Jet impingement cooling configurations for gas turbine combustion

Eric Ian Esposito
Louisiana State University and Agricultural and Mechanical College

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JET IMPINGEMENT COOLING CONFIGURATIONS
FOR GAS TURBINE COMBUSTION

A Thesis

Submitted to the Graduate Faculty of the
Louisiana State University and
Agricultural and Mechanical College
in partial fulfillment of the
requirements for the degree of
Master of Science in Mechanical Engineering

in

The Department of Mechanical Engineering

by
Eric Ian Esposito
B.S., Louisiana State University, 2004
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NOMENCLATURE

\[ z \] jet to target plate spacing
\[ d \] diameter of jet orifice
\[ x \] streamwise distance between jets
\[ y \] spanwise distance between jets
\[ L \] length of the impingement hole
\[ H \] height of impingement plate from target plate
\[ S_i \] spanwise length of impingement portion of Corrugated Wall
\[ S_t \] spanwise length of bypass portion of Corrugated Wall
\[ h \] local convection heat transfer coefficient
\[ k \] thermal conductivity
\[ \alpha \] thermal diffusivity
\[ \nu \] kinematic viscosity
\[ \text{Nu}_D \] Nusselt number
\[ \text{Re}_D \] Reynolds number
\[ M \] Mach number
\[ U \] average jet velocity
\[ T_i \] initial temperature of test section
\[ T_w \] local wall temperature
\[ T_m \] mainstream flow temperature
\[ t \] time
\[ \Delta t \] time step
\[ \tau_j \] time step for Duhamel’s superposition theorem
\[ T_{mj} \] temperature step for Duhamel’s superposition theorem
ABSTRACT

Impinging jets are commonly used to enhance heat transfer in modern gas turbine engines. Impinging jets used in turbine blade cooling typically operate at lower Reynolds numbers in the range of 10,000 to 20,000. In combustor liner cooling, the Reynolds numbers of the jets can be as high as 60,000. The present study is aimed at experimentally testing two different styles of jet impingement geometries to be used in backside combustor cooling. The higher jet Reynolds numbers lead to increased overall heat transfer characteristics, but also an increase in crossflow caused from spent air. The crossflow air has the effect of rapidly degrading the downstream jets at high jet Reynolds numbers. In an effort to increase the efficiency of the coolant air, configurations designed to reduce the harmful effects of crossflow are studied. Two main designs, a corrugated wall and extended ports, are tested. Variations of these configurations were tested for both sparse and dense arrays.

Local heat transfer coefficients are obtained for each test section through a transient liquid crystal technique. Results show that both geometries reduce the crossflow induced degradation on downstream jets, but the individual geometries perform better at different Reynolds numbers. The extended port and corrugated wall configurations show similar benefits at the high Reynolds numbers, but at low Reynolds numbers, the extended port design increases the overall level of heat transfer. This is attributed to the further developed jet profile that exits the tubes. The benefit of the developing velocity profile diminishes as jet velocities rise and the air has less time to develop prior to exiting.
CHAPTER 1: INTRODUCTION

Gas turbine engines have proven to be an effective means of converting fuel into usable power either through direct shaft power, or thrust produced from the high momentum exhaust gasses. Prior to World War I, gas turbine engines were nothing more than a concept conceived by physicists and engineers of the era. After the war, governments started to pursue the idea, looking for effective high output propulsion systems to power the greatest addition to the military arsenal, the airplane. The first gas turbine engines were aimed at performance at the expense of efficiency and emissions. Modern gas turbine military fighters took the world by surprise during World War II, and led to new markets for the device, commercial air travel and industrial power generation. Both markets needed gas turbine engines that were designed for reliable and efficient operation instead of overall performance. With the formation of the Environmental Protection Agency in 1970, restrictions on emission levels from gas turbines were scrutinized.

The fundamental thermodynamic principles that make gas turbine engines feasible are based on the properties of the working fluid, mainly air. As air is compressed and heated with the addition and burning of fuel, the energy released from the air expanding through the turbine is greater than the energy required to compress the air. This action is magnified as the fluid is compressed to higher pressures and raised to higher temperatures. Seeking an increase in thermodynamic efficiency, engineers designed engines with higher compressor pressure ratios and turbine inlet temperatures. Temperatures were over or near the failure point of the materials used to build the turbines, requiring extensive cooling of the internal parts exposed to the flow. Coolant air is pulled from the compressor section prior to the combustion chamber and routed throughout the engine to cool all parts in the hot gas path, including the combustor liner and turbine section.
1.1 Combustor Liner Cooling

Combustor liners required significant amounts of cooling air since they are exposed to both convective and radiative heating from the combustion process. Turbine engineers incorporated various methods to cool the liner including film cooling and jet impingement cooling. The combustion chamber is comprised of an internal liner exposed to the hot gas, and an outer shell used to separate the hot liner from other engine parts and to create a passage for the coolant air. Film cooling absorbs the heat load on the liner by allowing coolant air to pass along the backside of the liner, and then enter the combustion chamber through holes and flow close to the inner walls. This coolant mixes with the hot combustion gasses reducing the near wall temperature of the flow and therefore the convective heat flux into the combustor liner. Figure 1 shows a typical film cooled combustor liner.

