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Heat Transfer In A Coupled Impingement-effusion Cooling System

Mark W. Miller
University of Central Florida

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HEAT TRANSFER IN A COUPLED IMPINGEMENT-EFFUSION COOLING SYSTEM

by

MARK W. MILLER
B.S.M.E The Ohio State University, 2009

A thesis submitted in partial fulfillment of the requirements
for the degree of Master of Science
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Major Professor: Jay Kapat

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ABSTRACT

Gas turbine engines are prevalent in the today's aviation and power generation industries. The majority of commercial aircraft use a turbofan gas turbine engines. Gas turbines used for power generation can achieve thermodynamic efficiencies as high as 60% when coupled with a steam turbine as part of a combined cycle. The success of gas turbines is a direct result of a half century's development of the technology necessary to create such efficient, powerful, and reliable machines. One key area of technical advancement is the turbine cooling system.

In short, increasing the turbine inlet temperature leads to a rise in cycle efficiency. Before the development of modern turbine cooling schemes, this temperature was limited by the softening temperature of the metallic turbine components. The evolution of component cooling systems – in conjunction with metallurgical advancements and the introduction of Thermal Barrier Coatings (TBC) – allowed for gradual increases in power output and efficiency. Today, the walls of gas turbine combustors are protected by a cool film that bypassed combustion; the 1st (and often 2nd) stage turbine blades and vanes are cooled via internal convection, a combination of turbulent channel flow, pin fin arrays, and impingement cooling; and some coolant air is bled onto the external surface of the blade and the blade endwall to establish a protective film on the exposed geometry.

Modern research continues to focus on the optimization of these cooling designs, and a better understanding of the physics behind fluid behavior. The current study focuses on one particular cooling design: an impingement-effusion cooling system. While a single entity, the cooling schemes used in this system can be separated into impingement cooling on the backside

of the cooled component and full coverage film cooling on the exposed surface. The result of this combination is a very high level of cooling effectiveness.

The goal of this study is to explore a wide range of geometrical parameters and their effect on the overall cooling performance. Several parameters are taken outside the ranges normally investigated by the available literature. New methods of data comparison and normalization are offered in order to create an objective comparison of different configurations. Particular attention is given to the total coolant spent per unit surface area cooled.

This study is also unique as it is a multi-modal heat transfer study, unlike the majority of impingement-effusion investigations, which only evaluate impingement heat transfer. Through determination of impingement heat transfer, film cooling effectiveness, and film cooling heat transfer on the target wall, a simplified heat transfer model of the cooled component is developed to show the relative impact of each parameter on the overall cooling effectiveness.

The use of Temperature Sensitive Paint (TSP) for data acquisition allows for high resolution local heat transfer and effectiveness results. This has a quantitative benefit, giving the ability to average as desired and/or compare local data, for example the lateral distribution of film cooling effectiveness. However, the qualitative benefit of viewing the contours of heat transfer coefficient under an impinging jet array or downstream of a film cooling jet is instrumental in drawing conclusions about the behavior of the flow. The local data provides, in essence, a flow visualization on the test surface and adds (quite literally) another dimension to the heat transfer results.

Impingement arrays with local extraction of coolant via effusion are able to produce higher overall heat transfer, as no significant cross flow is present to deflect the impinging jets. Low jet-to-target-plate spacing produces the highest yet most non-uniform heat transfer

distribution; at high spacing the heat transfer rate is much less sensitive to impingement height. Arrays with high hole-to-hole spacing and high jet Reynold's number are more effective (per mass of coolant used) than tightly spaced holes at low jet Reynold's number.

On the effusion side, staggered hole arrangements provide significantly higher film cooling effectiveness than their in-line counterparts as the staggered arrangement minimizes jet interactions and promotes a more even lateral distribution of coolant. These full coverage film cooling geometries typically show increases in effectiveness with each row of injection. Some additional cases were shown with 15 film cooling rows, and generally the adiabatic wall temperature was decreasing through the last row. In the recovery region, results were highly dependant on blowing ratio; injection of excess coolant into the boundary layer at high blowing ratio allowed for cooling effectiveness to penetrate well downstream of the end of the array. From a heat transfer standpoint, compound angle injection resulted in higher enhancement than purely inclined injection, but this negative effect was outweighed by the substantial increase in film cooling effectiveness with the compounded geometry. Overall, the additive film superposition method under-predicted full coverage film cooling effectiveness trends for staggered hole arrangements; however, with more accurate estimation (or measurement) of recovery region trends for a single row of holes, this method may produce an acceptable result.

