The Effect of a Meter-Diffuser Offset on Shaped Film Cooling Hole Adiabatic Effectiveness

Shaped film cooling holes are used extensively in gas turbines to reduce component temperatures. These holes generally consist of a metering section through the material and a diffuser to spread coolant over the surface. These two hole features are created separately using electrical discharge machining (EDM), and occasionally, an offset can occur between the meter and diffuser due to misalignment. The current study examines the potential impact of this manufacturing defect to the film cooling effectiveness for a well-characterized shaped hole known as the 7-7-7 hole. Five meter-diffuser offset directions and two offset sizes were examined, both computationally and experimentally. Adiabatic effectiveness measurements were obtained at a density ratio of 1.2 and blowing ratios ranging from 0.5 to 3. The detriment in cooling relative to the baseline 7-7-7 hole was worst when the diffuser was shifted upstream (aft meter-diffuser offset), and least when the diffuser was shifted downstream (fore meter-diffuser offset). At some blowing ratios and offset sizes, the fore meter-diffuser offset resulted in slightly higher adiabatic effectiveness than the baseline hole, due to a reduction in the high-momentum region of the coolant jet caused by a separation region created inside the hole by the fore meter-diffuser offset. Steady Reynolds-averaging Navier–Stokes (RANS) predictions did not accurately capture the levels of adiabatic effectiveness or the trend in the offsets, but it did predict the fore offset’s improved performance. [DOI: 10.1115/1.4036199]

Introduction

Shaped film cooling holes are added to hot gas path components in gas turbines to help reduce the metal temperature of the component, increasing life, and improving durability. These holes consist of two components: a metering section and a diffuser. The metering section is generally cylindrical and is drilled through the surface of the part. The diffuser acts to reduce the momentum of the coolant jet and spread the coolant across the surface. Many diffuser shapes and sizes have been tested as designers and researchers work to achieve the best possible film cooling performance. The absence of a diffuser results in what is referred to as a cylindrical hole, which is simple to manufacture and less expensive than more complex geometries, but has been shown to perform more poorly than shaped holes [1]. High performing cooling holes are advantageous because they can use less coolant, which is diverted from the compressor and could otherwise be used to produce power.

Both the meter and the diffuser are often manufactured using EDM. A carefully controlled and localized electrical spark is created between an electrode and the area of interest on a workpiece. This spark reaches temperatures between 8000 and 12,000 °C, and the intense heat melts away the material [2]. Occasionally during manufacturing, the meter and diffuser are created in two separate steps. First, the metering section is drilled through the wall, either with laser drilling, EDM, or another inexpensive and simple manufacturing procedure. Then, the part is moved to a different machine, where the diffuser is created using EDM with an appropriately shaped electrode for the given diffuser design. Finite alignment errors can occur in this two-step process that result in an offset between the meter and the diffuser centerlines.

This study attempts to characterize the detriment to film cooling effectiveness caused by a meter-diffuser offset, in five directions and at two offset sizes. Designers and manufacturers could benefit from knowing the impact of this detriment when deciding on a manufacturing process.

Review of Relevant Literature

Extensive research has been performed on film cooling since its invention, including both cylindrical and shaped holes. The overwhelming consensus, highlighted in Bunker’s shaped film cooling review [1], is that shaped holes outperform cylindrical holes, especially as blowing ratio increases. This is because a shaped hole with its expanded diffuser more closely approaches a slot injection, which leads to a continuous layer of film cooling over the endwall and is the ideal scenario [1]. Goldstein and Eckert [3] were the first to quantify the advantage of shaped holes, and since then, researchers and designers have improved film cooling effectiveness by varying parameters of the hole, particularly in the diffuser. The primary intention of the diffuser is to reduce the momentum of the coolant jet, relative to a cylindrical hole, and thus, decrease jet penetration into the mainstream [3]. Shaped film cooling holes are able to achieve a higher momentum flux without the coolant jet separating from the endwall.