![Figure 1: Combustor Liner Film Cooling](image)

Film cooling was the preferred way of controlling temperatures since substantial heat loads could be dissipated with minimal coolant air. Although beneficial for cooling the combustor liner, film cooling does adversely affect the combustion process leading to increased emissions. The air/fuel mixture entering the combustor is initially rich since additional air needed for combustion comes from the coolant air entering through the liner walls which mixes with the mainstream during combustion. The coolant air entering the combustion chamber leads to nonuniform temperature distributions causing incomplete combustion and production of NO\textsubscript{x}, CO, and unburned hydrocarbon emissions.
In the 1970’s, restrictions began to be placed on the emission levels of gas turbines and have only grown stronger in the passing years. Initial methods to reduce emissions relied on lean combustion and steam injection. Lean combustion engines use low fuel to air ratio combustors to decrease the combustion temperature and reduce the coolant required for film cooling the liner. This technique results in more complete combustion and lower unburned hydrocarbon emissions. The use of steam injection into the combustor further reduces the combustion temperature and controlled NO\(_x\) levels but requires the addition of steam to the engine. Unfortunately, neither of these two methods have a significant affect on limiting the levels of CO.

Modern industrial gas turbine engines use Dry, no steam injection, Low Emission (DLE) combustors to overcome emission restrictions while maintaining efficiency. DLE combustors use lean, pre-mixed technology previously discussed and further limit emissions by reducing or eliminating the introduction of cool air used for film cooling on the liner. The heat load on the liner must be dealt with in other environmentally friendly ways.

**1.2 Jet Impingement Cooling**

Jet impingement cooling is an enhanced heat transfer method capable of cooling a combustor liner without injecting cool air directly into the combustion chamber. Cooling the liner from the backside enables engineers to dissipate the heat load and maintain more uniform temperatures in the combustion region needed for efficient combustion. Figure 2 shows a typical combustor liner using backside jet impingement cooling.

An impingement array is comprised of a *jet plate* typically having round holes which produces the impinging jets. The jets strike the surface to be cooled, referred to as the *target plate*. Traditionally, the structure of an impinging jet is broken down into three parts, the potential core, shear layer, and the wall jet. Figure 3 shows the impinging jet structure and the associated regions.
At the discharge of the jet plate, the velocity profile of the jet is relatively uniform. As the jet discharges, viscous forces acting in the shear layer cause the jet velocity profile to develop and expand. The potential core of the jet is defined as the region where the viscous forces have little or no effect on the velocity profile. Once the jet strikes the target plate, the wall jet is formed as the fluid travels along the wall. Again viscous forces act on the fluid decreasing the peak velocity and causing the wall jet to thicken as it moves away from the stagnation point.
In jet impingement arrays consisting of multiple jets, the ideal jet structure is significantly altered due to jet-to-jet interactions. The wall jet region is forced upwards as in collides with the adjacent wall jet in the array. This upward movement of the fluid amplifies the strength of the ring vortex near the jet exit. As the vortex strengthens, the mixing in the shear layer of hot spent air and coolant air decreases the effectiveness of the jet. Jet arrays, which are generally confined on three sides, contend with crossflow caused from the spent air of upstream jets intersecting downstream jets before exiting. The crossflow of hot spent air degrades the heat transfer characteristics of downstream jets and limits the practical size of jet impingement arrays.

Since most experiments are conducted at conditions suitable for accurate testing and not actual engine conditions, dimensionless parameters must be used to scale results. Impinging jets are characterized by the jet Reynolds and Mach numbers, while the heat transfer is characterized by the Nusselt number. The Reynolds number, \( \text{Re}_D \), is defined in Equation 1

\[
\text{Re}_D = \frac{U d}{\nu} \quad (1)
\]

where \( U \) is the mean jet velocity at the discharge, \( d \) is the jet hole diameter, and \( \nu \) is the kinematic viscosity of air. The Mach number, \( M \), is a dimensionless velocity term defined in Equation 2 as

\[
M = \frac{U}{a} \quad (2)
\]

where \( a \) is the local speed of sound at the jet discharge. The heat transfer coefficient, \( h \), is nondimensionalized by relating it to the hole diameter, \( d \), and the thermal conductivity, \( k \), of the fluid. Equation 3 defines the Nusselt number, \( \text{Nu}_D \).

\[
\text{Nu}_D = \frac{h d}{k} \quad (3)
\]
1.3 Literature Survey

Due to the complexity of impinging jet structures in actual arrays, researchers have systematically studied the effects of geometrical parameters on the heat transfer characteristics of impinging jets. Dano et al. researched the effects of nozzle geometry on the flow characteristics and heat transfer performance. San and Lai studied the effect of jet-to-jet spacing on heat transfer in staggered arrays. Cheong and Ireland experimentally measured local heat transfer coefficients under an impinging jet with low nozzle-to-plate, $z/d$, spacings. Several others have studied the effect of crossflow on jet structure and heat transfer including Florschuetz, and Kercher and Tabakoff. Both Florschuetz, and Kercher and Tabakoff developed correlations to predict the effect of crossflow on jet impingement heat transfer for inline and staggered arrays which are still used today in jet impingement research. Bailey and Bunker studied the effect of sparse and dense arrays for large numbers of jets. Hebert and Ekkad investigated the effect of a streamwise pressure gradient for an inline array of sparse and dense configurations.

