EFFECTS OF JET IMPINGEMENT ON CONVECTIVE HEAT TRANSFER
AND DISCHARGE COEFFICIENTS IN EFFUSION HOLES

A Thesis in
Mechanical Engineering

by

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Abstract

As inlet temperatures for gas turbines increase to improve power it becomes increasingly necessary to better understand cooling techniques within combustor liners to improve durability. Current combustor liner designs utilize a double-wall with impingement jets and effusion cooling holes. Convective cooling through the effusion holes plays a large role in cooling the liner walls. Discharge coefficients through the effusion holes are measured control flow and resize the cooling holes if necessary. The majority of the current research on impingement and effusion holes focuses on cooling effectiveness, or discharge coefficients without specific impingement placements. The focus of this study was to measure the local internal convection and discharge coefficients within the effusion hole based on varying impingement geometries to aid in the understanding of combustor liner design. A large-scale, additively manufactured effusion hole with a constant heat flux boundary was built, and both local convective heat transfer coefficients along with discharge coefficients were measured under a multitude of impingement geometries. Results indicated that there was a strong influence of the impingement hole relative to the effusion hole on convective heat transfer within effusion hole. However, discharge coefficients were only sensitive to impingement placement in only a few key locations.
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Nomenclature

A  local surface area
A_{\text{total}}  total surface area
D  impingement and effusion hole diameter
H  impingement jet-to-effusion plate spacing
r  radial distance from impingement plate center
h  local heat transfer coefficient
\bar{h}  area-averaged heat transfer coefficient
k_{\text{air}}  thermal conductivity of air
I  distance along effusion hole
L  effusion hole length
\Delta P  Pressure drop across effusion hole
\rho  Effusion hole air density
\dot{m}  mass flow rate
Nu  Nusselt number, hD/k_{\text{air}}
\bar{\text{Nu}}  area-averaged Nusselt number, \bar{h}D/k_{\text{air}}
\bar{\text{Nu}}_{0}  \bar{\text{Nu}}, no impingement case
q''  heat flux
Cd  discharge coefficient through effusion hole
Re  Reynolds number, \dot{m} \mu^{-1} \cdot (\pi/4)^{1} \cdot D^{1}
T_{\text{air}}  local mean air temperature
T_{w}  local effusion surface temperature
t_{e}  effusion plate thickness
\( t_i \)  
impingement plate thickness

\section*{Greek}

\( \alpha \)  
effusion inclination angle

\( \theta \)  
Impingement circumferential location angle

\( \mu \)  
dynamic viscosity

\( \Phi \)  
effusion circumferential location angle
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Chapter 1

Introduction

Gas turbines are widely used to power aircraft or produce power in industrial applications. In modern gas turbines, the temperatures of the combustion gases are higher than the melting temperature of the turbine components. Due to these elevated temperatures, it is necessary to use cooling air to protect turbine components. The cooling air is bled off from the compressor and used to cool the turbine components by feeding air through passages built into the components and exhausting the coolant to create a protective layer of air over the part surfaces. As the air moves through the internal passages it picks up heat through convection, and as it exits the passage the cooling layer of air also protects the components against the hot gases. However, the bleed off from the compressor leads to drops in thermal efficiency since the cooling air does not create work. Therefore, it is imperative to effectively protect components while keeping the amount of necessary cooling air to a minimum. The studies presented in this research will focus on analyzing cooling with regards to the combustor liner of a gas turbine.

1.1 Combustor Liner Cooling

Combustor liner cooling is performed using a variety of impingement jets and effusion jets. Mainly, this is done in a double-wall combination where the impingement jets are feeding air through the effusion holes. The effusion holes are angled to aid in creating a cooling film over the surface. A schematic of this double-wall design can be seen in Figure 1-1.
The impingement holes use coolant bled off from the compressor, creating a jet that cools the backside of the effusion wall and feeds into the effusion holes. The coolant travels through the effusion holes, cooling the internal hole walls through convection. As the air exits the hole, it provides a layer of coolant over the part surface, effectively protecting it from the hot gas path. A large portion of the overall combustor liner cooling comes from internal convection through the effusion holes. Impingement position changes how the effusion holes are fed and as such can also impact the internal convection within the effusion holes. To estimate flow rate through the effusion holes, a discharge coefficient measurement is often taken. This discharge coefficient can be used to aid in resizing effusion holes to implement the needed flow. Once again, impingement position can impact the discharge coefficient through the effusion hole. Understanding how impingement position affects both the internal convection and discharge coefficients through the effusion hole can help optimize combustor liner cooling by maximizing cooling effectiveness while minimizing the amount of coolant needed.
1.2 Objectives and Document Outline

This thesis reports the effects that the jet impingement position has on internal convective heat transfer and discharge coefficients within effusion holes. Internal convective heat transfer measurements will be broken up into local measurements along the effusion hole and overall heat transfer coefficients. The overall objective is to better understand how impingement location can change the previously mentioned variables to aid in the design of combustor liners. Chapter 2 will focus on the internal heat transfer coefficients through the effusion hole and is composed of a paper that is recommended for publication in the *Journal of Turbomachinery* and has been accepted into Turbo Expo 2020. Chapter 3 analyzes the discharge coefficients measured from the same experiments seen in the paper mentioned above and has been written for publication in the future. Chapter 4 will summarize the main findings and provide recommendations for future work.
Chapter 2

Effects of Jet Impingement on Convective Heat Transfer in Effusion Holes

2.1 Introduction

Resulting from increased temperatures seen in modern combustors, effective cooling is necessary to ensure combustor liner durability. Current combustor liners are highly engineered with a double-wall design containing impingement and effusion cooling. Impingement jets cool the backside wall, while effusion jets create a protective film over the external wall. Internal convection from the effusion holes has also been shown to play a large role in the overall cooling of the combustor liner walls. What is not known is how the coolant feed from the impingement hole that supplies the effusion hole affects the internal convective cooling from the effusion hole, especially at the entrance region of the effusion hole. In many designs, the impingement hole location relative to the effusion hole entrance can vary, whether it be from manufacturing tolerances or design constraints. This variation can, in fact, affect the internal convection from the effusion holes leading to either better cooling or detrimental temperatures.

In the existing literature, numerous studies have reported adiabatic effectiveness and overall effectiveness for double-wall combustor liners [1-4]. However, few studies have reported the details of internal convection within effusion holes that are supplied by impingement jets. The uniqueness of this paper is that numerous experiments were conducted in which the local and averaged internal convection coefficients of an effusion hole were measured for a wide variety of impingement jet locations. Given that the entrance conditions for the effusion jets can vary widely and the entrance flow is highly influential on the convective heat transfer, the focus of this study was on the entrance region of the effusion hole.
2.2 Literature Review

Only a few studies have been published on the internal heat transfer within film cooling holes. Boelter et al. [5] studied the heat transfer in the entrance region of long circular tubes with varying entrance conditions such as angled bends or flared openings. His paper showed high heat transfer at the entrance to the pipe in which he developed correlations to predict average heat transfer coefficients for the long pipes that were significantly longer than most film-cooling holes. In a study modeling a combustor wall using the naphthalene sublimation method, Cho et al. [6] found local and average mass transfer coefficients through a circular, short hole (L/D < 1.5). They showed that a separation region was present at the hole entrance that decreased with increasing Reynolds numbers until Re > 5000. It was also found that around 60% of the total mass transfer of the combustor liner was due to the convective heat transfer from the cooling holes. Kohli and Thole [7] performed a CFD analysis on angled film cooling holes that showed a similar separation region at the inlet to the film-cooling holes that was very sensitive to the coolant supply direction.