In any jet in crossflow, there will develop a counter-rotating vortex pair (CRVP), described in detail by Cortelezzi and Karagozian [4]. This CRVP dominates the flow field of a film cooling hole, and can draw the hot mainstream fluid to the endwall and even under the coolant jet [1]. Thole et al. [5] showed that the CRVP is weakened by shaped holes, and that there is less shear mixing between the coolant jet and mainstream, less mainstream penetration, and more lateral spreading of the coolant jet, as compared to a cylindrical hole.

Amongst shaped holes, the most common in literature are laid-back fan-shaped holes, a collection of which was curated by Schroeder and Thole [6]. Saumweber and Schulz [7] found that the expansion angle of the diffuser, the inclination angle of the hole, and the length of the metering section are some of the most important geometric parameters in the laid-back fan-shaped design. Many variations of those parameters have been tested in order to...
find the optimal film cooling hole shape. Schroeder and Thole [6] created a film cooling hole that is representative of the observed parameter variations, in order to serve as a baseline for shaped film cooling research. It is against this baseline hole that the current study is compared.

The baseline 7-7-7 hole performs best at $M = 1.0$, and the performance decreases as the blowing ratio increases, due to the narrowing of the jets and increased mainstream penetration. At lower blowing ratios, the coolant jet sees increased lateral spreading, but the cooling effectiveness does not extend far downstream [6].

In addition to the many experimental studies performed on film cooling holes, many computational studies have been attempted as well. Using computational fluid dynamics (CFD) is helpful in understanding the flowfield in regions where measurement is difficult, but it is also important to understand that CFD has many limitations, as pointed out by Denten [8]. In order to solve the turbulence closure problem in a RANS, turbulence models make several approximations. One of the common approximations is that the eddy viscosity (relationship between fluid strain and turbulent stresses) is isotropic, which is not true in film cooling flows.

Of the available isotropic eddy viscosity RANS models, Silleti et al. [9] found that the realizable $k-\varepsilon$ model agreed best with experimental data for a shaped film cooling hole, especially just downstream of the film cooling hole where the largest temperature and velocity gradients exist. Additionally, Harrison and Bogard [10] found that for cylindrical holes, the $k-\omega$ model most closely predicted the laterally averaged effectiveness and the realizable $k-\varepsilon$ model best predicted the centerline effectiveness. Recently, work by Li et al. [11] has shown that an algebraic anisotropic turbulence model is better able to predict the flow field and adiabatic effectiveness for cylindrical film cooling holes, as opposed to an isotropic turbulence model. The algebraic anisotropic model better predicts the CRVP and the lateral spreading of the jets.

To the authors’ best knowledge, there has not been any published research on the effect of a meter-diffuser offset. Bunker [12] examined the effect of manufacturing tolerances on film cooling performance, but his study did not include the meter-diffuser offset effect. Despite very little research being done on this topic, several patents from major gas turbine manufacturers cite the issue of a meter-diffuser misalignment [13–16]. In these patents, General Electric, United Technologies Corporation, and Chromalloy Gas Turbine Corporation all cite a meter-diffuser misalignment as a manufacturing defect that is to be corrected by the cited invention. In general, the inventions in these patents attempt to create both the meter and the diffuser in a single step, thus eliminating the potential for misalignment. However, these gas turbine companies do not make their manufacturing processes public, and the inventions in these patents have not been cited in open literature, so it is difficult to know how prevalent the discussed manufacturing techniques are.

**Experimental Facility**

Adiabatic effectiveness measurements for this study were collected using an IR camera in a closed-loop wind tunnel, previously described and validated by Eberly and Thole [17]. A schematic of the wind tunnel is shown in Fig. 1. The mainstream flow is circulated by an in-line axial fan and is kept at a uniform temperature of 295 K by a bank of electrical heating elements and a chilled water heat exchanger. At the beginning of the test section, there is a suction loop to remove the incoming mainstream boundary layer. Once inside the test section, a new boundary layer develops on the styrofoam endwall and transitions to a turbulent boundary layer due to a trip wire located at $x/D = -35$. Boundary layer measurements for this test section were measured and reported by Schroeder and Thole [6] at low freestream turbulence of $Tu = 0.5\%$. The turbulent boundary layer was measured at $x/D = -4.7$, and for at least five pitchwise locations. The average values were reported as $0/\text{D} = 0.14$, $H = 1.45$, $Re_u = 670$, $Re_{*} = 315$, and $u_* = 0.5$ m/s [6].