As more information on geometrical parameters and their effect on impinging jets became available, others studied ways of increasing the jet effectiveness through target surface modification. Surface geometries such as trip strips, protrusions, or dimples can significantly alter the jet structure and potentially provide enhanced heat transfer. Ekkad and Kontrovitz used a dimpled target surface. This concept proved inadequate in heat transfer enhancement and actually showed a drop in performance. Another concept employed in many internal cooling configurations is trip strips. Small strips placed on the surface break down boundary layers, increase local turbulence levels, and enhance heat transfer. The use of trip strips was studied in detail by Han et al. for internal channel flow, and later by Hebert and Ekkad in jet impingement configurations.

Transient liquid crystal techniques are the primary means of determining local heat transfer coefficients in jet impingement research. Ekkad and Han put forth a single color capturing method which was used in Part I of the work presented. Gillespie, Wang, and Ireland extended this technique to use wide band liquid crystal paints to capture temperatures over an extended time and temperature range. Their technique also used a fine mesh heater to provide near step changes in mainstream gas temperatures. This technique is followed in Part II of the study.
1.4 Experimental Objectives

The motivation behind this study is to develop and test innovative impingement configurations that are capable of reducing the degradation of heat transfer performance in downstream jets from spent air crossflow. Two main impingement geometries designed to redirect spent air away from downstream jets will be tested using a transient liquid crystal technique that provides detailed local heat transfer coefficients at the target plate. Experiments will be conducted to study the effect of corrugated wall and extended port configurations in sparse and dense arrays. Several variations of each geometry will be investigated to assist in future design changes.
CHAPTER 2: EXPERIMENTAL APPARATUS

A two part study was conducted to ensure all geometrical parameters were taken into consideration. Part I investigated the impingement configurations in a sparse arrays, while Part II focused on dense arrays. Two test rigs were built and will be described separately.

2.1 Compressed Air Supply

An Atlas Copco GR 110 two-stage screw compressor was used to compress the air used in the study. The compressor employs a cooling loop to keep outlet air at around 25°C which is needed for effective operation of the air dryer. After exiting the compressor, the pressurized air is sent to a desiccant dryer system to remove any moisture in the air. After exiting the dryer, compressed air is stored in a 45 m³ pressure vessel until needed. This storage tank reduces the line pressure fluctuations at the test rig from periodic cycling of the compressor.

2.2 Flow Metering

Compressed air from the storage tank is routed to the test rig where it passes through a manual pressure regulator used to adjust the flow. After exiting the regulator, the flow was sent through a 15 micron filter to remove any particles in the line. An orifice flow meter is used to measure the flowrate entering the test rig. The orifice meter is made of 5 cm ID steel schedule 80 pipe with an orifice diameter of 2.1 cm. The orifice plate is held in place by flanges which hold pressure taps 2.54 cm. before and after the orifice plate to measure the differential pressure across the plate.

2.3 Temperature Measurements

Temperature measurements were taken with type -k thermocouples. The thermocouples were connected to an Omega instruNET analog-to-digital conditioning box that uses an electronic reference junction. The signals from the conditioning box run through an Omega instruNET PCI interface to a Pentium III computer running instruNET software to simultaneously record up to 8 thermocouples. 40 gage type –k thermocouples
were used to measure air temperatures and a 13 micron thick foil thermocouple was used to read surface temperatures of the target plate.

2.4 Experimental Test Rig for Part I

The test rig built for Part I of the study is shown in Figure 4.

![Experimental Test Rig for Part I]

The 3 kW heater used in this rig was composed of an outer tube filled with helical shaped resistance heating elements stretched parallel to the flow direction. Although these heaters are readily available and inexpensive, the large thermal capacitance of the heating elements results in an exponential increase in temperature after the heater is switched on. The heater, pipes, and inner walls of the plenum were covered in insulation to reduce thermal losses before the test section. The plenum section allows the flow to decelerate and expand before the jet plate to ensure even flow distribution across the array. The test sections were manufactured out of acrylic and ABS plastics to reduce any thermal losses to the jet plate. The target plate was manufactured from clear acrylic to allow the camera to view the liquid crystal paint on the inner wall.
A Pulnix TMC-7DSP RGB camera was used to capture video frames of the liquid crystal coated target plate during each test at a resolution of 640x480. A Pentium III computer running Optimas imaging software captured video frames using a PCI frame grabber board.

2.5 Experimental Test Rig for Part II

Figure 5 shows the layout of the test rig used in Part II. Although similar to the previous rig, this setup used a mesh heater near the test section instead of the tube heater before the plenum.

The test rig was designed to provide improved heater response by reducing the thermal capacitance of the heater and placing it close to the test section. The heater was placed at the exit of the plenum and a spacer separated it from the jet plate. The camera used was a Sony XCD-X710CR RGB camera capable of capturing high resolution images of 1024x768 pixels at 15 frames per second. The camera transferred digital frame...
information to a Dell Pentium IV computer via firewire interface. I-Fire software included with the camera recorded the frames for processing.

2.5.1 Mesh Heater

The mesh heater concept developed by Gillepsie et. al. has the capability of providing an instantaneous temperature step to the mainstream air. The mesh used to build the heater was 304 Stainless Steel woven wire mesh with a 20 micron wire diameter. The free area of the mesh was 28% which helped in reducing the free stream turbulence before the test section. The heater frame was made out of Garrolite G-11, a high temperature fiberglass material. Figure 6 shows the layout of the mesh heater.