DEDICATED TO UBU
I'll miss you, buddy

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NOMENCLATURE

A – area [m^2]

α – film cooling hole inclination angle [$^\circ$]

β – film cooling hole compound angle [$^\circ$]

C – constant

C_d – discharge coefficient [-]

D – jet hole diameter [m]

DR – density ratio [-]

G – mass flux [$\text{kg}/\text{m}^2\text{s}$]

ζ – axial coordinate x normalized by blowing ratio (M) and hole pitch (X_e) [-]

η – film cooling effectiveness [-]

h – heat transfer coefficient [$\text{W}/\text{m}^2\text{K}$]

$h(\theta)$ – heat transfer coefficient based on mainstream to wall temperature difference [$\text{W}/\text{m}^2\text{K}$]

Θ – net heat flux reduction [-]

θ – dimensionless wall temperature [-]

I – momentum flux ratio [-]

I/I_{ref} – intensity ratio, with respect to TSP images

k – thermal conductivity of wall [W/mK]

k_f – thermal conductivity of fluid [W/mK]

L – length of film cooling hole [m]

M – blowing ratio or mass flux ratio [-]

$\dot{m}$ – mass flow rate [kg/s]

μ – dynamic viscosity [Ns/m²]

Nu_D – Nusselt number based on jet diameter [-]

n – row number

ν – dynamic viscosity [m²/s]

ξ – unheated starting length [m]

p – pressure [N/m²]

PR – picture ratio

q – heat flow rate [W]

q'' – heat flux [W/m²]

r – radial coordinate

Re_D – Reynolds number based on jet diameter [-]

Re^* - modified Reynolds number, Re_D divided by surface area cooled by one jet [-]

R_{η}'' – heat transfer resistance (per unit area) associated with film cooling effectiveness [m²K/W]

s – entropy [J/K]

T – temperature [K]

t – thickness of wall [m]

U – velocity [m/s]

x – streamwise coordinate

X – streamwise hole spacing [m]

y – spanwise coordinate

Y – spanwise hole spacing [m]

z – coordinate normal to surface

Z – impingement height [m]

Subscripts

aw – adiabatic wall

c – coolant

e – effusion

f – with film

i – impingement

m – mainstream

0 – without film

r – single row

w – wall

CHAPTER 1. INTRODUCTION

Section 1.1 Gas Turbine Cooling

Air-breathing gas turbine engines supply both propulsion for the aerospace industry and highly efficient, low-emissions solutions for the power generation industry. Gas turbines operate on a thermodynamic Brayton cycle, in which a high pressure, high temperature mixture of air and fuel is driven through multiple stages of blades (rotors) and vanes (stators) to generate useful power output. In Figure 1, the basic components of a Brayton cycle are identified, and the temperatures associated with each thermodynamic state are indicated on a T-s diagram.

Figure 1 Thermodynamically-Ideal Brayton Cycle

The thermodynamic efficiency of the ideal Brayton cycle is limited by the compressor pressure ratio (P_2/P_1). Consequently, increases in P_2/P_1 and, accordingly, T_2 and T_3 must occur in order to minimize fuel usage and maximize efficiency. Early gas turbines were, in fact, limited by T_3 as turbine inlet temperatures approached the material melting limit at the 1st stage vane. However, it was quickly discovered that relatively low temperature air from the compressor

could be driven through radially-drilled holes in the otherwise solid blade, so that the blade would act as a heat sink (Landis, 2009). As the rate of heat transfer into the cooling air increased, the temperature difference over the external convective resistance increased, and therefore higher temperatures could be maintained in the hot gas path while blade metal temperatures remained at the acceptable limits.

Figure 2 Historical Trend of Turbine Inlet Temperature Related to Cooling Scheme (Clifford, 1985)

Since the introduction of component cooling, modern gas turbine inlet temperatures have increased at a rate of 11°C (19°F) per year, over the past 50 years. This can be compared with material advancements, which account for an increase of 4°C (7°F) per year over the same period (Landis, 2009). To achieve such an increase, new, complex cooling methods have been developed to cool the 1st and 2nd stage blades and vanes, as depicted by Figure 2. An insert is placed within the hollow blade from which coolant jets impinge onto the backside of the surface to be cooled; internal channels with turbulence- and swirl-inducing mechanisms as well as heat transfer-enabling fins route the coolant within the blade; and the coolant, still at a considerably lower temperature than the mainstream gas, is ejected either at the blade tip or through effusion

holes that are designed to generate a protective film on the exposed blade surface. Modern-day turbines also utilize thermal barrier coating (TBC) which includes a thermally insulative ceramic layer to further reduce metal temperature.