Some air from the mainstream section of the flow is diverted to a coolant loop using a hermetically sealed blower. Due to the cryogenic temperatures experienced in the coolant loop, the air must be dried in order to avoid the formation of frost. This is accomplished by routing the air through a vent dryer filled with solid desiccant. The air is cooled in a heat exchanger with liquid nitrogen, and then, the gaseous nitrogen exiting the heat exchanger is added to the coolant loop. Before the plenum, but after the heat exchanger and addition of the gaseous nitrogen, the total coolant

![Fig. 1 Schematic of wind tunnel facility used in current study [17]](image-url)
flow rate is measured by a venturi flow meter. Inside the plenum, three flow conditioning screens are used to ensure flow uniformity at the entrance to the film cooling hole metering section.

Film cooling holes were machined in low conductivity Dow styrofoam \( (k = 0.029 \text{ W/m K}) \) in order to approximate an adiabatic endwall condition. Five holes were drilled in the endwall with a metering section diameter of \( D = 8.16 \text{ mm} \) and a lateral spacing of \( P/D = 6 \). The baseline hole used in this study is the 7-7-7 hole created by Schroeder and Thole [6] and shown in Fig. 2. The diffuser is defined as the region of the film cooling hole measured by \( L_{\text{fwd}} \) in Fig. 2. The meter-diffuser offset was created by translating the centerline of the diffuser relative to the meter by two offset sizes, \( 1/8D \) and \( 1/4D \), and five offset directions, as shown in Fig. 3. An isometric view of some of the offsets can be seen in Fig. 4.

A FLIR T650sc infrared camera was used to capture surface temperature measurements on the adiabatic foam endwall. An in situ calibration developed by Eberly and Thole [17] was done to ensure measurement accuracy of the camera, and the temperature measurements were then used to calculate adiabatic film effectiveness. Coolant and freestream temperatures were measured using multiple thermocouples in both the coolant plenum and on a thermocouple rake in the mainstream test section. Adiabatic effectiveness measurements made by Eberly and Thole for cylindrical holes in this facility showed good agreement with the literature [17].

Uncertainty Analysis

An uncertainty analysis was done for the variables of blowing ratio, density ratio, and adiabatic effectiveness by propagating measurement uncertainties, as shown in Figliola and Beasley [18]. All calculations were done using a 95% confidence interval.

Uncertainty in blowing ratio was found to be \( \pm 5\% \) for all blowing ratios, which is driven primarily by the bias uncertainty in the mainstream Pitot probe pressure transducer and also the coolant venturi flowmeter (\( \pm 0.25\% \) of full-scale flow and verified by separate tests with a laminar flow element in series). Uncertainty in density ratio was low, below \( \pm 0.01 \) for all blowing ratios. The uncertainty in adiabatic effectiveness was calculated to be \( \delta \eta = \pm 0.034 \). This is based on a surface temperature uncertainty of \( \pm 1.6 \text{ °C} \) and an average mainstream to coolant temperature difference of 49 °C. The uncertainty in surface temperature was due to both scatter in the in situ infrared camera calibration and also bias uncertainty of the thermocouples.

Repeatability was confirmed by repeating a measurement of the baseline 7-7-7 hole and comparing it to previous results found by Schroeder and Thole [6], who tested the same hole geometry but with a diameter that was 5% smaller. Several offset hatches were also retested, including a hatch that had been removed and reinstalled. The maximum difference in laterally averaged effectiveness was within the uncertainty range.