![Figure 6: Mesh Heater](image)

The low electrical resistance of the dense wire mesh necessitated the use of a low voltage, high amperage power source. Power was supplied to the heater using a Miller Dynasty 200DX TIG welding machine capable of supplying up to 3.5 kW of DC power. During initial testing of the heater, the test section was removed and an infrared (IR) camera was focused on the heater mesh. The IR camera was able to record frames at a rate of 60 Hz. It was found that once the heater was turned on, the steady state
temperature of the mesh was achieved in the next frame. This proves that the response of the heater is less than 33 milliseconds and agrees with a theoretical estimation of the response based on the capacitance of the wires and operating temperatures of about 27 milliseconds. Figure 7 shows the thermal response of the mesh heater measured with a 40 gage type $-k$ thermocouple.

![Mesh Heater Response](image)

Figure 7: Thermal Response of Mesh Heater

### 2.6 Impingement Configurations

A total of eight test geometries, Parts I and II consisting of four test sections each, were tested. All configurations consisted of 50 circular holes having a diameter of 3.175 millimeters arranged in five rows in the spanwise direction and ten holes in the streamwise direction. Data was collected on the three middle rows of jets to reduce the wall effects on the impinging jets in the streamwise direction. Part I was to investigate the new impingement configurations in a sparse array of jets. Part II used the results of Part I to further improve the designs and study their effects in a dense array. A baseline
geometry was run for each set of experiments as a reference to compare the corrugated wall and extended port geometries.

All test section geometrical parameters are stated in nondimensionalized fashion in order to appropriately scale the results. As with the Reynolds and Nusselt numbers, length units are given with respect to the hole diameter. All length units are recorded as a ratio of the physical length to the hole diameter.

2.6.1 Baseline

The baseline impingement plate consisted of a flat jet plate with circular holes that were 1 hole diameter in length \((L/d = 1)\). Figure 8 shows the baseline configuration. The spacing of the holes in the spanwise, \(y/d\), and streamwise, \(x/d\), directions was 11 for Part I and 5 for Part II.

![Figure 8: Baseline Configuration](image-url)
The baseline configuration which is confined on three sides of the array suffers from decreased heat transfer characteristics in downstream jets due to the presence of crossflow. Figure 9 graphically represents the crossflow in an impingement array and shows how the downstream jets are pushed away from the target plate.

![Figure 9: Crossflow in an Impingement Array](image)

The following geometries were designed with this problem in mind and allow alternate paths for the spent air to exit the array with minimal disruption to downstream jet structures.

2.6.2 Corrugated Wall

The first new geometry tested was a corrugated wall design. The corrugations in the wall allow spent air from upstream jets to exit the impingement array without interfering with the downstream jets. Figure 10 shows how the spent air is expected to exit the impingement array.

![Figure 10: Routing of Spent Air in Corrugated Channel](image)
In addition to routing the spent air around the downstream jets, the increase in cross-sectional area of the section due to the corrugations also decreases the overall crossflow velocity. The layout of this configuration with the labeled geometrical parameters is pictured in Figure 11.

![Figure 11: Corrugated Wall Configuration](image)

Two major variations of this design were tested. Part I contained two versions that had differing widths of the corrugations, while Part II tested only one version.
2.6.3 Extended Ports

The final geometry tested was an extended port design. This design offers the highest cross-sectional area for crossflow therefore reducing the overall crossflow velocity the greatest. The additional benefit of the extended ports is to allow a partially developed flow to exit the jet increasing the peak jet velocity and further reduce crossflow effects by increasing the jet to crossflow velocity ratio. Figure 12 shows the extended port configuration.

![Figure 12: Extended Port Configuration](image)

A variation to the extended port design was also tested on the dense arrays called the variable extended ports. The length of the extended ports was linearly varied from the first to the last rows of jets. All ports were of uniform length in the spanwise direction. Figure 13 gives a graphical depiction of the configuration.

![Figure 13: Variable Extended Ports](image)
Table 1 outlines all geometrical parameters in dimensionless form for all eight configurations tested.

<table>
<thead>
<tr>
<th>Part I</th>
<th>Baseline</th>
<th>Corrugated (1)</th>
<th>Corrugated (2)</th>
<th>Extended Ports</th>
</tr>
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<tbody>
<tr>
<td>$z/d$</td>
<td></td>
<td>3.33</td>
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<td></td>
</tr>
<tr>
<td>$H/d$</td>
<td></td>
<td>6.66</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$x/d$ and $y/d$</td>
<td>11</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$L/d$</td>
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<td>1</td>
<td>4.33</td>
<td></td>
</tr>
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<td>5.17</td>
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<td></td>
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<tr>
<td>$S_{t}/d$</td>
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<td>5.83</td>
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<table>
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<th>Extended Ports</th>
<th>Variable Extended</th>
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<tr>
<td>$z/d$</td>
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<td>3</td>
<td></td>
<td>Vary from 6 to 3</td>
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<tr>
<td>$H/d$</td>
<td></td>
<td>6</td>
<td></td>
<td>5.17</td>
</tr>
<tr>
<td>$x/d$ and $y/d$</td>
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<td></td>
<td></td>
<td>5.83</td>
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<tr>
<td>$S_{t}/d$</td>
<td>1.65</td>
<td></td>
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</table>
CHAPTER 3: EXPERIMENTAL METHODOLOGY

Achieving a reliable experimental methodology was necessary to accurately determine the local heat transfer characteristics of each impingement configuration. In the first part of the study a simplified transient liquid crystal method was used. In an effort to increase accuracy and repeatability, a more detailed approach involving the regression analysis of an overconstrained system of equations was developed for Part II.