Figure 3 Gross Cooling Efficiency of Various Blade Cooling Schemes (Bunker, 2005)

The efficiency of such blade cooling schemes becomes progressively higher as the mass flow rate of coolant increases, as indicated by Figure 3. In this figure, the Heat Loading Parameter (HLP) is defined as the ratio of the thermal capacity of the coolant to the total amount of energy that must be driven into the blade. Higher coolant flow rates therefore allow for higher combustion temperatures, but excessive compressor work required to supply the coolant may lead to an overall reduction in cycle thermodynamic efficiency. In this regard, the design of cooling systems involves a careful balance of cooling efficiency and aerodynamic losses.

Section 1.2 Impingement-Effusion Cooling Schemes

Coupled impingement-effusion cooling systems are prevalent throughout both the combustor and high pressure turbine stages. Impingement cooling on the backside of the blade wall provides high local heat transfer coefficients and utilizes a simple design that can be

administered in confined spaces. It is desirable to purge the spent coolant from the impingement channel locally via effusion, rather than via a channel flow that impact neighboring jets. This purged coolant can be used to create an effective film and protect the exposed surface of the part. Such an arrangement is typical for showerhead cooling at the leading edge of the airfoil, as shown in Figure 4.

Figure 4 Showerhead Cooling System Arrangement (Liang, 2006)

© Siemens

Figure 5 Modern Turbine Vane (left) and Blade (right) with Full Coverage Film Cooling (Jessen, 2008)

Some 1st stage blades and vanes require considerable quantities of film on the pressure side and endwall of the airfoil to maintain a high heat load. Multi-row film cooling arrangements

are used, as shown in Figure 5, often in conjunction with backside impingement. Such effusion configurations are dubbed Full Coverage Film Cooling, as the basic goal is to uniformly cool a large surface via several rows of small coolant holes (rather than one row with larger holes).

The only component in a gas turbine that sees higher hot gas temperatures than the 1st stage vane is the combustor itself. Combustor design has been driven by high performance, high efficiency, and reduced emissions. New combustor designs must reduce the amount of bypass air that acts as combustor liner coolant to meet these demands (Schulz, 2006). More efficient use of coolant is achieved using hybrid cooling systems (see Figure 6) such as impingement-effusion cooling. Schulz (2006) notes that, in fact, current research and development efforts in combustor cooling focus on full coverage film cooling with backside impingement.

Figure 6 Various Combustor Liner Cooling Methods (Cerri, 2007)

Section 1.3 Objectives of the Current Work

The purpose of this study is to investigate the heat transfer associated with several different impingement-effusion cooling system geometries. The advantage of doing so is two-fold: first, it adds to the current body of knowledge available in the open literature, which is lacking in certain areas; and second, a wide-ranging study with variation of many design

parameters allows objective comparison and promotes optimization of impingement-effusion cooling system design.

In this study, several geometrical parameters are pushed beyond their previously-documented limits. For example, there exists very little literature studying impingement heat transfer at high Z/D , much less array impingement heat transfer at such heights. As cooling system designers work within global constraints, it may be necessary to use such a “tall” impingement geometry in practice where moderate cooling is needed. In this case, the performance of such arrangements should be known with respect to more commonplace configurations. Also, the use of unshaped film cooling holes angled normal to the mainstream flow is not ideal from a standpoint of coolant attachment, but there are many cases where manufacturing considerations may lead to the use of such a design.

This work is aimed not only to be a study in variation of parameters, but also an exercise in practical cooling system design. Average impingement Nusselt numbers are compared versus the mass of coolant spent per unit surface area cooled; full coverage film cooling results are predicted via correlation and superposition; film cooling effectiveness and heat transfer data is collected for large full coverage arrays with angled injection to allow comparison with normal injection; the total heat transfer rate and corresponding wall temperature of each impingement-effusion cooling system is calculated to promote a sense of scale for each involved heat transfer method. Overall, the aim of this work is to characterize the heat transfer performance of a cooling system design from an objective and practical standpoint.

One additional feature exploited by this work is the ability to have local resolution of heat transfer and effectiveness data via high resolution measurements using Temperature Sensitive Paint (TSP). Data can be presented locally for qualitative investigation, and then averaged in as

necessary for ease in quantitative evaluation. Impingement and film cooling heat transfer profiles are highly non-uniform, and local resolution allows visualization of interactions between jets so as to help understand flow phenomena. Furthermore, such non-uniformities in heat transfer distributions should be known to the designer, who must consider thermal stresses and other adverse effects of these varying thermal boundary conditions.

CHAPTER 2. BACKGROUND

Section 2.1 Impingement Cooling

Figure 7 Impinging Jets from Insert in Gas Turbine Blade (Florschuetz, 1981)

Fluid jets impinging on a target surface yield high local convective heat transfer coefficients which are desirable when high cooling rates are needed, such as a first stage gas turbine blade or vane. The impinging fluid and surrounding fluid may either be in the same state (submerged impingement) or different states (unsubmerged impingement). The latter of these, for example a water jet striking a hot plate in the ambient air, would likely not be applicable to an impingement-effusion cooling system and therefore only submerged impingement will be discussed in this section.