An analytical model of film cooling holes in a combustor wall by Martiny et al. [8] presented a method for optimization of cooling hole distributions. Similar to Cho et al. [6], they found that 60% of the heat transfer was due to the internal heat transfer in the effusion holes. This assertion was further verified by Terrell et al. [9] through a combination of experimental and CFD results from film cooling holes in a leading edge of a model turbine blade. They found that the coolant hole internal heat transfer accounted for 50 – 80% of the convective heat transfer.

Recently, Bryant et al. [10] devised CFD models to isolate the cooling mechanisms present in the overall effectiveness of a single wall external film cooling design by varying boundary conditions. To evaluate the effect of internal hole cooling on the overall effectiveness, surfaces of the film cooling hole were set to be adiabatic to remove the effect of internal cooling
and results were then compared to the baseline case. In Bryant et al.’s results, internal hole cooling was most effective at cooling the upstream surface of the effusion hole and near the exit of the hole.

None of the previously discussed studies have evaluated internal convective cooling of effusion holes in relation to impingement positions, especially with respect to local measurements. To begin to close this knowledge gap, the current study adds data on the local heat transfer within the effusion hole as affected by varying impingement patterns.

### 2.3 Impingement and Effusion Geometries

A scaled-up, double-wall combustor liner with a single effusion hole and single impingement hole was constructed with the details given in Table 2-1 and shown in Figure 2-1. Note that the entrance effects were the primary interest for this study and, as such, the exit cross-flow was not simulated for the effusion hole since it does not have significant affects to the internal heat transfer as shown by Cho and Goldstein [11].

Both the effusion and impingement holes had the same diameter, \( D = 2.54 \text{ cm} \) as shown in Figure 2-1. The effusion hole was constructed as a single pipe that was 3D printed from a low thermal conductivity plastic. The printed effusion hole (pipe) allowed for specific thermocouple placement along the length of the pipe as well as around the circumference. Three different impingement plates containing a single hole were constructed, which allowed for testing at a multitude of positions, both radially as well as circumferentially, relative to the effusion hole. The impingement plate thickness, \( t_i \), and impingement and effusion hole diameters, \( D \), remained the same for each of the plates.
Table 2-1. Effusion and Impingement Variables

<table>
<thead>
<tr>
<th></th>
<th>Impingement</th>
<th>Effusion</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$ (deg)</td>
<td>90</td>
<td>30</td>
</tr>
<tr>
<td>$t_i$ / D</td>
<td>1</td>
<td>N/A</td>
</tr>
<tr>
<td>$t_e$ / D</td>
<td>N/A</td>
<td>1.5, 3</td>
</tr>
<tr>
<td>H/D</td>
<td>3, 6</td>
<td>N/A</td>
</tr>
<tr>
<td>L/D</td>
<td>N/A</td>
<td>3, 6</td>
</tr>
<tr>
<td>r/D</td>
<td>0, 1, 3</td>
<td>N/A</td>
</tr>
<tr>
<td>$\theta$ (°)</td>
<td>0, 90, 180, 270</td>
<td>N/A</td>
</tr>
<tr>
<td>$\Phi$ (°)</td>
<td>N/A</td>
<td>-180, -135, -90, -45, 0, 45, 90, 135, 180</td>
</tr>
</tbody>
</table>

Figure 2-1. Effusion hole and impingement hole geometries. Flow is exiting the page for the bottom models.
The impingement hole positions were varied between three radial positions, \( r/D = 0 \) (direct impingement), 1, and 3, as illustrated in Figures 2-1 and 2-2. The impingement and effusion double-walled geometry combinations were created by switching out the impingement plate and by changing the impingement plate circumferential position by 90° intervals. Two jet-to-target spacing values were chosen for testing, \( H/D = 3 \) and 6, which are displayed in Figure 2-1. In total, 19 different geometry combinations were tested.

![Figure 2-2. Impingement and effusion geometries studied.](image)

**2.4 Experimental Setup and Methods**

To perform the heat transfer experiments, compressed air flowed through a sealed test rig containing the scaled-up impingement and effusion plates as shown in Figure 2-3. The impingement plate was mounted inside of a larger pipe with two endcaps placed over either side of the pipe. The test rig itself had a diameter that was 8 effusion hole diameters to avoid any sidewall effects. A foam insulation block surrounded the effusion hole to reduce heat losses.
during testing. The endcap upstream of the impingement plate included a nozzle and splash plate to diffuse the air flow entering the rig. Air flow was controlled using a mass flow controller.

![Diagram showing test facility with impingement and effusion test plates]

**Figure 2-3. Test facility showing impingement and effusion test plates.**

Thermocouples were inserted into 38 slots located along the length and around the circumference of the effusion hole so that the thermocouple beads were flush with the inner surface of the effusion hole as shown in Figure 2-4. Each slot was then filled with a thermally conductive epoxy to ensure that the bead was in contact with the inner heat flux surface. Thermocouple placement started at the entrance to the pipe (effusion hole) and continued at every 10% interval along L/D up to \( \theta/D = 2.7 \).
Figure 2-4. Effusion hole instrumentation showing the heated surface with thermocouples placed radially around the effusion hole. Flow is exiting the page.

A constant heat flux boundary condition was applied to the pipe walls of the effusion hole. The heater was made by adhering 16 stainless steel strips, each 0.0254 mm thick and 2.54 mm wide, between two Kapton sheets. The Kapton-steel strip sandwich was then glued to the inner surface of the effusion hole with double-sided, high temperature adhesive. Thermocouples were attached to the backside of the heater sandwich to measure the local surface temperatures. For a given experiment, a constant power was input to the heater and a Reynolds number at the effusion hole was set based on the mass flow rate of air measured and controlled by the mass flow controller. The static pressure and flow temperature were measured at the inlet of the effusion hole. Reynolds numbers were varied between $2000 < \text{Re} < 11,000$.