3.1 Heat Transfer Theory

3.1.1 Steady State Heat Transfer

The experimental determination of heat transfer coefficients can be accomplished using steady-state or transient techniques. The steady-state method is generally preferred whenever possible and usually offers the least experimental uncertainty due to the lack of time dependence on the data collected. Steady-state experiments used to compute heat transfer coefficients usually involve the use of a thin foil heater capable of providing a uniform heat flux. Consider flow over a flat plate with a uniform heat flux heater which is insulated on the outer side as shown in Figure 14.

Figure 14: Uniform Heat flux Boundary Condition on an Insulated Plate
At steady-state conditions, an energy balance performed at the heater yields Equation 4

\[ q'' = h(T_w - T_m) \]  

where \( q'' \) is the heat flux of the heater, \( h \) is the heat transfer coefficient, \( T_w \) is the wall temperature, and \( T_m \) is the mainstream temperature of the fluid. This technique is applicable to situations where the heat transfer gradient, and therefore temperature gradient in the heater, is low enough to neglect lateral conduction through the heater and wall. Since the characteristics of jet impingement cooling are enhanced heat transfer near the stagnation point and rapidly decays between jets, this technique would lead to lateral conduction problems which can be difficult to quantify.

3.1.2 Transient Heat Transfer

To overcome the problems with the steady-state approach, a transient method was used to determine local heat transfer coefficients. Instead of applying a uniform wall heat flux, the transient method requires a step change in mainstream fluid temperature. Consider flow over an infinitely thick plate as depicted in Figure 15.

\[ \text{Figure 15: Flow Over an Infinitely Thick Plate} \]
In the transient case, the wall and mainstream temperature are uniform at a given initial temperature, \(T_i\). For all time \(t > 0\), the mainstream temperature is increased as a step response and the convective boundary condition provides a heat flux to the wall. Equation 5 is the result of a thermal energy balance of the wall for \(t > 0\), where \(T\) is the local wall temperature and \(\alpha\) is the thermal diffusivity of the wall.

\[
\left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) = \frac{1}{\alpha} \frac{\partial T}{\partial t} \quad (5)
\]

If lateral conduction in the plate is neglected and only conduction into the plate is considered, Equation 5 reduces to Equation 6.

\[
\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \quad (6)
\]

To solve this second order partial differential equation, an initial condition, and two boundary conditions are necessary. Equation 7 gives the initial condition for the temperature of the wall at \(t = 0\). The two boundary conditions at \(x = 0\) and infinity are given in Equations 8 and 9.

\[
T = T_i \text{ at } t = 0 \quad (7)
\]

\[
-k \frac{\partial T}{\partial y} = h(T_w - T_m) \text{ at } x = 0 \text{ and } t \geq 0 \quad (8)
\]

\[
T = T_i \text{ at } x = \infty \text{ for all } t \quad (9)
\]
Solving the partial differential equation of Equation 6 at the point \( x = 0 \), gives the transient wall temperature response due to the convective heat flux into the wall for all time. The solution is given in Equation 10

\[
\frac{T_w - T_i}{T_m - T_i} = 1 - \exp\left(\frac{h^2 \alpha t}{k^2}\right) \text{erfc}\left(\frac{h}{\sqrt{\alpha t}}\right)
\]

(10)

where \( k \) is the thermal conductivity of the wall. Other than the wall material properties and the initial temperature, only one transient wall temperature is needed with the corresponding time that the temperature takes place to solve for \( h \).

3.1.3 Validation of One Dimensional Semi-Infinite Assumption

A major assumption of the thermal model presented is an infinitely thick wall. Although this is an abstract concept, the model will remain valid for finite wall thicknesses and finite time periods. As long as the thermal pulse into the wall does not reach the outer surface, the boundary condition in Equation 9 remains the same. Therefore, the target plate used in the experiment must be adequately thick to ensure that the outer temperature of the wall does not change through the course of the experiment. Thermal diffusivity is the material property that quantifies how fast a thermal pulse travels through a given material. The maximum depth of penetration of the thermal pulse is only dependent on the material property and time as given in Equation 11.

\[
\text{Penetration Depth} = \sqrt{\alpha t}
\]

(11)

The target plate was made of acrylic having a low thermal conductivity and thermal diffusivity. The penetration depth for a typical heat transfer test was approximately 2 millimeters.

The assumption one dimensional conduction perpendicular to the wall and neglecting the lateral conduction in the plate was made based on the small penetration depth of the thermal pulse. Since a temperature gradient must be present for heat flux to travel via conduction, the only area possible for lateral conduction to occur in within the
thermally affected region behind the thermal pulse traveling through the material. The maximum penetration depth is relatively short and therefore minimal area is subjected to lateral conduction. Calculations showed that less than 1% of the total heat flux into the target plate was lost to lateral conduction.