Computational Setup

A grid sensitivity analysis was performed to ensure the resultant CFD solution results were independent of the mesh size. Three different meshes were run on the fore 1/4D offset geometry with sizes of \( 3.00 \times 10^6 \), \( 4.02 \times 10^6 \), and \( 4.93 \times 10^6 \) cells. The results of this study can be seen in Fig. 5(a), for both centerline and...
laterally averaged effectiveness. The maximum difference in laterally averaged effectiveness at any \(x/D\) location was 0.0032 between the 3.00 and 4.02 million element cases, and was 0.0008 between the 4.02 and 4.93 million element cases. The maximum centerline effectiveness difference at any \(x/D\) location between these same cases was 0.0106 and 0.0112. These differences were deemed small enough for grid independence. Based on these results, a 5 million element size mesh, seen in Fig. 6, was selected to be used for the remaining CFD cases.

ANSYS FLUENT was used to run the CFD, using the realizable \(k-\varepsilon\) turbulence model. The CFD cases were run for 500 iterations with first order discretization, then switched to second order discretization for 2000 more iterations. Convergence was ensured through several monitors including an area-weighted average temperature monitor on the film surface downstream of the hole exit as well as four individual point monitors. The results can be seen in Fig. 5(b). Points 1 and 2 were located at an \(x/D = 1\) with point 1 in-line with the diffuser and point 2 near the lateral edge of the CFD domain. Points 3 and 4 were located at an \(x/D = 40\) with point 3 in-line with the diffuser and point 4 near the lateral edge of the CFD domain. The temperature monitors converged after about 1000 total iterations.

Discussion of Results

Contours of measured adiabatic effectiveness are shown at four blowing ratios for the baseline 7-7-7 hole in Fig. 7. At each blowing ratio, the central three holes in the array of five are displayed. Good periodicity is observed at all blowing ratios for these three holes, and also for the two outer holes, not pictured. Adiabatic effectiveness in the hole outlet is omitted, since the measured surface is not on the same plane as the IR camera’s focal length. Inspecting the contours visually, it is clear that from \(M = 0.5\) to \(M = 1.0\), the baseline hole sees improved effectiveness, not only near the hole but especially downstream. As the blowing ratio increases beyond 1.0, the footprint of the coolant on the surface narrows. From \(M = 1.0\) to \(M = 2.0\), the contour narrows, but the degradation in centerline adiabatic effectiveness is reduced, with the higher blowing ratio displaying effectiveness levels of \(\eta = 0.3\) at 35 diameters downstream. At the highest blowing ratio however, the contour not only narrows, but has a comparatively lower centerline effectiveness downstream. These trends were previously described by Schroeder and Thole [6] and our results match very well.

Figure 8 shows laterally averaged adiabatic effectiveness as a function of nondimensional distance downstream of the diffuser trailing edge for the four cases depicted in Fig. 7. As was seen in the contours, a blowing ratio of \(M = 1.0\) produces the highest cooling effectiveness near the hole. \(M = 2.0\) has slightly higher effectiveness than \(M = 1.0\) downstream of \(x/D = 20\), due to the slower rate of centerline effectiveness degradation and also due to the greater amount of coolant present over the endwall at the higher blowing ratio. Figure 8 also indicates good agreement to the study by Schroeder and Thole [6]. This no-offset case is used as a baseline throughout this study in order to determine the relative effect of a meter-diffuser offset in any given size or direction.

Effect of Offset Direction. Adiabatic effectiveness contours are shown in Fig. 9 for the baseline 7-7-7 hole and all five 1/4\(D\) offset directions, all at \(M = 1.0\) and \(DR = 1.2\). Visually inspecting the contours for both footprint width and centerline effectiveness degradation, it is clear that while the fore 1/4\(D\) offset is competitive with the baseline, all other offsets are a detriment, with performance decreasing as the offset moves from fore (best) to aft (worst). For cases that include a left diffuser offset, the coolant jets are noticeably positioned toward the right side of the diffuser since the meter is closer to that side (see Fig. 4).

