline{\eta}$ decreases at M = 1.5. Slightly rough holes had $\overline{\eta}$ 21% less than smooth holes, while the two roughest specimens both showed $\overline{\eta}$ decreases of 61%. 
The area-averaged results show that in-hole roughness becomes increasingly influential with higher blowing ratios and can decrease film cooling effectiveness by a factor of two or more. At $M = 3$ the reductions in effectiveness suggest that roughness in the diffused outlet has strong influence on boundary layers and core flow through the shaped hole.

Roughness Effects at High Freestream Turbulence Intensity

As discussed above, in-hole roughness was found to cause decreased adiabatic effectiveness principally due to increased dilution of the coolant jet and higher penetration into the mainstream. Often more-representative of actual engine conditions is high freestream turbulence, which causes increased dispersion of coolant. Adiabatic effectiveness was measured for smooth holes and for rough diffuser holes at $T_u = 13.2\%$ to determine whether in-hole roughness effects were important at high freestream turbulence conditions.

Contours of adiabatic effectiveness with high freestream turbulence intensity are shown in Figures 5.14a-b for the two roughness configurations at $M = 3$. By comparing...
Figure 5.14. Contours of adiabatic effectiveness at M = 3.0 with high freestream turbulence intensity for the (a) smooth holes and (b) rough diffuser holes.

back to the low freestream turbulence cases in Figures 5.4e and 5.4h, it is evident that freestream turbulence caused lower centerline effectiveness and increased lateral spreading of coolant toward the midpitches. Similar to the low freestream turbulence cases, effectiveness for the rough diffuser hole was still lower than that of the smooth hole at Tu∞ = 13.2%. Even at the midpitch, effectiveness values for the rough diffuser hole were often one contour level lower (Δη = 0.05) than those with the smooth hole.

The detrimental effect of in-hole roughness is quantified for low and high freestream turbulence conditions in the bar graph of Figure 5.15. Data for both M = 1.5 and M = 3 are shown. At high freestream turbulence, a similar reduction to that of the low freestream turbulence case was observed at M = 1.5: rough diffuser holes showed a 29% decrease in $\bar{\eta}$ relative to smooth holes at the same condition. For M = 3 and Tu∞ = 13.2%, the reduction in effectiveness due to rough outlets was large at 46%, yet was less severe than the reduction of 61% observed for M = 3 at low freestream turbulence intensity.

For smooth holes, Schroeder and Thole [73] previously noted that laterally-averaged effectiveness is less sensitive to freestream turbulence as the blowing ratio increases. At higher blowing ratios the coolant jet core with lowest temperatures is above the wall and, as such, freestream turbulence can help spread this coolant towards the wall. This trend was similarly observed with the rough diffuser holes, as seen in Figure 5.15. For the rough diffuser hole the high freestream turbulence was actually beneficial at M = 3, causing a 50% increase in $\bar{\eta}$ relative to the Tu∞ = 0.5% performance (compare the two striped bars for M = 3.0). Beneficial spreading at M = 3 by the Tu∞ = 13.2% turbulence was also reason for
the less-severe \( \bar{\eta} \) reduction due to rough outlets, as compared to the reduction occurring with low freestream turbulence.

![Bar chart showing area-averaged effectiveness at low and high freestream turbulence intensities for smooth holes and rough diffuser holes. Percent change from smooth hole effectiveness is listed.]

**Figure 5.15.** Area-averaged effectiveness at low and high freestream turbulence intensities for smooth holes and rough diffuser holes. Percent change from smooth hole effectiveness is listed.

### 5.5 Conclusions

Film cooling performance was characterized for shaped holes with different levels of roughness along the interior walls of the shaped holes. Measurements of the roughness showed that ranges tested were representative of those thought to exist in shaped holes in engine hardware.

Adiabatic effectiveness measurements showed that in-hole roughness can severely decrease effectiveness, with more detrimental effects occurring with larger roughness sizes. For the largest roughness size tested, area-averaged effectiveness decreased over 60% due to in-hole roughness at the high blowing ratio tested. This percent decrease in effectiveness was larger than the percent decrease observed just for smooth holes with blowing ratio
increases, which highlights that in-hole roughness is just as important a consideration as blowing ratio.

The influence of in-hole roughness itself was a function of the blowing ratio, with more severe decreases in effectiveness occurring at larger blowing ratios. Also, roughness of the diffused outlet alone was found to be increasingly influential as blowing ratio was increased. At high blowing ratio, the rough diffuser hole exhibited the same decrease in effectiveness as the hole with roughness everywhere, while at low blowing ratio a more moderate decrease in effectiveness occurred with the rough diffuser hole.