The heater power and surface temperatures were measured once steady state was reached, which typically required ~3 hours. The heat loss was accounted for by using thermocouples placed inside the insulating foam around the effusion hole and performing a conduction analysis. Conductive heat losses stayed within 2%-4% for all tests. After the heat loss was subtracted from the input power, local heat transfer coefficients were calculated based on the local mean temperature of the fluid, which was calculated using a first law analysis. The local heat transfer
coefficient, \( h \), for each position was calculated using Equation 2-1, where the air temperature, \( T_{air} \), was the local mean temperature and \( T_w \) was the wall surface temperature.

\[
h = \frac{q}{T_w - T_{air}}
\]  

(2-1)

An area-averaged heat transfer coefficient, \( \overline{h} \), was found through an area-weighted formula shown in Equation 2-2.

\[
\overline{h} = \sum \frac{h \cdot A}{A_{total}}
\]  

(2-2)

**Experimental Uncertainty**

Uncertainty in the experiments was calculated using the methods from Kline and McClintock [12]. The uncertainty of the local Nusselt numbers was found to be between 3%-7% with uncertainty decreasing as the hole length increased, and increasing as the Reynold’s number increased. This increase in uncertainty is attributed to the smaller temperature difference between the cooling flow and the effusion wall temperature. Uncertainty for the averaged Nusselt numbers was found to be under 1% for all cases. Finally, uncertainty in the Reynolds number was found to be less than 1% for all cases.

**2.5 Effect of Angular Impingement Location**

As was stated in the literature review, Kohli and Thole’s [7] computational studies showed the influence that the coolant supply direction had on the film-cooling performance as well as the flow-field within the cooling hole. Figures 2-5 and 2-6 show the area-averaged Nusselt numbers for the short and long effusion holes (\( L/D = 3 \) and 6) over the range of Reynolds numbers for differing jet impingement locations for the closest radial position, \( r/D = 1 \), and closest impingement distance, \( H/D = 3 \). Also shown in Figures 2-5 and 2-6 is what would be
expected from a fully-developed pipe flow, as indicated by the Gnielinski correlation [13], as well as the measured convective heat transfer coefficients for direct jet impingement and no impingement cases.

For both effusion hole lengths, shown in Figures 2-5 and 2-6, there is a substantially higher Nusselt number for all the cases relative to the fully-developed pipe flow correlations, which indicates the benefit of the entry region on the convective cooling of the combustor liner. For the shorter effusion hole, L/D = 3, at a Re = 8000, there is a 148% higher Nusselt number for the no impingement case relative to the fully-developed correlation. At the same Re = 8,000 for the longer effusion hole, there is only a 104% higher Nusselt number for the no impingement case relative to the fully-developed correlation. As the effusion hole increases in length with more surface area, the benefits of the high heat transfer coefficients in the entry region are reduced.

Figure 2-5. Area-averaged Nusselt number for the short effusion hole, L/D = 3, with constant radial spacing, r/D = 1, and constant jet-to-target spacing, H/D = 3.
Figure 2-6. Area-averaged Nusselt number for the long effusion hole, $L/D = 6$, with constant radial spacing, $r/D = 1$, and constant jet-to-target spacing, $H/D = 3$.

Although it might be expected that direct impingement would result in the highest Nusselt numbers, the data in Figures 2-5 and 2-6 indicate otherwise. In fact, the direct impingement is between the side injection ($90^\circ$ and $270^\circ$) cases and the in-line injection ($0^\circ$ and $180^\circ$). These results indicate a major flowfield change when feeding the effusion hole using side injection.

In addition to the augmentation of the convective heat transfer that occurs resulting from the entry region of the effusion hole relative to a fully-developed pipe flow, there is also significant augmentation for the impingement cases relative to that of no impingement. Figures 2-7 and 2-8 directly show the heat transfer augmentation for the jet impingement relative to the no impingement case for both the long and short effusion holes for the circumferentially varying jet locations. The augmentation results indicate that there is no dependence upon Reynolds number with essentially a constant value for a given geometry. For each impingement case, the
augmentation is always higher for the shorter hole length at $L/D = 3$ relative to the longer hole length at $L/D = 6$.

In Figures 2-7 and 2-8, the augmentation results for both the long and short effusion holes indicate that the highest Nusselt number augmentations occur when the impingement jet is positioned at the sides of the film-cooling holes, $\theta = 90^\circ$ and $270^\circ$ even moreso than when there is direct impingement, which is also shown in Figure 2-5 and 2-6. It may be expected that, given these locations are symmetric, the same augmentation levels would result. For the short effusion hole, the same augmentation did occur for the $\theta = 90^\circ$ and $270^\circ$ cases, but for the longer effusion hole, there was a slightly higher augmentation for the $\theta = 90^\circ$ case relative to the $\theta = 270^\circ$ case. This difference is attributed to the significant sensitivity of the flowfield at the entrance to the cooling hole.

The lowest Nusselt number augmentation for the impingement cases occur when the impingement is positioned most closely to the effusion hole upstream entrance location at $\theta = 180^\circ$ as shown in Figures 2-7 and 2-8.

Figure 2-7. Nusselt number augmentation relative to the no impingement case for $L/D = 3$. 
Figure 2-8. Nusselt number augmentation relative to the no impingement case for L/D = 6.

To further evaluate the differences in augmentation from varying the impingement positions seen in Figures 2-7 and 2-8, local Nusselt numbers at the entrance of the effusion holes (10% along the length of the hole, \( t/D = 0.3 \) for the L/D = 3; and \( t/D = 0.6 \) for the L/D = 6) were plotted in Figures 2-9 and 2-10 for Re= 8,000. As would be expected in Figures 2-9 and 2-10, the local heat transfer coefficients are relatively symmetric for the impingement locations of \( \theta = 0^\circ \) and \( 180^\circ \) with only slight anomalies for the \( \theta = 0^\circ \) in Figure 2-9 for the short L/D hole, which is attributed to the sensitivity due to the separation region.

For both L/D geometries in Figures 2-9 and 2-10, the general spread of the local convective heat transfer coefficients when plotted were similar. At the same case for the \( \phi = 0^\circ \) position, a substantial drop in Nusselt number is observed regardless of L/D. This drop is indicative of the flow separation region as described in Kohli and Thole’s [7] CFD analyses, which is caused by the turning angle of the fluid into the hole. It is important to note, however, that not exactly the same values occurred, which is related to the fact that in both cases these measurements were taken at 10% of the total length of the hole, which is effectively a larger downstream distance for the longer cooling hole case of L/D = 6 relative to the shorter cooling hole case of L/D = 3. When comparing L/D = 3 to L/D = 6 at \( \phi = 0^\circ \), the local heat transfer
coefficients are nominally 30% higher for the L/D = 3 case further illustrating the importance of the entrance region effects on heat transfer from the effusion holes.

As expected, Figures 2-9 and 2-10 show that there is an increase in the local heat transfer coefficients for the θ = 90° and 270° cases on the side of the effusion hole where the impingement took place. These local increases translate to higher overall averages of the heat transfer augmentation for the θ = 90° and 270° cases as was shown in Figures 2-7 and 2-8. The reason for these increased heat transfer coefficients, more so than other impingement locations, is because of the differing entrance flow conditions. Kohli and Thole’s [7] study showed that, depending upon the feed location of a film-cooling hole, it is possible to alter the flow separation angle, which is present in the case of most film-cooling type flows, and even induce a swirl.