3.2 Thermochromic Liquid Crystals

To determine local wall temperatures during the transient heat transfer experiment, thermochromic liquid crystal paint was tracked with a RGB camera. Cholesteric liquid crystals have characteristics that allow them to change color depending on their temperature. The liquid crystal structure changes with temperature and reflects light at varying wavelengths according to temperature. Liquid crystals alone are fairly sensitive to harsh environments. To overcome this, manufacturers micro encapsulate the liquid crystals in polymer spheres and add these micro encapsulated liquid crystals to a binder solution to be used as sprayable paint coatings. The liquid crystal paint can be ordered from the manufacturer, Hallcrest, to react over specified temperature ranges. In Part I, SPN R30C5W liquid crystal paint was used, which changed color between 30°C and 35°C. In Part II, wideband SPN R20C20W liquid crystal paint was used, which changed color between 20°C and 40°C. The liquid crystal paint was applied to the target plate using a fine spray airbrush to apply thin coats of paint evenly. Following application of the liquid crystals, a thin layer of black paint was applied to give the reflected colors a background to reflect against. The overall thickness of the layer was approximately 10-15 microns.

3.3 Single Temperature Method

A single temperature method was used in Part I of the study on sparse arrays. Equation 10 was used to solve for $h$ by recording the time a specific wall temperature occurred on the target plate for every location. Only one color band, green which is the most dominant, was tracked with the RGB camera. The narrow band liquid crystal used was calibrated and the peak in green intensity of the paint was found to occur at 31.4 °C.

In the actual experiment the heater response was not adequate to assume a step mainstream gas temperature. The heater’s internal capacitance caused the temperature rise to follow somewhat of an exponential response. This response was captured with a
thermocouple located at the entrance to the impingement plate. The thermocouple data was then discretized into small time steps at a frequency of 4 Hz. Figure 16 shows a typical temperature response and the discretization of the data.

![Discretized Temperature Response](image)

Figure 16: Discretized Temperature Response

With this discretized data, the original transient conduction equation (Eq. 10) can be rewritten using Duhamel’s superposition theorem which sums small temperature steps of the mainstream gas temperature to solve for the local heat transfer coefficient. Equation 12 incorporates Duhamel’s superposition and was used in the data reduction

\[
T_w - T_i = \sum_{j=1}^{N} \left[ 1 - \exp \left( \frac{h^2 \alpha (t - \tau_j)}{k^2} \right) \right] \text{erfc} \left( \frac{h\sqrt{\alpha (t - \tau)}}{k} \right) \Delta T_m \tag{12}
\]

where \( \tau_j \) is the discretized time step, and \( \Delta T_m \) is the discretized mainstream temperature step. This equation accounts for the exponential rise in mainstream gas temperature and was used to compute heat transfer coefficients for Part I.
3.4 Regression Analysis Method

The test rig designed for Part II used a slightly different technique to further reduce experimental error. The methodology used in Part I relies on the summation of small temperature and time steps associated with the transient nature of the mainstream temperature rise. Although this method is commonly accepted, the uncertainty of the thermocouple reading of the mainstream temperature can lead to overall uncertainty in the experiment of around 12%. In an effort to reduce this uncertainty, the mesh heater was designed to provide a true step change in air temperature at the start of a test. Due to the extremely fast response of the heater, the Duhamel’s superposition method previously used to account for slow heater responses was not needed for this rig. The true step change in mainstream temperature allows the use of the original solution to the transient heating of the wall, Equation 10. This decreases uncertainty by eliminating the reliance on recorded thermal response of the heaters.

The second technique used to reduce uncertainty in experimental measurements was to collect multiple wall temperature-time data pairs over a broad range of temperatures to use in an overconstrained system of equations to solve for $h$. The liquid crystal chosen had a 20°C band of color play. A thin type –k thermocouple glued to the target plate recorded point temperatures during each test and was used in the “In Situ” calibration of the liquid crystals. The temperature data from the surface mounted thermocouple was used to generate a hue-temperature calibration of the liquid crystal paint for every test. This method reduces the possibility of calibration errors caused from inconsistent lighting conditions or aging of the liquid crystal paint.

Since the liquid crystal paint chosen was able to give wall temperatures over a wide range of temperatures, a regression analysis method was used to eliminate random errors in recorded local wall temperatures. The regression analysis put all terms of the conduction equation to the right hand side of the equation and was solved for all points of data for each pixel. This resulted in a residual error for each time-temperature data pair. The residual error was minimized in a least squares sense solving for the heat transfer coefficient that best fit all data. This reduced the random camera read errors of the liquid crystal paint and also relaxes the dependence on the initial temperature of the target plate.
on the results. Figure 17 shows a typical time-temperature response of a single pixel and the best curve fit solution of the data.

![Time-Temperature History of Single Pixel](image)

Figure 17: Curve Fit of Raw Data from a Single Pixel

### 3.5 Uncertainty Analysis

The distribution of uncertainty in the recorded measurements is given in Table 2.