This qualitative comparison is corroborated by the laterally averaged effectiveness curves for each 1/4\(D\) offset, shown in Fig. 10. At the blowing ratio of \(M = 1.0\), which is where the baseline exhibits peak performance, the fore 1/4\(D\) offset shows a slight improvement relative to the baseline. All other 1/4\(D\) offsets are a detriment to performance, relative to the baseline. The two
diagonal cases, fore-left 1/4D and aft-left 1/4D, have very similar effectiveness as the left 1/4D offset, to within experimental uncertainty. The fore-left 1/4D diagonal offset does seem to perform slightly better than the left, perhaps somewhat influenced by its advantageous fore offset component, but the improvement is only seen near the trailing edge of the hole and is within the experimental uncertainty between cases. Finally, the aft 1/4D offset is the worst performing offset direction relative to the baseline. A physical explanation for these trends is presented in the Predicted Flowfield Inside Holes section of the paper.

Effect of Offset Size. Two offset sizes were considered for this study: 1/4D and 1/8D. It was expected that the larger 1/4D offset would be worse than the 1/8D offset. This was indeed the case for four out of the five offset directions, except the fore offset.

Figure 11 shows contours of adiabatic effectiveness for three offset directions and both offset sizes, all at M = 1.0. For both the left and the aft offset cases, the footprint of the coolant on the endwall is noticeably narrower, and the degradation of centerline effectiveness is more pronounced with a larger, 1/4D offset. In the case of the fore offset, however, the 1/4D offset has a slightly wider footprint, and its centerline effectiveness is about the same as the 1/8D offset.

Also, it can be observed from the contours in Fig. 11 that the fore and aft offsets result in symmetric footprints. In the left offset, however, the coolant tends toward the side of the diffuser that is closer to the metering section exit. For the 1/4D left offset, this leads to sidewall separation in the diffuser. This effect is evident by the low effectiveness levels immediately at the trailing edge on the left side of the diffuser.

The effect of offset size can also be observed quantitatively by examining the laterally averaged effectiveness shown in Fig. 12 for the same cases depicted in Fig. 11. As expected from the contours, the left and aft offsets perform worse at a larger offset size. The fore offset, however, seems to be insensitive to offset size for these ranges. The effect of offset size is also demonstrated by the area-averaged effectiveness of these same cases, shown in Fig. 13 at all four tested blowing ratios. Area-averaged effectiveness is calculated from x/D = 3 to x/D = 15, across the pitch of the hole. The trends observed in Fig. 12 for each offset direction are
generally maintained at moderate blowing ratios, with the left and aft offset direction performing worse at an offset of 1/4D, and the fore offset direction performing the same or slightly better at an offset of 1/4D (although within experimental uncertainty) for all blowing ratios. However, at the lowest (M = 0.5) and highest (M = 3.0) blowing ratios, there is little impact of offset size. The effect of blowing ratio on all of the offset sizes and directions will be discussed later in this paper.

Effect of Blowing Ratio. Every offset size and offset direction was tested at four blowing ratios: M = 0.5, 1.0, 2.0, and 3.0. For every hole, the trend with blowing ratio was the same as the baseline described earlier; Fig. 14 shows an example of the left 1/4D offset. Effectiveness increases from M = 0.5 to M = 1.0, as the footprint of the coolant remains wide and high centerline effectiveness persists further downstream. From M = 1.0 to M = 3.0, the footprint of the coolant narrows with increasing blowing ratio and the effectiveness decreases. This decrease in effectiveness at high blowing ratio is especially pronounced in the left 1/4D offset. Since the coolant entering the diffuser in the left offset case is askew, it tends to separate from the side of the diffuser. This can be seen from the contours, since the high effectiveness levels right at the exit of the hole do not extend across the entire trailing edge. This only becomes more pronounced as the blowing ratio increases, which leads to very poor effectiveness.

In fact, the sidewall separation in the left 1/4D offset at M = 3.0 is such a detriment that at this blowing ratio, the left 1/4D offset performs as badly as the aft 1/4D offset. This is seen more clearly in Fig. 15, which shows the area averaged effectiveness for all cases (experimental uncertainty indicated).
in solid black. In Fig. 15, the trend of declining effectiveness with increasing blowing ratio generally levels out beyond $M = 2.0$ for the aft offset hole, but continues to decline sharply for the left offset hole.