The hypothesis of an induced swirl for the θ = 90° and 270° impingement cases at the hole entrance is supported by contrasting the data given in Figure 2-11, which shows the streamwise distribution of the local heat transfer coefficients on both the upstream and downstream walls of the effusion holes for the θ = 0° and 90° tests for the short L/D = 3 hole. Referring back to Figure 2-2, it is important to note that for the θ = 0°, the impinging fluid is targeting the upstream wall of the hole entrance. For the impingement fluid to enter the effusion hole, a large flow turning angle is required, which induces a separation region on the upstream wall. It would be expected that for the θ = 0° case, the large separation along the upstream wall of the effusion hole would result in lower convective heat transfer coefficients which, in fact, is shown in Figure 2-11. In contrast, for the θ = 90° case, the upstream wall has significantly higher local heat transfer coefficients along the wall. In both the θ = 0° and 90° cases, the local heat transfer coefficients along the downstream wall are nominally the same along the length of the pipe.
Figure 2-9. Local Nusselt numbers around the effusion hole entrance at the \( t/D = 0.3 \) position for the \( L/D = 3 \) effusion hole.

Figure 2-10. Local Nusselt numbers around the effusion hole entrance at the \( t/D = 0.6 \) position for the \( L/D = 6 \) effusion hole.
Figure 2-11. Local Nusselt numbers along the upstream and downstream lengths of the effusion holes contrasting two different impingement locations for $L/D = 3$, $r/D = 1$, and $H/D = 3$.

2.6 Effect of Impingement Radial Spacing

To determine the sensitivity of the radial location of the impingement jets, the impingement hole positioned at $r/D = 3$ was contrasted to that of $r/D = 1$ leaving the jet-to-target spacing constant at $H/D = 3$ for both effusion hole lengths. In Figures 2-12 and 2-13, the area-averaged Nusselt numbers for the increased radial position are compared to the close radial position. Three angular $\theta$ locations, $\theta = 0^\circ$, $90^\circ$, and $180^\circ$, are shown in this section, as the $\theta = 90^\circ$ and $270^\circ$ impingement locations have similar results.

It is expected that for the varying radial spacing of the impingement jet in Figures 2-12 and 2-13, there is a substantial decrease in the heat transfer for the larger radial impingement placement ($r/D = 3$) for both the $\theta = 0^\circ$ and $90^\circ$ cases relative to the close spacing, regardless of Reynolds number. This decrease is seen for both $L/D = 3$ and $L/D = 6$. In contrast, for the $\theta = 180^\circ$ impingement location, higher heat transfer occurs for the larger radial spacing ($r/D = 3$)
relative to the close spacing (r/D = 1) for both effusion hole lengths. This effect is due to a separation region created by the farther radial spacing.

As shown in Figure 2-14, for the close radial spacing the upstream wall has an increased Nusselt number along /D < 1.5 when compared to the far radial spacing. However, at /D ≥ 1.5, the upstream wall heat transfer collapses onto the same line for both r/D impingement positions. In addition, the downstream wall Nusselt number of the close radial spacing test is lower throughout the entire length of the effusion hole, while the downstream wall of the far radial spacing Nusselt number is on par with its respective upstream wall results past /D=0. At the close radial spacing for the θ = 180° case, the upstream and downstream wall results are indicative of a direct impingement-like effect where the flow is attached to the upstream wall along the majority of the effusion hole length, leading to the large difference in Nusselt number for both walls. For the far radial impingement at θ = 180° the flow was more like a co-flowing crossflow as shown in Kohli and Thole [7]. This crossflow initially had less cooling effectiveness at the entrance, but since the flow was more like a fully-developed flow along the length of the hole rather than just the upstream wall as in the r/D = 1 position, it had better cumulative heat transfer. This effect of higher heat transfer at downstream /D locations is also supported by the circumferential local Nusselt numbers at /D = 1.5 as seen in Figure 2-15. In Figure 2-15 for the close radial spacing, the upstream wall has higher heat transfer that decreases by a significant amount along the downstream wall. On the other hand, the far radial spacing at /D = 1.5 has an almost constant convection effect around the circumference of the effusion hole that is on par with the upstream wall values seen in the r/D = 1 case.

The heat transfer augmentation of the r/D = 1 locations over the r/D = 3 locations is displayed in Figure 2-16. Since the area-averaged Nusselt number augmentation is generally the same between effusion hole lengths, as seen in the previous figures, Figure 2-16 was not
replicated for the long effusion hole. For both the $\theta = 90^\circ$ and $0^\circ$ cases, the close radial spacing of the impingement holes leads to roughly a 10% increase in area-averaged Nusselt number across all Reynolds numbers when compared to the far radial spacing. The upstream impingement location, $\theta = 0^\circ$, is not affected as much by the increase in radial spacing and there is, on average, a 3% decrease in area-averaged Nusselt number between the $r/D = 1$ impingement site and the $r/D = 3$ impingement site.

![Figure 2-12. Area-averaged Nusselt number with changing $r/D$ and $\theta$ for the L/D = 3 effusion hole.](image)

Figure 2-13. Area-averaged Nusselt number with changing r/D and θ for the L/D = 6 effusion hole.

Figure 2-14. Local Nusselt number along the upstream and downstream walls contrasting two r/D values at θ = 180°.
Figure 2-15. Local Nusselt numbers around the effusion hole at the midpoint, \( l/D = 1.5 \), position for the \( L/D = 3 \) effusion hole.

Figure 2-16. Average Nusselt number augmentation of the close radial spacing compared to the far radial spacing for the \( L/D = 3 \) and \( H/D = 3 \) geometry.
2.7 Effect of Jet-to-Target Spacing

The impingement jet-to-target spacing was varied to determine the effects on the sensitivity of the convective heat transfer. For these experiments, the closest radial spacing was used at \( r/D = 1 \) for both short and long hole lengths. The data for the two hole lengths are shown in Figures 2-17 and 2-18 to compare the area averaged Nusselt numbers for the close impingement location, \( H/D = 3 \), to the far impingement location, \( H/D = 6 \).

The results in Figure 2-17 and 2-18 show that there is little sensitivity in the area averaged Nusselt number when changing the jet-to-target distance except for the \( \theta = 90^\circ \) case, which shows a higher convective heat transfer coefficient for the closer target spacing of \( H/D = 3 \) relative to \( H/D = 6 \). Given that all the results presented in this paper up to this point consistently point to the turning of the impingement flow into the cooling hole having a large impact on the effusion hole convection, it is expected that the data shown in Figures 2-17 and 2-18 is consistent.