<table>
<thead>
<tr>
<th>Input</th>
<th>Part I (1 Data Point)</th>
<th>Part II (20 Data Points)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Value</td>
<td>Uncertainty</td>
</tr>
<tr>
<td>$T_i$</td>
<td>20.5 C</td>
<td>±0.6 C</td>
</tr>
<tr>
<td>$T_m$</td>
<td>40 C</td>
<td>±0.6 C</td>
</tr>
<tr>
<td>$T_w$</td>
<td>31.4 C</td>
<td>±0.6 C</td>
</tr>
<tr>
<td>time</td>
<td>18 s</td>
<td>250 ms</td>
</tr>
</tbody>
</table>

| Overall Error | 17.0% | Overall Error | 8.5% |
A simple algorithm that allowed for the systematic varying of experimental uncertainties in recorded data was used to estimate the overall uncertainty of the results for both parts of the study. This was calculated for an average heat transfer coefficient value for the experiments and for the worse case scenario for the accumulation of error. The largest error in the experiment was caused from the mainstream temperature and wall temperature liquid crystal readings. Part I of the study used Duhamel’s superposition theorem to determine the heat transfer coefficient based on incremental step changes in mainstream temperature. In this case, the errors from thermocouple readings were compiled in each incremental step and lead to higher uncertainty. The use of the mesh heater in Part II provided a true step change in temperature thus limiting this thermocouple error to one incident and therefore reducing the overall error of the experiment. Part I, using the single temperature method, showed that the uncertainty in an average $h$ value of 500 W/(m$^2$ K) was about 17%. When the regression analysis technique was applied to Part II for an average $h$ value of 1000 W/(m$^2$ K), the overall uncertainty dropped to about 8.5%.
CHAPTER 4: RESULTS

The results obtained from the baseline geometry were used as a reference for all other configurations. In order to validate the test rig, the baseline geometry results were plotted against a commonly accepted impingement correlation put forth by Florschuetz.

4.1 Part I Results for Sparse Arrays

Experiments for the dense arrays were performed at Reynolds numbers of 20000, 30000, 40000, 50000 and 60000. Figure 18 is an area averaged plot of the baseline Nusselt number plotted with the Florschuetz correlation.

Figure 18: Area Averaged Baseline Data Plotted with Florschuetz
The plot shows that the overall levels of Nusselt numbers are comparable with the correlation although the effects of crossflow are somewhat greater in the experimental data. This is thought to be attributed to the given baseline geometry and Reynolds numbers tested. The parameters of the small $z/d$ spacing and relatively high Reynolds numbers lie on the outer limits of the acceptable ranges for the correlation.

4.1.1 Spanwise Averaged Nusselt Numbers

The spanwise averaged Nusselt number plots help to show the detrimental effects of crossflow on downstream jets. As the crossflow hits the jets and attempts to wash them downstream, the stagnation region moves further downstream and the jet no longer is symmetrical. Typically the crossflow will cause a sharp rise in Nusselt number to the stagnation point, and then gradually fade away in the downstream direction. Figures 19-22 show the spanwise averages for each configuration.

![Figure 19: Baseline Spanwise Averaged Nusselt Number](image-url)
Figure 20: Corrugated Wall (1) Spanwise Averaged Nusselt Number

Figure 21: Corrugated Wall (2) Spanwise Averaged Nusselt Number
4.1.2 Area Averaged Nusselt Numbers

Area averaged Nusselt number data from each test case was plotted together for comparison of the tested configurations. The corrugated wall geometries were compared to one another to show the effect of differing corrugated bypass channel widths. Figure 23 is a plot of both corrugated wall configurations tested for the range of Reynolds numbers.
In the following plots, the corrugated wall designs, extended ports, and the baseline configurations were plotted together for comparison at each Reynolds number tested. Figures 24-28 show the combined area averaged plots.

Figure 24: Area Averaged Plot for 20,000 Reynolds Number

Figure 25: Area Averaged Plot for 30,000 Reynolds Number
Figure 26: Area Averaged Plot for 40,000 Reynolds Number

Figure 27: Area Averaged Plot for 50,000 Reynolds Number
4.2 Discussion of Part I Results

Both the corrugated wall designs and the extended port configurations proved to be beneficial in reducing the adverse affects of crossflow, although sometimes at the expense of overall Nusselt number averages in the beginning jets. All of the configurations tested did exhibit a similar behavior where the Nusselt number averages peaked not at the first jet, but several jets down the array where the crossflow starts to change the jet structure. Originally this was thought to be from a misdistribution of air temperatures in the plenum. Fitting the plenum with numerous thermocouples proved that the air was actually uniform within about ± 0.4 °C. After completing all experiments and plotting the results, a different explanation for this condition arose. The limited amount of crossflow in the jet numbers 3-5 seemed to have the effect of confining part of the leading edge of the jet and actually increasing the stagnation area where the heat transfer is greatest. This confining of the jets and leading to increased stagnation regions is likely due to the geometrical parameters of the arrays. The small z/d spacing results in high jet to crossflow velocity ratios allowing the jets to combat the detrimental effects of crossflow in the beginning of the array. As the crossflow velocity increases towards the
downstream jets, the jets are overcome by the increase in crossflow velocity and the heat transfer characteristics suffer.

The corrugated wall designs performed quite differently. The larger bypass configuration, corrugated wall (2), showed lower overall heat transfer performance when compared to the corrugated wall (1) geometry, particularly at higher Reynolds numbers. This is possibly due to the nature of the large low pressure bypass channels. This low pressure region may be causing the jets to prematurely expand and therefore lower the jet’s peak velocity prior to striking the target plate. Since the overall performance of the jet is largely due to the high heat transfer coefficients found near the stagnation region, this slowing of the jets would significantly lower the effectiveness of the core jet region. The larger jet exit surface of the corrugated wall (1) geometry seemed to be a better balance between bypass channel sizes without adversely changing the jet structure.