Thermal field and flowfield measurements showed that in-hole roughness caused thicker boundary layers along hole interior walls, which resulted in a faster jet core flow with correspondingly higher penetration angles as compared to a smooth hole at the same blowing ratio. Faster core velocities also led to higher turbulence intensity and therefore increased mixing of the coolant jet, reducing adiabatic effectiveness.

High freestream turbulence of 13.2% altered the distribution of coolant downstream of the shaped holes, but did not eliminate the effect of in-hole roughness. At high blowing ratio, roughness isolated to the diffused outlet caused large reductions in effectiveness relative to smooth interior holes.
Chapter 6.
Conclusions

This dissertation investigated how the film cooling performance of a shaped hole was influenced by roughness along interior walls of the hole. To accomplish this investigation, first the performance of the smooth-walled version of a shaped hole was studied and documented. Then several configurations of in-hole roughness were tested for the same shaped hole geometry and were compared back to the smooth-walled baseline case. This chapter is similarly organized with distinction between discussion of the smooth holes and discussion of holes featuring in-hole roughness. First, flow phenomena and performance are summarized for the smooth holes. Then, building on the observed baseline performance, discussion and conclusions are given regarding the effect of in-hole roughness. Finally, reflection is given on the significance of in-hole roughness and recommendations are made for future research.

6.1 Summary of Smooth Hole Performance

The shaped hole selected for this study had a straightforward design: cylindrical metering section and expansion angles of $7^\circ$ in each of the three directions from the hole centerline, resulting in a moderate exit-to-inlet area ratio of 2.5. Expansion of the diffused outlet was intentionally chosen to be representative of literature while not being overly-aggressive, since it was desired that the shaped hole would not exhibit complex flow phenomena such as separation in the diffused outlet of the hole. As a consequence of this choice, the shaped hole performed differently than some aggressively-expanded shaped holes in literature.

Film cooling performance in this dissertation was quantified through measure of adiabatic effectiveness, both in a local sense and with lateral- and area-averages. It was discovered that the shaped hole in this study exhibited effectiveness that increased with coolant flowrate up to a blowing ratio of 1.5. Beyond this blowing ratio, coolant patterns on the surface narrowed with increasing blowing ratio, resulting in laterally- and area-averaged effectiveness values that slightly decreased with the increases in blowing ratio.
This adiabatic effectiveness behavior with blowing ratio was observed at all coolant-to-mainstream density ratios tested and at all freestream turbulence conditions tested.

Flowfield and thermal field measurements were employed to determine the physical mechanism causing narrowed effectiveness patterns with increased blowing ratio. As coolant flowrate was increased to high blowing ratios, flowfields revealed high penetration of the coolant jet as indicated from location of peak streamwise velocity in the jet centerline plane. Blowing ratio increases were also accompanied by a stronger counter-rotating vortex pair (CRVP) measured in a lateral crossplane downstream of the shaped hole. Measurements of the resulting thermal fields showed that the CRVP brought coolant increasingly upward, away from the wall, with increases in blowing ratio. The CRVP also gradually brought mainstream fluid underneath sides of the coolant jet, causing the narrowed effectiveness patterns on the surface. At positions far downstream of the shaped hole at high blowing ratio, the CRVP brought mainstream fluid completely underneath the jet centerline, such that highest coolant concentration in the jet occurred above, not on, the wall.

Flowfield data revealed that jets from the shaped holes also had a concentrated core of high velocity, even though the diffused outlet of the hole served to expand and slow the coolant. At high blowing ratio this core flow exhibited velocity greater than that of the mainstream, even though the effective (at the hole exit) velocity ratio was less than unity for this coolant flowrate.

Elevated freestream turbulence intensity caused increased dilution and lateral spreading of the coolant jet from the shaped holes, as shown by measurements of adiabatic effectiveness and thermal fields. Velocity measurements showed that flowfield structure of the jet changed little due to high freestream turbulence. Inside the jet the mean velocities, the anisotropy of rms velocity fluctuations, and the streamwise-vertical component of turbulent shear stress ($\overline{u'v'}$) changed little due to elevation of freestream turbulence intensity. Increased dilution and increased spreading with elevated freestream turbulence were due to increased turbulent fluctuations surrounding the coolant jet.

Thermal field measurements showed that high freestream turbulence contributed to increased lateral spreading but not increased vertical spreading of coolant, despite comparable magnitudes for lateral and vertical rms velocity fluctuations along the perimeter.
of the coolant jet. The preferential lateral spreading was due to the wall redirecting eddies of the turbulent freestream when these eddies penetrated low in the coolant jet, where coolant concentration was greater than at the top of the jet. Eddies impacting the wall were directed laterally and outward, as evidenced by changes in turbulent shear stress between the low and high freestream turbulence cases. Turbulent shear stresses involving the lateral fluctuating velocity component, w', showed greater lateral extent at high freestream turbulence intensity than at low freestream turbulence intensity, especially in the region nearest the wall. These observations regarding dispersion behavior with high freestream turbulence can help designers build improved correlations and turbulence models for film cooling from shaped holes.