By placing the impingement hole at \( \theta = 90^\circ \), increases in convective heat transfer can be gained; however, as the impingement jet is moved further from the effusion hole, this benefit decreases resulting in lower convective heat transfer for the \( \theta = 90^\circ \) case with larger jet-to-target distance of \( H/D = 6 \). In Figure 2-19, it is shown that there is little to no change in heat transfer augmentation of the \( H/D = 3 \) location vs the \( H/D = 6 \) location for the \( \theta = 0^\circ \) and \( 180^\circ \) angular impingement positions, though the \( \theta = 90^\circ \) site had a 3.5% decrease on average when increasing the spacing.
Figure 2-17. Total area-averaged Nusselt number with varying jet-to-target spacing for $r/D = 1$ and $L/D = 3$.

Figure 2-18. Total area-averaged Nusselt number with varying jet-to-target spacing for $r/D = 1$ and $L/D = 6$. 
Figure 2-19. Average Nusselt number augmentation of the close impingement jet-to-effusion hole spacing compared to the large spacing for a radial spacing for the r/D = 1 and L/D = 3 geometry.

2.8 Summary

Overall, impingement positioning can have a large effect on the convective heat transfer inside of effusion holes. Figures 2-20 and 2-21 show a complete summary of the effects impingement position has on area-averaged Nusselt number as put forth in this study. The effusion hole length did not strongly affect the trends seen between impingement cases although the shorter L/D = 3 effusion hole had higher Nusselt numbers given the stronger influence of the entrance region. For all the cases tested, Figures 2-20 and 2-21 show that higher convective heat transfer within the effusion hole occurred when there was impingement relative to no impingement case by 10%-30%.

The highest effusion hole heat transfer came from cases with close radial spacing, close impingement jet-to-effusion plate distance when the impingement jet was positioned at either the
$\theta = 90^\circ$ and $270^\circ$ locations. As previously mentioned, these cases create a swirl at the entrance region overcoming any jet separation effects.

Increasing radial spacing, $r/D$, led to decreases in convection for most cases as seen in Figures 2-20 and 2-21, except for $\theta = 180^\circ$ as previously explained. Finally, increasing jet-to-target spacing, $H/D$, had little effect on area-averaged Nusselt number except in the direct impingement case and the $\theta = 90^\circ$ case shown in the figures, where it slightly decreased.

![Figure 2-20. Total area-averaged Nusselt number summary for L/D = 3.](image-url)
Figure 2-21. Total area-averaged Nusselt number summary for L/D = 6.

2.9 Conclusions

The cooling of combustor liners are typically done through the use of a double-wall geometry containing both an impingement jet that supplies an effusion hole. These designs heavily depend upon the convective nature of the effusion holes to contribute to the overall liner cooling.

An experiment was designed and constructed to contain a scaled-up, double-wall containing a single impingement jet feeding a single effusion hole to better understand the convective the influences on the effusion hole. The impingement hole was varied in angular and circumferential positions relative to the effusion hole for two different effusion hole lengths. In addition, the impingement jet-to-effusion hole positioning was changed. A constant heat flux was imposed on the effusion hole inner surface to evaluate the convective heat transfer at various Reynolds numbers.
All impingement cases were found to be much higher than the fully developed pipe flow correlation as expected due to the short hole lengths and the entry region effects. The experiments showed that impingement contributes to higher convective heat transfer by as much as 10%-30% depending on impingement placement. At a close radial spacing and close jet-to-target spacing, the highest convective heat transfer occurred for the jet impingement occurring at the sides of the effusion hole. The increased augmentation seen for the side impingement locations was due to an induced swirl seen in previous studies, which increases the heat transfer along the length of the effusion hole and reduces the separation region at the hole entrance. This increase was seen in the local Nusselt number measurements at the entrance and on the upstream and downstream walls along the length of the hole.

The data presented that there was little effect on the convective heat transfer within the effusion hole with differing impingement jet-to-effusion hole spacing. This particular result enforces the finding that the dominating effect on the effusion hole convection has to do with how the coolant enters into the effusion hole. Also, while the effusion hole lengths tested affected the Nusselt number values, they did not have a large effect on the general trends between impingement cases.

In summary, it was found that impingement location can have a large effect on the heat transfer inside effusion cooling holes. This study adds to the body of knowledge for impingement and effusion cooling in double-wall combustor liners. The results displayed can assist in understanding and optimizing impingement hole placement for improved in-hole heat transfer, which is an important parameter to cooling combustor liners.
Chapter 3

Effects of Jet Impingement on Flow Discharge Coefficients for Combustor Effusion Holes

3.1 Introduction

Advanced cooling techniques are required to ensure that modern combustor liners are robust enough to survive the increased temperatures they are subjected to. A double-wall scheme with effusion cooling and impingement is utilized in modern combustor liners. Impingement jets are made to cool the bottom surface of the effusion wall, as well as feeding the effusion jets. Effusion jets form a shielding film over the effusion wall surface. Understanding the discharge coefficients in these cooling schemes is vital to creating an accurate liner design. Changes in discharge coefficients can affect the amount of cooling flow and by extension the internal heat transfer through the effusion holes which plays a large role in the overall cooling of the liner walls. Impingement hole location can vary in most combustor liners due to constrictions in the design or manufacturing variation. These changes in location can affect the discharge coefficient of the effusion holes leading to changes in the internal convection of the effusion holes.

In the current literature, multiple studies have reported discharge coefficients, heat transfer, and overall effectiveness data for effusion holes [3, 6, 9, 14]. There are not many studies that have looked at discharge coefficients through effusion holes in relation to the impingement hole location or its link to internal convection within effusion holes. The contribution made by this paper is that experiments were conducted where discharge coefficients and internal heat transfer coefficients of an effusion hole were measured for many different impingement hole setups.
3.2 Literature Review

Many past studies have evaluated discharge coefficients in the field of gas turbines, mainly analyzing discharge coefficients in film cooling holes without impingement. Hays and Lampard [15] previously put together a broad review of the research in this area, breaking up the studies based on geometries such as effusion hole angles [16-18] and flow conditions such as internal [19-21] or external crossflow [22-23]. Most studies with a double wall impingement and effusion setup focused on the heat transfer and cooling effectiveness or only evaluate the discharge coefficient of the double-wall rather than the effusion discharge coefficient. Not much of the current literature has looked at discharge coefficients through effusion holes when impingement is present. Andrews et al. [24] studied the effects of Reynolds number on discharge coefficients through effusion holes and also the effects of adding impingement. The effusion holes used in their experiment ranged from an L/D of 2.4 to 10. Their results showed that past a Reynolds number of 2000, the discharge coefficient was independent of Reynolds number. Their data also revealed that impingement had little effect on the effusion discharge coefficient.

Wei-hau et al. [25] researched a double-wall impingement and effusion cooling scheme that was curved, evaluating the effects of changing the effusion hole-to-hole spacing and the effusion angle on the double-wall discharge coefficient. Yang found that decreasing the effusion angle led to an increase in the double-wall discharge coefficient. Also, as the distance between effusion holes was increased, the discharge coefficient decreased.