At $M = 1.0$, all holes perform the best, and the trends previously described are apparent, which are that the fore 1/4D offset is advantageous, the aft 1/4D offset is the most detrimental, and the left and diagonal 1/4D offsets have very similar area averaged effectiveness. Also, while that general trend seems to hold for the 1/8D offsets as well, the difference for all of them relative to the baseline 7-7-7 is less than experimental uncertainty, and thus, they have a minor impact to cooling effectiveness.

**Predicted Versus Measured Trends.** CFD is a powerful tool to investigate the flow field where measurements cannot be taken, but typical RANS-based modeling makes several approximations which are not valid for film cooling flows. Despite having matched boundary conditions and a spatially resolved mesh, the CFD predictions did not match the experimental data in this study, either for predicting absolute levels of cooling or even cooling detriment trends for the various offset directions. Figure 16 shows contours of both experimental data and CFD for the baseline 7-7-7 hole and three offset directions, all at a 1/4D offset. These results are for a blowing ratio of 2.0. Several clear differences exist between the experimental and computational contours. The centerline effectiveness is over predicted by the CFD, especially near the hole, which is consistent with previous CFD studies by both Silieti et al. [9] and Harrison and Bogard [10]. For the baseline case and fore 1/4D offset, the CFD captures the jet spreading fairly well, but for the left 1/4D and especially the aft 1/4D offset cases, the footprint on the endwall is poorly predicted. Also, based on the left 1/4D offset contours, the CFD seems to have over predicted the sidewall separation inside the diffuser, which leads to the jet being narrower and more askew than the experimental data.

All of these differences can also be observed quantitatively. Figure 17 shows the laterally averaged effectiveness for three representative cases, for both predicted and experimental data. The baseline 7-7-7 prediction is somewhat reasonable near the hole, since the centerline effectiveness is overpredicted but lateral spreading is underpredicted. However, the lateral spreading and centerline effectiveness are both greatly over predicted for the aft 1/4D offset, so the laterally averaged effectiveness is very greatly over predicted. The prediction of the left 1/4D offset has a narrower and more askew coolant footprint relative to the experiment, but it captures the centerline effectiveness (defined at the meter centerline) fairly well, which leads to an underprediction of laterally averaged effectiveness.

Since some cases were under predicted, most notably the left 1/4D offset, and others were over predicted, most notably the 1/4D aft offset, the general trend of the effect of offset was not well captured by the CFD. This can also be seen by comparing the area-averaged effectiveness for both CFD and experimental data, for all holes at an offset of 1/4D, shown in Fig. 18. Some area-averaged predictions are reasonable or within experimental uncertainty (the baseline, fore and fore-left offsets), while others are not well captured (aft, left, and aft-left). However, the ranking of the best to worst performing hole is not well captured, with the CFD indicating that the left offset is worst while the experiments indicate the aft offset is worst.

**Predicted Flowfield Inside Holes.** Although the CFD did not accurately predict effectiveness magnitudes or trends of offset degradation for every case, it was nonetheless inspected to help understand the physics of the effect of offsets. Shown in Fig. 19 are plots of nondimensionalized velocity magnitude at the centerline plane of three film cooling holes: the 7-7-7 baseline, the fore 1/4D offset, and the aft 1/4D offset. To the best knowledge of the authors, experimental measurements have never been taken inside a shaped film cooling hole, so the flow field predicted by CFD has not been confirmed experimentally. Despite this, it is still reasonable to assume that CFD can capture the general flow features...
One such common flow feature is a large, low velocity, recirculating vortex located in the bottom of the diffuser. The existence of this recirculating vortex has already been reported in literature in other computational studies, for example by Kohli and Thole [19]. The size and shape of this vortex is dependent on the shape of the diffuser. Another important feature of the film cooling flow field is the area of high coolant jet momentum located at the top of the metering section where it enters the diffuser. This high velocity region can be seen in each depicted case, with some important differences. In each of the offset cases, a discontinuity of the wall occurs due to the offset. At this discontinuity, a separation region is created. For the fore 1/4D offset case, the separation region occurs right where the high velocity region of the coolant jet is located, which acts as an additional expansion and significantly reduces the velocity of the coolant jet, as compared to the 7-7-7 baseline case. The effect of this reduction in momentum can be seen more evidently when the coolant first interacts with the mainstream. The velocity at the hole outlet is lower in the fore 1/4D case than in the 7-7-7 baseline, which makes it less likely to penetrate into the mainstream flow and separate from the endwall downstream.