There exists a great deal of literature relating to the structure of the impinging jet and the resulting heat transfer profile on the target surface. Laboratory experiments often involve unconfined jets where no top wall or exiting channel is provided for the coolant. However, as practical applications often require arrays of nozzles and a compact design, the effect of jet confinement has been studied. Predominately, confined impingement studies involve an array of jets in a channel so that the effect of a built-up cross flow on downstream impingement heat

transfer may be observed. A schematic of unconfined and confined impingement is provided as Figure 8.

Figure 8 Unconfined vs. Confined Impinging Jets

Section 2.1.1 Nomenclature

Figure 9 Impingement Array Geometry

In Figure 9 above, the coordinate systems and geometrical definitions for an impinging jet within an array are shown. The jet exit is spaced at a height Z from the target surface; the mean velocity at which the jet exits is U . The holes are spaced in the span direction by Y and in the stream direction by X . For an individual jet, the heat transfer profile is expected to be uniform about the circumference, so the radial component is often used to plot the evolution of heat transfer away from the jet stagnation point.

The velocity impinging jet described via the jet Reynolds number, based nozzle diameter, as shown in (1). ν is the kinematic viscosity of the fluid.

$$\text{Re}_D = \frac{UD}{\nu} \quad (1)$$

Heat transfer performance is also described via a non-dimensional parameter known as the Nusselt number, shown in (2). The heat transfer coefficient, h , and the thermal conductivity of the fluid, k_f , are used in the calculation of Nu_D .

$$\text{Nu}_D = \frac{hD}{k_f} \quad (2)$$

Section 2.1.2 Unconfined Impingement

Martin (1977) provides an extensive review of the early work done on impinging jets. Specifically, correlations are presented for four impingement configurations: single round nozzles (SRN), single slot nozzles (SSN), arrays of round nozzles (ARN), and arrays of slot nozzles (ASN). Shown in Figure 10, three regions of the unconfined SRN are established: the free jet, the stagnation region, and the wall jet. Due to viscous interaction with the ambient fluid, the free jet expands linearly with distance from the nozzle exit. In this region, the fluid momentum is unidirectional throughout. In the stagnation region the streamlines bend and the fluid is turned 90° so that pure radial flow exists at the exit. The fluid momentum continues to decrease in the wall jet as the fluid spreads radially and the boundary layer thickens as stagnant air above the wall jet is pulled into the flow.

Figure 10 Schematic of Unconfined SRN Impingement from (Martin, 1977)

The highest heat transfer rates occur in the stagnation region, with radially decreasing heat transfer coefficients as the coolant disperses. This non-uniform heat transfer profile was studied with the intention to develop a correlation for local impingement Nusselt number using several extensive sets of local heat and mass transfer data. The resulting SRN correlation is presented in (3) and (4).

$$\overline{Nu_D} = \frac{1 - 1.1/(r/D)}{(r/D) + 0.1(Z/D - 6)} F(Re_D) Pr^{0.42} \quad (3)$$

$$F(Re_D) = \begin{cases} 1.36 Re_D^{0.574} & 2000 < Re_D < 30000 \\ 0.54 Re_D^{0.667} & 30000 < Re_D < 120000 \\ 0.151 Re_D^{0.778} & 120000 < Re_D < 400000 \end{cases} \quad (4)$$

This correlation is limited to $r/D > 2.5$ and $2 \leq Z/D \leq 12$. Local data shows a wide variety of heat transfer coefficient distributions in the stagnation region—monotonically decreasing at large Z/D and exhibiting a secondary maximum at low Z/D – and therefore this region was not included in the correlation. The form of the correlation suggests that Nusselt number decreases with increasing nozzle-to-plate spacing Z/D . Also noteworthy is that the rate of Nu_D increase with Re_D diminishes as Re_D increases. Similar trends are presented in correlations for SSN,

ARN, and ASN configurations. Slot nozzles, while an interesting analytical problem, are not commonly used in modern cooling schemes and will not be discussed in this review. The ARN configuration and derived correlation are relevant to this study, however, and Martin's work on ARN configurations will be briefly summarized.