Zhang et al. [26] looked at the change in discharge coefficients for a double-wall impingement and effusion setup for differing area ratios. The effusion and impingement holes were arranged in a staggered format. Overall discharge coefficients were found to be similar for all area ratios across the double wall. However, the discharge coefficients were different when broken into their individual impingement and effusion discharge coefficients. At lower pressure
parameters, the discharge coefficient of the impingement wall was lower than the effusion wall. As the pressure parameter was increased, the impingement wall discharge coefficient increased past the coefficient of the effusion wall.

In the previous studies, there has been little work in analyzing the discharge coefficients of an effusion hole based on the placement of the impingement hole feeding it or its relation to the internal heat transfer in the effusion hole. This study looks to add to this body of research by presenting data on the discharge coefficients through single effusion holes as the impingement hole position is changed. The heat transfer results shown in this paper are from a previous paper by Huelsmann and Thole [27] analyzing the internal convective heat transfer of the effusion hole using the exact same experimental setup. These results will be used to try and relate the discharge coefficients from this research to some of the heat transfer effects that were seen in the previous study.

### 3.3 Impingement and Effusion Geometries

Table 3-1 and Figure 3-1 detail the geometry of the scaled-up double-wall combustor liner built for the experiments. An exit flow was not simulated for the effusion hole since, as shown by Cho and Goldstein [11], it does not have noticeable effects on the internal heat transfer. However exit flow would have an effect on the discharge coefficients for turbine cooling applications, but the focus of the research was on the entrance effects due to changing impingement location on discharge coefficients and as such no exit crossflow was tested.

The effusion hole and impingement hole diameters, D, were equal for all tests (D = 2.54 cm) as seen in Figure 3-1. Additive manufacturing was used to create the effusion hole out of a low thermal conductivity plastic. This additive effusion hole (pipe) had slots allowing for specific thermocouple placement along the entire pipe length. A single hole was drilled into three
different impingement plates to allow for various testing positions, both circumferentially and radially, in relation to the effusion hole. The thickness of the impingement plate, $t_i$, along with the diameters of the impingement and effusion holes, $D$, was the same for all plates.

Table 3-1. Impingement and Effusion Geometry

<table>
<thead>
<tr>
<th></th>
<th>Impingement</th>
<th>Effusion</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$ (deg)</td>
<td>90</td>
<td>30</td>
</tr>
<tr>
<td>$t_i / D$</td>
<td>1</td>
<td>N/A</td>
</tr>
<tr>
<td>$t_e / D$</td>
<td>N/A</td>
<td>1.5, 3</td>
</tr>
<tr>
<td>$H/D$</td>
<td>3, 6</td>
<td>N/A</td>
</tr>
<tr>
<td>$L/D$</td>
<td>N/A</td>
<td>3, 6</td>
</tr>
<tr>
<td>$r/D$</td>
<td>0, 1, 3</td>
<td>N/A</td>
</tr>
<tr>
<td>$\theta$ (°)</td>
<td>0, 90, 180, 270</td>
<td>N/A</td>
</tr>
<tr>
<td>$\Phi$ (°)</td>
<td>N/A</td>
<td>-180, -135, -90, -45, 0, 45, 90, 135, 180</td>
</tr>
</tbody>
</table>

Figure 3-1. Impingement and effusion hole geometries. In the bottom models flow is exiting the page.

Three radial locations were chosen for the impingement hole plates, $r/D = 0$ (direct impingement), 1, and 3, illustrated in Figures 3-1 and 3-2. The double-walled impingement and
effusion arrangements were made by swapping out the impingement plate and by rotating the circumferential position of the impingement plates by 90° intervals. Figure 3-1 shows the two jet-to-target spacing values, H/D = 3 and 6, that were selected for testing. 19 geometry combinations were tested in these experiments. However, the discharge coefficient data is not displayed for the 90° and 270° impingement positions in the results of this paper.

![Diagram of effusion and impingement geometries studied.](image)

### 3.4 Experimental Setup and Methods

To perform the discharge coefficient experiments, a sealed experimental rig with the large-scale impingement plates and effusion hole was created as shown in Figure 3-3. A large pipe had two endcaps placed over each side of the pipe and the impingement plate was mounted inside. The diameter of the test rig was eight effusion hole diameters so that sidewall effects could be avoided during testing. The endcap downstream of the impingement plate included the effusion hole and a foam insulation block around the effusion hole to reduce any heat loss.
Upstream of the impingement plate, the endcap had a splash plate and nozzle to diffuse the air and allow the air to enter the rig respectively. A mass flow controller was used to control the air flow through the rig.

Figure 3-3. Experimental rig with impingement plates and the additively manufactured effusion hole.

The mass flow rate of the air was set to match a chosen Reynolds number through the effusion hole. The calculation for the Reynolds number is shown in Equation 3-1. The Reynolds numbers for the experiments ranged from 2,000 to 11,000. 3 pressure taps were placed 2.29 cm upstream of the effusion hole entrance in the test rig sidewall to measure the static pressure at the inlet in different locations. 3 different measurements were taken to ensure the pressure measurement was not affected by the measurement location along the sidewall. For all tests, this static pressure was borderline identical regardless of which pressure tap was chosen. This static pressure was measured with a differential pressure transducer with respect to the atmospheric pressure at the outlet of the effusion hole. The aforementioned atmospheric pressure was measured with a separate pressure transducer. This static pressure measurement was used to
calculate the discharge coefficient, $C_d$, as shown in Equation 3-2. Utilizing both the differential pressure measurement across the effusion hole and the atmospheric pressure measurement, a pressure ratio was also calculated across the effusion hole. Pressure ratio ranged from 1 to 1.00007. This pressure ratio is very low due to the large scale of the effusion hole used for testing.

$$Re = \frac{\dot{m}}{D(\frac{\rho}{\mu})}$$

(3-1)

$$C_d = \frac{\dot{m}}{\pi(\frac{D}{4})^2 \sqrt{2 \rho \Delta P}}$$

(3-2)

As was discussed in our previous paper [17], which was focused on the convective heat transfer, slots around the length and circumference of the effusion hole had thermocouples placed inside them and filled with thermally conductive epoxy so that they were flush with the inner surface of the effusion hole as shown in the Figure 4 diagram. Placement of the thermocouples began at the entrance of the pipe (effusion hole) and continued at 10% intervals along L/D up to $\psi/D = 2.7$.

![Figure 3-4. Effusion hole instrumentation showing the heated surface with thermocouples placed radially around the effusion hole. Flow is exiting the page.](image)
A heater was made to impose a constant heat flux boundary condition on the inner pipe walls of the effusion hole which is discussed in [27]. The heater was then adhered to the inner surface of the pipe where it made contact with the aforementioned thermocouples. All tests input a constant power into to the heater. Entrance flow temperature was also measured for the effusion hole.