Unlike the corrugated wall designs which seemed more effective at higher Reynolds numbers, the extended port configuration actually proved more beneficial at lower Reynolds numbers. In all cases, the extended port configuration showed almost symmetrical jet profiles throughout the array. As the Reynolds numbers increased, so too did the mean jet velocity through the extended ports. At lower velocities, the exiting velocity profile was further developed than at higher velocities since the length of the ports is constant. Again, the partial confining of the jets which was leading to larger stagnation regions appears in the extended port results. However, due to the much greater cross-sectional area of the geometry and therefore lower crossflow velocities, the peak jets in the array moved slightly further downstream, proving the jet confinement theory.

4.3 Part II Results: Dense Arrays

Experiments for the dense arrays were performed at Reynolds numbers of 20,000, 40,000 and 60,000.

4.3.1 Local Nusselt Number

The local Nusselt number distributions for the all geometries are presented in Figures 29-31. All were plotted on the same scale for each Reynolds number.
Figure 29: 20k Local Nusselt Number
Figure 30: 40k Local Nusselt Number
Figure 31: 60k Local Nusselt Number

Baseline 60k

Corrugated 60k

Extended Port 60k

Variable Extended 60k

100 120 140 160 180 200 220 240 260 280
4.3.2 Spanwise Averaged Nusselt Number

Figures 32-34 show the spanwise averaged Nusselt number plots for each Reynolds number.

Figure 32: 20k Spanwise Averaged Nusselt Number

Figure 33: 40k Spanwise Averaged Nusselt Number
4.3.3 Area Averaged Nusselt Number

Figures 35-37 are plots of the area averaged Nusselt numbers.
Figure 36: 40k Area Averaged Nusselt Number

Figure 37: 60k Area Averaged Nusselt Number
4.4 Discussion of Dense Array Results

As with the sparse arrays, the results for the dense arrays showed improved crossflow tolerance throughout the array. The corrugated wall configuration showed little overall effects from crossflow at high Reynolds numbers. Both the extended port and variable extended port configurations aided in reducing the effects of crossflow, but again also elevated overall heat transfer levels at lower Reynolds numbers. As previously stated, this is thought to be from the lower jet velocities in the extended ports allowing the flow to develop further than the higher velocity cases prior to exiting the tubes.

The peak in heat transfer not at the first jet, but several jets downstream is also noted in the dense array results. It was thought that although crossflow was generally thought to be only detrimental to the jet structure, in small amounts and in low $z/d$ spacings, may actually help to confine the jets and increase the effective size of the stagnation region.
In the presented work, variations of a corrugated wall and extended port configurations were studied in an effort to find improved jet impingement geometries that were less susceptible to the effects of crossflow. The configurations were tested for sparse arrays with jet spacings of 11, and dense arrays with spacings of 5. In all cases the detrimental effects of crossflow on the downstream jets was reduced. The designs did show several noteworthy points. The corrugated wall configuration was tested for two sizes of bypass channels. The larger channel size, although predicted to be more beneficial, proved less effective than the smaller bypass channel geometry. This is thought to be due to the large low pressure region in the bypass channel causing the jet to prematurely expand and the small area of the impingement plate at the jet exit becoming unable to hold the jet near the target surface. This change in jet structure caused by the larger bypass area reduced the stagnation region which dominates the overall heat transfer of the jet.

The extended port configurations allowed the largest cross-sectional area of all geometries tested. This aided in reducing the average crossflow velocity throughout the test section. Although the larger bypass channels proved less effective in the corrugated sections, the extended port design utilizes tubes that allow for the flow to develop prior to exiting. The results showed that for low Reynolds numbers the extended port configuration not only reduced crossflow effects, but also increased the overall average Nusselt numbers of the array. This increase in overall effectiveness at higher Reynolds numbers may be possible if the extended ports were lengthened proportional to the mean jet velocity. This would allow for more development of the flow at higher Reynolds numbers and may lead to the increase in overall heat transfer gained at the lower flowrates.
5.1 Future Work

Based on the findings of the present study, the corrugated wall geometry proved to be more effective at higher Reynolds numbers and the extended port design more efficient at lower Reynolds numbers. This effect may be altered with varying jet Mach numbers. Little research has been conducted in the area of jet Mach number effect on jet structures and heat transfer. Modern gas turbine engine combustor impingement configurations operate at relatively low Mach numbers in the range of 0.1-0.3. Higher jet Mach numbers could prove a substantial increase in pressure drop, but may also yield heat transfer enhancement.
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VITA

Eric Ian Esposito was born in Baton Rouge, Louisiana on September 7, 1978. He attended E. D. White Catholic High School in Thibodaux, Louisiana and graduated in May of 1996. Eric then studied at Louisiana State University where he earned his Bachelor of Science degree in Mechanical Engineering in May of 2004. He continued his education at Louisiana State University pursuing a Masters of Science degree in Mechanical Engineering under the guidance of Dr. Srinath Ekkad. Having fulfilled the requirements set forth by the Graduate School and the Department of Mechanical Engineering, Eric will graduate in August of 2006.