The flow field of an ARN configuration is comprised of individual jet structures much like that in Figure 10, with the exception that a secondary stagnation region forms upon the collision of neighboring wall jets. The heat transfer distribution under each jet in an ARN configuration is essentially identical to the SRN result so long as the nozzle-to-plate spacing Z/D does not exceed a certain limit that is a function of the array geometry. Above this limit, the neighboring jets interact before reaching the target surface and subsequently decrease heat transfer, as compared to the performance of the SRN. The ARN correlation (Martin, 1977) provides an area-averaged value of Nu_D only; this correlation is presented in (5), (6), and (7).

$$\overline{Nu_D} = K(Z/D, f) \frac{1 - 2.2f^{0.5}}{f^{-0.5} + 0.2(Z/D - 6)} Re_D^{2/3} Pr^{0.42} \quad (5)$$

$$K(Z/D, f) = \left[1 + \left(\frac{Z/D}{0.6f^{-0.5}} \right)^6 \right]^{-0.05} \quad (6)$$

$$f = \frac{\pi}{4} \left(\frac{X}{D} \right)^{-2} \quad (7)$$

In comparing (3) and (5) it can be seen that the behavior of Nu_D with Re_D and Pr is essentially identical for SRN and ARN configurations. This suggests the integrity of the individual jets is maintained in the data used to develop the correlation, and effects of jet-to-jet interaction are minimized. Both are monotonically decreasing functions of impingement height Z/D , but (5) suggests an optimal value of hole-to-hole spacing X/D at a given Z/D . As Z/D increases, the optimal value of X/D becomes greater. Note that the data used to generate the

ARN correlation included confined and unconfined results where the flow may exit in different directions, so no factor was included to evaluate the effect of exiting flow on downstream jets.

A more recent and equally thorough review of impingement heat transfer was published by Viskanta (1993). The physics of impinging flow are thoroughly elaborated and the effects on the resulting heat transfer profile are discussed. Viskanta first notes that non-uniformity of the heat transfer distribution is a primary disadvantage of impingement cooling systems, and therefore many researchers have investigated a wide variety of geometries (circular and slot jets, oblique impingement, impingement on curved surfaces) to determine an optimal configuration. Nevertheless, the fundamental unconfined SRN jet impinging on a flat surface has been the primary focus of the impingement heat transfer community.

Of particular significance in heat transfer performance is the impinging jet's potential core. The potential core is defined as the fluid at the center of an impinging jet whose momentum has not been decreased by interaction with the surrounding fluid. Figure 11 shows clearly the axial variation in a jet's centerline velocity; in this case (Hoogendoorn, 1977), the potential core is shown to last approximately 5 diameters downstream of the nozzle exit. Most researchers identify the axisymmetric impinging jet potential core length to be between 6 and 7 diameters (Livingood, 1973).

Also of importance in Figure 11 is the development of the turbulence intensity profile. Turbulence levels increase significantly in the mixing region between the potential core and stagnant air. As the jet continues to entrain ambient fluid, its momentum decreases while turbulence levels increase. This creates a maximum stagnation point impingement Nusselt number at an intermediate (rather than low) Z/D . Typically, this optimal jet standoff is reported

to be equal to or slightly greater than the length of the potential core, or Z/D between 5 and 8 for SRN impingement.

Figure 11 Velocity and Turbulence Intensity Profiles for a Free Jet (Hoogerdoorn, 1977)

SRN impingement at low Z/D has also been the focus of intense study, primarily due to the unique appearance of the resulting heat transfer profile: a secondary heat transfer maxima occurs 1 to 2 diameters from the stagnation point. This maxima is especially pronounced at very low Z/D and high Re_D , as shown in the plots taken from Lytle (1991) in Figure 12.

Lytle correlated the location of this peak with respect to both Z/D and Re_D . He also provided a radial distribution of turbulence intensity that clearly shows a prompt rise in turbulence at the location of the second peak. This high level of turbulence is maintained beyond the second peak, suggesting that the wall jet boundary layer transitions from laminar to turbulent at this point, and therefore creates the secondary maxima in heat transfer.

Figure 12 Radial Distribution of Nusselt Number for $Re = 23,000$ (left), $Z/D = 0.2$ (right) (Lytle, 1991)

Section 2.1.3 Confined Impingement

Several researchers (Goldstein, 1986 for example) chose to investigate SRN impingement using a confined jet. Often a nozzle was used to make the jet velocity profile flat, rather than parabolic. The effect of such different inlet conditions were studied by Obot (1982) who was able to show a significant decrease in heat transfer for the confined jet at $Z/D < 2$. This was attributed to the jet confinement restricting the flow of entrained ambient air and therefore reducing turbulent mixing of the jet. The region of reduced heat transfer was therefore primarily confined to $r/D < 2$, where reductions of 30 to 40% were common at $Z/D = 2$. At higher Z/D , these reductions were significantly reduced.