After steady state was reached, which usually took around 3 hours, the heater power and surface temperatures were measured along with the static pressure. Thermocouples embedded into the foam insulation block were utilized to account for the heat loss by performing a conduction analysis. For all tests the conductive heat losses were within 2%-4%. Local heat transfer coefficients, $h$, were calculated after the heat loss was subtracted from the input power. A more detailed explanation of the local heat transfer measurements can be found in our previous paper [27]. These local measurements were used in an area-weighted equation to find the area-averaged heat transfer coefficient, $\bar{h}$, presented in Equation 3-3 below. Since this paper focuses on discharge coefficients and their relation to the overall internal heat transfer, only the area-averaged heat transfer coefficient will be shown in the results. The majority of the results will show how the discharge coefficients were affected by the impingement hole location.

$$\bar{h} = \frac{\sum h \cdot A}{A_{total}} \quad (3-3)$$

**Experimental Uncertainty**

The methods laid out in Kline and McClintock [12] were used to calculate the uncertainty in the experiments. Uncertainty of the discharge coefficients was found to be around 8% for all cases. The majority of this uncertainty comes from the accuracy of the mass flow controller. Local Nusselt number uncertainty was between 3%-7%. This uncertainty increased with increasing Reynolds number and decreased for the longer effusion hole. The uncertainty increase for the increasing Reynolds numbers is attributed to a decrease in temperature difference between
the cooling flow temperature and the temperature of the effusion hole surface. For the averaged Nusselt numbers, uncertainty was under 1% for all tests. Uncertainty for the averaged Nusselt numbers was found to be under 1% for all cases. Lastly, it was calculated that there was less than 1% uncertainty in the Reynolds number for all cases.

3.5 Effect of Angular Impingement Location

Figures 3-5 and 3-6 shows the discharge coefficients for the short and long effusion holes (L/D= 3 and 6) under different impingement locations for a radial spacing, r/D = 1, and a jet-to-target spacing, H/D = 3 for a range of Reynolds numbers. The direct impingement case used r/D = 0. As expected, the discharge coefficients were fairly constant across all Reynolds numbers since tests were done at or above Re = 2000. At this point discharge coefficient is independent of Re as shown by Andrews et al. [24].

As can be seen in Figures 3-5 and 3-6 the highest discharge coefficients for the effusion hole are found at the direct impingement case along with the 180° angular position case. These discharge coefficients are almost identical across the entire range of Reynolds numbers tested. These high discharge coefficients are to be expected since the 180° position and the direct impingement case most readily feed the effusion hole without any large turning effects at the entrance. The lowest discharge coefficient is seen at the 0° impingement position where the air flow must make a large turn to enter and continue flowing through the effusion hole. The no impingement cases yield discharge coefficients that fall between the highest and lowest cases. These results are similar both in overall trends regardless of effusion L/D. At the increased L/D the discharge coefficients slightly decreased in most cases. The same information is displayed in Figures 3-7 and 3-8 only with the pressure ratio across the effusion hole displayed on the x axis instead of Reynolds number.
It was originally thought that discharge coefficient could be linked to the internal heat transfer through the effusion hole. As shown in Figures 3-9 and 3-10, which show the area-averaged Nusselt number for the same cases seen in Figures 3-5 and 3-6 along with the 90° and 270° impingement data, the highest heat transfer is attained with an angular position of either 90° or 270°. Figures 3-9 and 3-10 also include Nusselt number calculations from Gnielinski [13].

The increase in heat transfer for the side injection positions is due to an induced swirl similar to an effect found in Kohli and Thole’s research [7]. This induced swirl effect along with local heat transfer measurements from these tests is further expanded upon in our previous paper [27]. For the 0° case, the initial turn into the effusion hole causes the drop in discharge coefficient when compared to the side injection locations. Since this turn does not lead to any swirl within the effusion hole, it does not have as large of an effect on the internal heat transfer through the effusion hole even though the discharge coefficient of the effusion hole is lower. Overall, impingement had a minor effect on the discharge coefficient except in the cases of the 180° impingement location or the direct impingement case.

![Diagram](image)

**Figure 3-5.** Discharge coefficients for the short effusion hole, L/D = 3, with radial spacing r/D = 1 and jet-to-target spacing H/D = 3 for the angular impingement positions.
Figure 3-6. Discharge coefficients for the long effusion hole, $L/D = 6$, with radial spacing $r/D = 1$ and jet-to-target spacing $H/D = 3$ for the angular impingement positions.

Figure 3-7. Discharge coefficients vs. Pressure Ratio for the short effusion hole, $L/D = 3$, with radial spacing $r/D = 1$ and jet-to-target spacing $H/D = 3$ for the angular impingement positions.
Figure 3-8. Discharge coefficients vs. Pressure Ratio for the long effusion hole, L/D = 6, with radial spacing r/D = 1 and jet-to-target spacing H/D = 3 for the angular impingement positions.

Figure 3-9. Area-averaged Nusselt number for the short effusion hole, L/D = 3, with constant radial spacing, r/D = 1, and constant jet-to-target spacing, H/D = 3.
Figure 3-10. Area-averaged Nusselt number for the long effusion hole, L/D = 6, with constant radial and jet-to-target spacing.

3.6 Effect of Impingement Radial Spacing

The effect of radial spacing on the discharge coefficient is shown in Figures 3-11 and 3-12. These figures show the discharge coefficients across a range of Reynolds numbers for all angular impingement positions for both an r/D = 1 and an r/D = 3. Both effusion hole lengths are also shown. The jet-to-target spacing was kept constant at H/D = 3. Once again effusion length has little effect on the discharge coefficients, only leading to minor decreases for most cases at the longer length. At 0° there is no large change in discharge coefficient regardless of r/D. However, at 180° the increase in radial spacing to r/D = 3 leads to a large drop in discharge coefficient. This is the case for both effusion lengths. At an r/D = 3 there is very little difference in discharge coefficient regardless of angular position since all impingement setups must first impinge on the effusion wall before entering the effusion hole. Figures 3-13 and 3-14 show the area-averaged Nusselt number for the same cases as Figures 3-11 and 3-12. Interestingly, the
large drop in discharge coefficient seen at 180° for the extended radial distance is not tied to a similar change in the internal heat transfer of the effusion hole. There is only a minor increase in the area-averaged Nusselt number for the 180° position at the increased radial spacing. The 270° case was not shown in these heat transfer figures since it was so similar to the 90° location.

Figure 3-11. Discharge coefficients for the short effusion hole, $L/D = 3$, with varying radial spacing and constant jet-to-target spacing, $H/D = 3$, for each angular impingement position.
Figure 3-12. Discharge coefficients for the long effusion hole, L/D = 6, with varying radial spacing and constant jet-to-target spacing, H/D = 3, for each angular impingement position.

Figure 3-13. Area-averaged Nusselt number for the short effusion hole, L/D = 3, with varying radial spacing and constant jet-to-target spacing, H/D = 3.
Figure 3-14. Area-averaged Nusselt number for the long effusion hole, L/D = 6, with varying radial spacing and constant jet-to-target spacing, H/D = 3.