Conversely, the aft 1/4D offset has a higher velocity at the film cooling hole outlet, making it more likely to penetrate into the mainstream flow. This is because the high velocity region of the coolant jet is still present, and, because of the position of the diffuser relative to the meter, the jet is under-diffused relative to the baseline. This restricts the diffusion of the high momentum region of the coolant jet and leads to the highest amount of mainstream jet penetration of any of the studied holes, which may explain why the aft 1/4D offset is the worst performing hole experimentally.

Figure 20 also shows this effect, by depicting the nondimensionalized velocity magnitude at the exit plane of the film cooling hole for each of the three cases discussed in this section, all at M = 2.0. The exit plane of the film cooling hole is defined as a plane normal to the metering section and coincident with the leading edge of the diffuser (Aexit in Fig. 2). These contour plots make it clear that, compared to the baseline 7-7-7, the fore 1/4D offset case has a region of relatively high momentum, and the aft 1/4D offset case has a region of relatively low momentum. Therefore, the fore 1/4D offset has less jet penetration into the mainstream and thus higher effectiveness, and the aft 1/4D offset has more jet penetration and lower effectiveness.

Schroeder [20] recently reported contours of time-mean streamwise velocity and $\overline{u'v'}$ turbulent shear stress in the centerline plane for DR = 1.5, $M = 3.0$ for the baseline 7-7-7 hole, from Schroeder and Thole [20].
hole were not taken in Schroeder and Thole’s study, it is still clear that there is a high momentum region located near the leading edge of the film cooling hole outlet, which suggests that the predicted velocity inside the hole is likely reasonable.

Also in Fig. 21 is a contour plot of the turbulent shear stress. The turbulent shear stress is highest at the leading edge of the hole outlet, exactly where the region of high momentum is also located. This higher turbulent shear stress leads to greater turbulent mixing, both of the velocity and the temperature, which could lead to coolant jet degradation downstream. This could explain why offset cases with regions of higher momentum perform more poorly as compared to the baseline.

**Downstream Predicted Flowfield.** Figure 22 shows contours of nondimensional temperature at a plane coincident with the centerline of the meter-diffuser section. These contours are at M = 2.0, and they show the relative levels of coolant jet penetration. The 7-7-7 and 1/4D fore- and aft offset coolant jets remain close to the wall, which is consistent with the velocity plots discussed in the Predicted Flowfield section. From these contours, it appears as if the 1/4D aft jet is attached to the wall, but the temperature of the coolant at the hole degrades quickly as compared to the baseline case. For the 1/4D diagonal offset cases, and especially the 1/4D left offset case, the core of the coolant jet is lifted up away from the endwall, so the effectiveness on the wall is low, as was seen previously. It should be noted for the left offset cases, however, that since these coolant jets are askew with respect to the meter centerline, Fig. 22 does not necessarily capture the true center of the jet.

Another way to visualize the size and location of the coolant jet is depicted in Fig. 23. These are downstream planes, five diameters from the trailing edge, normal to the mainstream flow direction, which is into the page. The contour is of dimensionless temperature and the streamlines are based on in-plane velocity components. All five 1/4D offsets are depicted at M = 2.0.