Figure 13 Typical Impingement Array Geometry (Florschuetz, 1981)

Most practical impingement applications require an array of impinging jets (for the sake of brevity, ARN) to evenly distribute cooling over a large target surface. In a laboratory experiment, it is practical to construct an ARN using a perforated plate attached to a constant pressure plenum rather than an array of individual pipes. Therefore, ARN are, by their nature, confined. A typical ARN geometry, from Florschuetz (1981), is shown in Figure 13.

The extraction of spent coolant from an ARN may be accomplished through several methods: cross flow depletion, target plate effusion depletion, or jet plate effusion depletion. The first of these schemes (as shown in Figure 13) allows for a simple design, as air freely exits to a sink at one side of the array. The latter two require a sink to be placed either above or below the parallel plates which confine the array. Discrete holes or porous blockages are used to locally extract spent coolant near each jet's stagnation point. Therefore, impinging jets at one end of the array are not deflected by the crossflow of exiting air, allowing for high, uniform heat transfer throughout the array.

Florschuetz (1981) measured area-averaged Nu distributions for a wide array of streamwise (X/D) and spanwise (Y/D) hole spacings. The effect of Z/D , Re_D and jet-to-crossflow momentum flux ratio (G_c/G_j) were also studied. A correlation was developed that encapsulated a wide range of each of these parameters; the form of this correlation is shown in (8). (The coefficients A , B , m , and n are functions of geometrical parameters X/D , Y/D , and Z/D .) Florschuetz developed this correlation using a wide range of experimental data for X/D between 5 and 15, Y/D between 4 and 8, and Z/D between 1 and 3. He showed that increasing G_c/G_j decreased Nu at a given spanwise row and crossflow depletion subsequently inhibits heat transfer in an ARN.

$$\overline{Nu} = A Re_j^m Pr^{1/2} \left[1 - B \left(\frac{X}{D} \frac{G_c}{G_j} \right)^n \right] \quad (8)$$

An earlier study by Kercher (1970) includes the development of a correlation based on confined array impingement geometries to that shown by Figure 13. The correlation has a similar form as Florschuetz's, The correlation constants are determined graphically (as functions of the hole-to-hole spacing X/D) and separate curves are presented for low Re_D (less than 3000) and high Re_D . This confirms that turbulence at the jet exit has an effect on impingement heat transfer performance. The majority of the examined data fell within 10% of the correlation prediction.

While several other studies exist that expand on the geometrical variations studied by Kercher and Florschuetz, there are relatively few investigations into alternative coolant extraction schemes. Jet plate depletion scenarios have been recently investigated by Rhee (1996) and Onstad (2005) to address the consideration that it may be undesirable or impossible to force spent coolant onto the opposite side of the surface being cooled. The impingement and effusion arrays are staggered in both directions to encourage a symmetrical dispersion of coolant from the stagnation point. This research concludes that local extraction of coolant leads to an identical heat transfer distribution under each impinging jet. Furthermore, Rhee shows a significant increase in span averaged Nu over an identical crossflow depleted array at low Z/D .

Jet plate depletion schemes require a sophisticated design, however, to route incoming and outgoing coolant above the jet plate. A common application for ARN geometries is the gas turbine blade or vane, where impinging coolant may be removed through the target wall and used as a cooling film on the exposed surface of the component. Therefore, the use of target plate depletion for such an application provides a three-part benefit: impingement heat transfer on the back side of the wall is increased; the coolant picks up heat as it travels through the effusion

holes; and the coolant film reduces the adiabatic wall temperature on the hottest surface of the component.

Hollworth (1981) developed an extensive test matrix of X/D , Y/D , and Z/D for an ARN geometry with target plate effusion depletion. Hollworth's depiction of this "hybrid cooling system" is displayed in Figure 14. The effusion holes were initially oriented in both in-line and staggered patterns with respect to the impingement array; however, it was quickly shown that the in-line holes capture too much of the impinging coolant, as they were placed at the jet stagnation points, and the staggered arrangement provides a clear heat transfer advantage. Peak heat transfer rates were reported at an intermediate Z/D of 3 to 5. Also, considering the amount of coolant spent per unit area of target surface cooled, large X/D and Y/D spacing was determined to be more efficient than smaller spacing, though the low spacing provided a greater magnitude of heat transfer.

Figure 14 "Hybrid Cooling System", Side View (left) and Top View (right)

In Part 2 of the same study (Hollworth, 1983), local values of Nu at discrete points were provided. Because Hollworth employed an arrangement where the spacing between impingement holes was twice the spacing between effusion holes, there were regions between the rows of impinging jets that remained virtually uncooled. All other studies of impingement-effusion

geometries have altered this arrangement so that the impingement and effusion spacings are equal.