3.7 Effect of Jet-to-Target Spacing

The results of increasing the jet-to-target spacing from H/D = 3 to H/D = 6 is shown in Figures 3-15 and 3-16, which show the discharge coefficient for a range of Reynolds numbers at the close radial spacing, r/D = 1, for both the short and long effusion hole at the varying angular impingement locations. Both H/D = 3 and H/D = 6 are shown in the results. There is no discernible change in discharge coefficient for the 0° location across either jet-to-target spacing. This matches with results seen in Andrews et al. [24] where no change in discharge coefficient was seen unless jet-to-target spacing was reduced to 1. However, as seen previously with the increased radial spacing, when impingement is at the 180° position the discharge coefficient decreases with increasing H/D. Also, as seen in previous results, the length of the effusion hole did not strongly affect the final results. Figures 3-17 and 3-18 follow the format of Figures 3-15 and 3-16 except the area-averaged Nusselt number is evaluated. The
lowest Nusselt number was seen at the 180° impingement position regardless of L/D or H/D which was to be expected since the highest discharge coefficient was seen at that location. For the 0° location, the discharge coefficient was constant regardless of L/D which matches what was seen in the Nusselt number plots.

Figure 3-15. Discharge coefficients for the short effusion hole, L/D = 3, with constant radial spacing, r/D = 1, and varying jet-to-target spacing for each angular impingement position.
Figure 3-16. Discharge coefficients for the long effusion hole, L/D = 6, with constant radial spacing, r/D = 1, and varying jet-to-target spacing for each angular impingement position.

Figure 3-17. Area-averaged Nusselt number for the short effusion hole, L/D = 3, with constant radial spacing, r/D = 1, and varying jet-to-target spacing.
Figure 3-18. Area-averaged Nusselt number for the long effusion hole, L/D = 6, with constant radial spacing, r/D = 1, and varying jet-to-target spacing.

3.8 Summary

Figures 3-19 and 3-20 show a summary of all the major impingement changes and their effects on the discharge coefficient for both the short and long effusion hole. These figures display the discharge coefficients at a Reynolds number of 8000. Overall, changing impingement locations can lead to large changes in discharge coefficients in certain situations, mainly changes in the angular θ positions as shown in the figures. For the most part, changing the radial spacing or the jet-to-target spacing led to very minor changes in discharge coefficient except at specific impingement locations like 180° as discussed earlier in the paper or the direct impingement case as seen in Figures 3-19 and 3-20. Increasing the effusion hole length led to a slight decrease in discharge coefficient for most cases.
Figure 3-19. Discharge Coefficient summary for L/D = 3.

Figure 3-20. Discharge Coefficient summary for L/D = 6.
3.9 Conclusions

A double-wall design is typically used to cool combustor liners with an impinging jet feeding an effusion hole. An understanding of the discharge coefficients within the effusion holes is necessary to properly design the liner to supply the amount of coolant needed. A large scale double-wall design with a single impingement and effusion hole was built to understand how the discharge coefficient through the effusion hole changes as the impingement hole position is altered. The discharge coefficients link to internal convective heat transfer in the effusion hole was also explored.

The impingement hole angular position or impingement type did lead to changes in the discharge coefficient at the close radial spacing and close jet-to-target spacing. The experiments showed that the 180° location and direct impingement position had the highest discharge coefficient, above 0.8. The lowest discharge coefficient was seen at the 0° position with a value around 0.6. The no impingement case, 90°, and 270° had discharge coefficients between 0.6 and 0.7. The increase in heat transfer for the 90° and 270° cases can be linked to a swirl appearing inside the effusion hole. The 0° position does have a lower discharge coefficient but it was thought that this does not correlate to increased heat transfer because the bulk of the increased discharge coefficient comes from the initial turn into the effusion hole rather than any flow effects inside the hole.

Experiments revealed that radial spacing had very little effect on the discharge coefficients through the effusion hole except at 180°. The data also showed that jet-to-target spacing had a minimal effect on discharge coefficients except at the 180° or direct impingement location which decreased with increasing H/D. Effusion hole length only had a noticeable effect on discharge coefficient at the close radial spacing with the close jet-to-target spacing and discharge coefficient decreased slightly in all cases for the higher L/D.
To summarize, the angular impingement location led to the changes in discharge coefficient especially at the 180° position or the direct impingement position. Effusion length had minor effects on discharge coefficient. Radial spacing and jet-to-target spacing only had an effect when the impingement hole was at the 180° location. This study adds to the current research for impingement and effusion double-wall combustor liner cooling.
Chapter 4

Conclusions

A double-wall, composed of impingement and effusion holes, is an important cooling technology for combustor liners. The convective heat transfer through the effusion holes plays a large role in the overall cooling of the combustor liner, and discharge coefficient measurements are needed to predict the flow and ultimately to size the cooling holes. In this thesis, an experiment was designed utilizing a large-scale double-wall with a single impingement and effusion hole to measure local convective heat transfer coefficients inside the effusion hole as well as discharge coefficients through the effusion hole. Local measurements were also used to calculate an average convective heat transfer coefficient. Angular position, radial spacing, and jet-to-target spacing of the impingement hole were varied in relation to the effusion hole for two effusion lengths. The effusion hole was additively manufactured and a heater was designed to implement a constant heat flux boundary condition at the inner surface of the effusion hole.

The average heat transfer measurements showed that any impingement leads to an improvement over the no impingement case by up to 30% as impingement position is changed. The highest convective heat transfer was seen at the side impingement locations with the close radial and jet-to-target spacing due to a swirling effect as the air entered the effusion hole. Unsurprisingly, analysis of the local measurements showed that Nusselt number was generally higher at the side of the effusion hole where the impingement hole was located, at least at the inlet measurements. Changes in jet-to-target spacing were shown to have little effect on heat transfer measurements and increase effusion hole length led to decreases in Nusselt number although the trends between impingement geometries were unchanged.

Discharge coefficient measurements only had large changes when angular impingement was altered at the close radial spacing and jet-to-target spacing. The highest discharge
coefficients were above 0.8, seen at the 180° position and the direct impingement position. The lowest was attained at the 0° location, just below 0.6. The decreased discharge coefficient at the 0° locations is most likely linked to the large turn that the airflow must undergo to enter the effusion hole. No strong link between effusion hole convective heat transfer and the discharge coefficients was seen in the results. Overall, the results seen in this study could be used by designers to aid in the optimization of their impingement hole placement for double-wall combustor liner designs.

4.1 Recommendations for Future Work

The results displayed in this study could be expanded in multiple ways. Currently, the local heat transfer measurements are only measured around the entire circumference of the effusion hole in three locations. Increasing the number of measurement locations could better capture the swirl effect described earlier in the study. A study focusing impingement hole placement at 45° intervals rather than 90° could help designers further understand sensitivity of impingement hole placement on effusion hole convective heat transfer. A study with multiple impingement and effusion holes is also recommended to more accurately mimic the double-wall design. Finally, measuring discharge coefficients for the above studies would also be of interest.
References