6.2 Summary of In-Hole Roughness Effects

Increasing the roughness level on interior walls of the shaped hole resulted in decreased film cooling performance relative to the smooth-walled version of the hole. Adiabatic effectiveness decreases greater than 60% were observed for rough holes relative to adiabatic effectiveness at the same condition with smooth holes. Even for small in-hole roughness of the slightly rough holes, effectiveness decreased greater than 20% relative to the smooth holes. These percent changes in adiabatic effectiveness showed that in-hole roughness can be one of the greatest determinants of shaped hole performance, having equal influence to even that of blowing ratio through the holes.

Percent decreases in adiabatic effectiveness were greater with increasing blowing ratio through the hole. For instance the slightly rough holes had the same effectiveness as smooth holes at low blowing ratios. However, with increased blowing ratios the slightly rough holes gradually and monotonically developed a percent decrease in effectiveness relative to that of smooth holes. Roughness in the shaped hole diffused outlet was also found to be increasingly important with increasing blowing ratio. At high blowing ratio the rough diffuser hole and rough hole performed equally-poorly, having the same low value for area-averaged effectiveness.

The heightened influence of roughness with increased roughness size and with increased blowing ratio were both consistent with the fact that roughness elements were influential when they protruded beyond the viscous sublayer of turbulent boundary layers
developing in the hole. Larger roughness elements were more likely to have heights greater than the viscous sublayer thickness, thus satisfying the criterion for roughness to have effect on the coolant flow. Likewise, increased coolant flowrates (blowing ratios) generally promoted thinner in-hole boundary layers and thereby increased the ratio of roughness size to boundary layer thickness. For in-hole roughness with a spectrum of roughness heights, as was the case with the irregularly-rough surfaces of this dissertation, more of the roughness elements protruded beyond the viscous sublayer at high blowing ratio than at low blowing ratio.

Thermal fields and flowfields measured with the rough hole indicated that in-hole roughness indeed had effect on the in-hole boundary layers and caused them to become thicker, resulting in faster core flow through the hole than occurred with smooth holes at the same blowing ratio. Consequences of the faster core flow were increased turbulence intensity inside the jet which hastened mixing between the coolant and mainstream, and increased jet penetration height that corresponded with coolant having less contact with the wall and therefore narrower footprints of adiabatic effectiveness.

At high freestream turbulence intensity, in-hole roughness still caused a large decrease in adiabatic effectiveness relative to smooth hole performance. As seen previously at low freestream turbulence intensity, the detrimental effect of in-hole roughness was more severe at high blowing ratio. Also, at high blowing ratio the high freestream turbulence was observed to actually be beneficial as compared to low freestream turbulence intensity. For the rough diffuser hole at high blowing ratio, area-averaged effectiveness increased 50% between the low and the high freestream turbulence cases. This increase in effectiveness was attributed to how the jet from the rough diffuser hole had tendency to penetrate high into the mainstream and have little contact with the wall. The elevated freestream turbulence spread the coolant laterally, increasing the coolant footprint from the rough diffuser hole.

6.3 Significance and Recommendations for Future Work

This dissertation showed that film cooling performance of a shaped hole can be detrimentally affected, sometimes quite severely, by small roughness levels along interior walls of the hole. The sensitivity of shaped hole film cooling to in-hole roughness has not
been previously reported in literature and perhaps is not considered in design codes used by gas turbine manufacturers. As a first step to implement results of this dissertation to improve design codes, designers can consider their desired coolant blowing ratios and consider how roughness inside film cooling holes of their manufactured components compares to the generally-engine-relevant roughness types selected for this dissertation: irregularly-rough with \( R_a/D \leq 0.02 \). If roughness heights and blowing ratios with the manufactured components are at the high end of those studied in this dissertation, designers could implement factors to account for the percent reductions in effectiveness seen in this dissertation for the shaped hole.

However, there is a multiplicity of possible surface roughnesses and some manufactured surfaces are not well represented by irregular roughness elements that were used in this study. Moreover, with the nascence of additively-manufactured gas turbine components, the diversity of surface roughness in cooling system components is likely only to increase. To improve fidelity of “in-hole roughness factors” implemented in design codes, film cooling performance could be measured for the specific hole geometry and roughness of interest. Computed tomography (CT) scanning could be used to measure local in-hole roughness in actual turbine components and then stereo lithography could be used to print scaled-up versions of the cooling holes for wind tunnel tests.