Cho and Rhee (1995) provide local heat transfer results for a variety of Z/D and Re and compare the results to a numerical simulation. This study shows that, for a low Z/D of 2, the radial distribution of heat transfer under each jet in a staggered, effusion depleted ARN system is virtually identical to that of a SRN, except in the small region where neighboring wall jets collide. Using the same geometry, Rhee (9) investigates the performance of the impingement-effusion geometry with a variable induced crossflow between the jet and target plates.

As the application of target plate depletion schemes is common in gas turbine applications, Metzger (1975) and Xu (2005) present local heat transfer and flow field measurements for particular application-based impingement-effusion geometries. Additionally, Jingzhou (2003) completed a numerical study and provided heat transfer distributions on either side of the target plate. In both Xu (2005) and Jingzhou (2003), the effusion holes were angled to reduce the risk of jet blow-off on the hot side of the target surface.

Section 2.2 Effusion (Film) Cooling

Section 2.2.1 Nomenclature

Figure 15 Film Cooling Array Geometry, Single Row (left) and Multiple Row (right)

Geometrical parameters used to describe film cooling are shown in Figure 15, below. Again, the x coordinate represents the mainstream (hot gas) flow direction and the y coordinate is across the span. Holes are of length L and diameter D, and may be inclined at angle α and compounded at angle β .

The ratio of coolant mass flux to mainstream mass flux is known as the Blowing Ratio, M, shown in (9). This dimensionless parameter is typically used to describe the flow rate of coolant and the jet behavior at the hole exit. The momentum flux ratio and the density ratio are also referenced often and are shown in (10) and (11). A relationship between the three is provided in (12).

$$M = \frac{\rho_c U_c}{\rho_m U_m} \quad (9)$$

$$I = \frac{\rho_c U_c^2}{\rho_m U_m^2} \quad (10)$$

$$DR = \frac{\rho_c}{\rho_m} \quad (11)$$

$$I = \frac{M^2}{DR} \quad (12)$$

The adiabatic film cooling effectiveness, η , shown in (13), is a dimensionless temperature that describes the relationship between the local temperature of the cooled surface to the coolant and mainstream temperatures, T_m and T_c . Since the wall temperature is dependent on the rate of heat transfer through the boundary layer and thermally conducting wall, the adiabatic wall temperature T_{aw} is used in the definition of η so as to create a standard definition. T_{aw} therefore represents the temperature of the gas directly adjacent to the cooled surface.

$$\eta = \frac{T_m - T_{aw}}{T_m - T_c} \quad (13)$$

The heat transfer coefficient on the film cooled wall is presented as the heat transfer coefficient, h , as shown in (14). The heat flux oriented into the wall is denoted as q'' .

$$h_f = \frac{q''}{T_w - T_{aw}} \quad (14)$$

Section 2.2.2 Film Cooling from a Single Row of Holes

The term “film cooling” is used to describe a cooling scenario in which coolant is injected into a high temperature boundary layer to reduce the temperature of the surface. Goldstein (1960) provides a thorough review of early film cooling studies, all of which were single row configurations – in other words, coolant was injected at a single streamwise position. The ideal case, slot cooling, provides an even lateral distribution of coolant and thus a uniform lateral temperature profile. See Figure 16 for a depiction of slot cooling, as compared to discrete hole cooling. Due to manufacturing and structural requirements, discrete hole film cooling is most common in practice. The downsides to this film injection technique are an uneven lateral distribution of coolant and inhibiting vortical structures that are the subject of much modern day film cooling research.

Figure 16 Thermal Profile on Cooled Surface with Slot Cooling (top) and Discrete Hole Film Cooling (bottom)

Discrete hole film cooling is typically accomplished with evenly spaced cylindrical holes. Cylindrical holes are used for their ease of manufacturing; however, to improve film cooling performance, the holes are inclined with respect to the mainstream. This reduces the tendency for the coolant to lift off of the surface at high blowing rates. Furthermore, the holes may be oriented at a compound angle with respect to the mainstream direction, as this increases the coverage, or proportion of open area across the span. Finally, the hole exit may be shaped – as in Figure 17 – through advanced manufacturing technology to suppress kidney vortices that naturally develop in a cylindrical hole (Landis, 2009). These vortices entrain the hot mainstream fluid and thus sustain jet blow-off.

Figure 17 Inclined Cylindrical Film Cooling Hole with Shaped Exit (Colban, 2008)

There are a variety of articles published regarding experimental and numerical single row, cylindrical hole, shaped exit film cooling studies. In fact, there have been several attempts to correlate the adiabatic film cooling effectiveness results using the vast array of data available. Colban (2008) provides such a correlation for shaped holes, while Baldauf (2002) captures the effect of a wide range of variables through a comprehensive parametric study. However, our review on these topics is brief, as this study is based on multi-row, unshaped, normal (inclination

angle $\alpha = 90^\circ$) injection. In fact, the type of multi-row injection used in this impingement-effusion cooling system design, known commonly as “Full Coverage Film Cooling”, belongs to a nearly independent field of film cooling research which is outlined in the following section.