From this figure, it is possible to see the shape of the coolant jets and explain the footprints they are leaving on the endwall. In each case, the CFD predicts that the cooling jets produce counter-rotating vortices (evidence of this was measured by Schroeder [20]). Note that the streamlines are just showing the direction of the in-plane velocity, not the magnitude, so this depicts the size and shape of the counter rotating vortices, but not the strength. The jet for a 1/4D fore offset remains attached to the wall and is providing high effectiveness, and has a small CRVP close to the wall. The 1/4D aft offset case is also attached to the wall, but compared to the fore offset case, it is experiencing more thermal diffusion in the direction away from the endwall and appears to have CRVP cores that are further away from the endwall. As previously mentioned, the diagonal 1/4D offset cases and the left 1/4D offset case have coolant jets whose cores are not near the endwall. These jets, especially for the left 1/4D offset, are showing signs of separation. This explains why the CFD predicts such a low effectiveness for the left 1/4D offset, because the coolant jet is separating and not providing coolant to the endwall. Also, the presence of a left offset makes the CRVP uneven, with the side without the offset having a larger vortex.

**Conclusions**

Experiments and computational predictions were performed to find the effect of a meter-diffuser offset in shaped film cooling holes. The baseline case was the 7-7-7 hole developed by Schroeder and Thole [6], and two offset sizes, 1/8D and 1/4D, were tested at five offset directions: fore, fore-left, left, aft-left, and aft. Adiabatic effectiveness measurements were taken in a large scale wind tunnel at DR = 1.2 and M = 0.5, 1.0, 2.0, and 3.0. The computational study was performed in ANSYS Fluent using the realizable k-ε turbulence model. The CFD predictions did not match the trend of the experimental data, but were within experimental uncertainty for all but two cases.

Both experimental data and computational predictions indicated that a fore offset has slightly higher film cooling effectiveness than the baseline 7-7-7. A predicted flowfield inside the film cooling hole indicates that a separation region on the upstream side of the hole caused by that offset configuration was responsible for diffusing a region of high momentum near the top of the cooling jet and reducing its interaction with the mainstream. Aside from the fore offset case, the separation regions inside the film cooling holes resulting from other offsets were detrimental to film cooling performance, especially as the size of the offset increased and as the blowing ratio increased. The aft offset was the worst performing case, because this offset configuration caused a restricted diffusion zone in the region of high cooling jet momentum which preserved high jet momentum inside the hole and led to poor cooling effectiveness.

This study shows that gas turbine manufacturers should try to minimize or eliminate a meter-diffuser offset, since it is generally detrimental in all directions but one. If the manufacturing process is such that an offset will occur, a fore offset is preferable. Also, the effect of the fore offset in this study suggests that film cooling holes could be designed such that the region of high coolant jet momentum in the film cooling hole is reduced, which would lead to less mainstream jet penetration and higher film cooling effectiveness.

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**Nomenclature**

\[ A = \text{hole cross-sectional area} \]
\[ AR = \text{area ratio, } A_{\text{exit}}/A_{\text{inlet}} \]
\[ c_f = \text{skin friction coefficient} \]
\[ D = \text{diameter of film cooling holes} \]
\[ DR = \text{density ratio, } \rho_i/\rho_\infty \]
\[ H = \text{shape factor, } \delta^*/\theta \]
\[ I = \text{momentum flux ratio, } \rho_i U_i^2/\rho_\infty U_\infty^2 \]
Greek Symbols

- \( \alpha \) = hole injection angle
- \( \beta \) = expansion angle for diffuser
- \( \delta \) = 99% boundary layer thickness
- \( \delta^* \) = displacement thickness
- \( \eta \) = local adiabatic effectiveness, \( (T_\infty - T_{aw})/(T_\infty - T_c) \)
- \( \theta \) = momentum thickness
- \( \nu \) = kinematic viscosity
- \( \rho \) = fluid density
- \( \varphi \) = nondimensional fluid temperature, \( (T_\infty - T)/(T_\infty - T_c) \)

Subscripts

- \( aw \) = adiabatic wall
- \( c \) = coolant, at hole inlet
- \( CL \) = centerline
- \( eff \) = effective, at hole exit
- \( exit \) = exit plane of the film cooling hole
- \( fwd \) = forward expansion of diffuser
- \( inlet \) = inlet plane of the film cooling hole
- \( lat \) = lateral expansion of the diffuser
- \( m \) = metering section
- \( \infty \) = mainstream

Superscripts

- \( \text{l} \) = laterally averaged
- \( \text{a} \) = area-averaged

References