In-hole roughness also has implications for film cooling researchers. At the present time, most wind tunnel studies of film cooling performance do not document roughness inside the hole. To ensure correct interpretation and lab-to-lab repeatability of film cooling experiments, researchers should either 1) show that their holes are aerodynamically smooth or 2) document the in-hole roughness size and pattern. Correctly specifying the in-hole roughness levels not only will contribute to experiment repeatability, but will also aid in the proper selection of wall boundary conditions for CFD simulations.

This dissertation investigated performance of several roughness configurations inside a shaped hole and also examined the physical mechanism underlying the in-hole roughness effect. This dissertation serves as a foundation for future studies that can answer more-specialized questions about in-hole roughness. For instance, subsequent adiabatic effectiveness investigations could be performed to answer questions such as:
Is adiabatic effectiveness with other film cooling hole types, such as axial-cylindrical or shaped-compound-angle holes, detrimentally affected by in-hole roughness as severely as seen with the shaped hole of this dissertation?

Film cooling implementations in engine hardware often have crossflow at the film cooling hole entrance. How does magnitude of the in-hole roughness effect change with introduction of crossflow at the hole entrance, as opposed to the plenum inlet condition used in this dissertation?

To borrow from the traditional “displacement thickness” iterative method of airfoil design, could displacement thickness along interior walls of a rough shaped hole be estimated so as to guide the design of a slightly-larger rough shaped hole that would deliver the same core flow, and therefore effectiveness performance, as the original smooth-walled shaped hole?

Additional insight could also be gained at the more fundamental level of flowfields and thermal fields. Highly fruitful, but also difficult, would be flowfield measurements deep inside the shaped hole. With such data, flowfields inside smooth and rough holes could be compared to identify specific changes in separation regions and secondary flows caused by in-hole roughness. This data could possibly lead to insights on how to design holes that mitigate the detrimental effect of in-hole roughness. In-hole flowfield data would also provide insight that could lead to improvements in turbulence models used for CFD prediction of film cooling. Outside the shaped hole, more detail could be learned about the mechanism causing preferential lateral dispersion of coolant at high freestream turbulence intensity. Coincident instantaneous flowfield and thermal field measurements would be especially illuminating. Several measurement techniques, such as hot wire, cold wire, laser Doppler velocimetry (LDV), and particle image velocimetry (PIV) could contribute to such measurements. Perhaps most powerful is the technique of thermographic PIV, which could provide instantaneous flowfields and thermal fields over an entire 2D field of view. The instantaneous flow structures shown in detail by such measurements could be used to validate URANS, LES, and other time-resolved film cooling simulations—ultimately contributing to improvement in computational tools available to designers of gas turbine cooling systems.
Appendix.

Shaped Hole Literature Review Database

The spreadsheet below documents over 120 different shaped hole geometries identified in open literature. Geometries included conical holes, laidback holes, fanshaped holes, and many novel designs. The hole shape most frequent in literature was the laidback fanshaped, with over 50 variations identified for the laidback fanshaped type holes.

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<td>Bell, C., Hamakawa, H., and Ligrani, P., 2000, &quot;Film cooling from shaped holes,&quot; J. Heat Transfer, 122(2), pp. 224-232.</td>
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Conical Holes

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References


**Vita**

**Robert P. Schroeder**

Robert Schroeder was born in South Bend, IN in 1985, the eldest child of John and Cynthia Schroeder. Growing up in Indiana, he became an Eagle Scout in 2003 and graduated from high school in 2004. He attended Valparaiso University that fall and graduated in 2009 with a Bachelor of Science in Mechanical Engineering. During his time as an undergraduate, Robert completed internships at MORryde International in Elkhart, IN and at Caterpillar Inc. in Peoria, IL. He also completed four work rotations in the cooperative education program at NASA Johnson Space Center in Houston, TX.

Choosing to attend graduate school, Robert followed the suggestion of his last NASA co-op mentor, Ryan Stephan, and considered joining the Experimental and Computational Convection Laboratory (ExCCL) at Penn State to study gas turbine heat transfer under the guidance of Dr. Karen Thole. After completing a Valparaiso University solar energy research experience in the summer of 2009 with Dr. Robert Palumbo, Robert enrolled in graduate study at Penn State and indeed joined the ExCCL. Starting off, Robert was the beginning student in the installation and benchmarking of a new wind tunnel facility for film cooling research. During time as a graduate student Robert was a Tau Beta Pi Fellow from 2009-10, a NASA Aeronautics Scholarship Program Fellow from 2011-14, and a recipient of the Allan J. Brockett Student Award from Pratt & Whitney and Penn State in 2014. Receiving the award was a privilege and an honor, since Al Brockett was a familiar face from many previous Penn State visits. Robert completed his Master of Science in Mechanical Engineering in 2014 and his Doctor of Philosophy in Mechanical Engineering in 2015, both from Penn State.

Following graduate school, Robert has accepted a position at Sargent & Lundy in Chicago where he will contribute to the design of electricity power plants.