Section 2.2.3 Full Coverage Film Cooling

Figure 18 Single Row Film Cooling Effectiveness Decay with Streamwise Position

Full coverage film cooling is used in practice when a large surface area needs to be cooled. As seen in Figure 18 (from Baldauf, 2002), single row injection at $\alpha = 90^\circ$ at low M yields high spanwise averaged values of η immediately downstream of injection, but this quickly decays with streamwise position. On the other hand, high M leads to lift-off and subsequent reattachment downstream, promoting high η as many as 80 hole diameters downstream. However, this comes at the expense of excess coolant used to maintain the high blowing rate.

Figure 19 Full Coverage Film Cooling, $Y/D = 7$ and $\alpha = 90^\circ$

Figure 19 provides results from a full coverage film cooling profile with low M from Harrington (2001). While η is initially low, due to the low M and high Y/D , a high level of effectiveness has been established by the 8th and 9th rows of injection. From a direct comparison of the two cases in Figure 18 and Figure 19, it is apparent that coolant is used more efficiently in the case of multi-row, low M injection as compared to single row, moderate or high M injection. This is primarily a result of the majority of blown off coolant, whose enthalpy is dissipated into the boundary layer.

Full coverage film cooling studies have primarily focused on unshaped cylindrical holes. Crawford (1980) presents heat transfer results for three arrangements: $\alpha = 90^\circ$, $\beta = 0^\circ$; $\alpha = 30^\circ$, $\beta = 0^\circ$; $\alpha = 30^\circ$, $\beta = 45^\circ$. It is concluded that inclined arrangements provide better surface protection both within the array and in the recovery region (downstream of the last row of holes). Arrays with a hole spacing of $5D$ outperform those with a spacing of $10D$, as significantly more coolant is injected in the case of the former. A minimum in cooled to uncooled Stanton number ratio shows a clear optimum blowing ratio of 0.4 ; as seen with single row injection, excess

coolant at higher M blows off, entrains hot fluid, and thus inhibits performance near the point of injection.

An earlier study by Mayle (1975) focuses only on arrays with $\alpha = 30^\circ$ and $\beta = 45^\circ$. Instead, Mayle examines the variation in hole spacing, up to a particularly high spacing of $X/D = 14$. It is concluded that η and the Stanton number fall off significantly with increasing X/D and Y/D , and the lateral η profile is highly non-uniform due to the integrity of the individual jets, as shown in Figure 20. The resulting sharp temperature gradients present a thermal stress issue and illustrate the advantages of collecting full coverage film cooling data on a local (rather than area-averaged) basis in order to evaluate variations in hole spacing and blowing ratio.

Figure 20 Spanwise Variation of Effectiveness, $X/D = 10$, $M = 1.0$

Two studies by Metzger (1973, 1976) pertain specifically to this study as in-line and staggered full coverage film cooling arrangements are compared. See Figure 21 for a depiction of Metzger's test surface. At low blowing ($M \leq 0.2$) the staggered geometry yields higher η in the latter half of the hole array. In fact, the in-line span average η profile levels off while the staggered profile continues climbing throughout the 10-row array. It is concluded that jet interaction reduces the efficiency of the in-line array, whereas a staggered jet may travel twice as

far downstream before being deflected by a new row of effusion. For a 4-row in-line or staggered configuration, $M = 0.2$ yielded the highest overall η for in-line and $M = 0.3$ for staggered. At higher M , the effects of jet lift-off (which is prevalent at relatively low M for normal injection) reduced performance.

Figure 21 In-line and Staggered Effusion Geometries (Metzger, 1973)

Section 2.2.4 Film Cooling Superposition

In an attempt to predict full coverage film cooling performance based on the wide range of available single row effectiveness data, various methods of superposition have been proposed. The most widely-used method, the Additive Film Method, was proposed by Sellers (1963) (see Figure 22) and subsequently used by Mayle, Metzger, and others. For this method, the η profile is calculated by adding the first row η with all the subsequent row η , where the subsequent row η is determined using the local value of T_{aw} (produced by upstream injection) in place of the mainstream T_m . This creates a total η profile that increases at a slower rate with each additional row of injection. Mayle (1975) showed generally good agreement between this method and experimental results for inclined, compounded film cooling holes in a staggered arrangement, as seen in Figure 23. Metzger also showed the Additive Film Method captured the general trend of staggered geometry η results, but over-predicted the in-line η . He suggested that each row's η profile be taken to zero after the second downstream interaction to compensate for this over-prediction. Similarly, the performance of the Additive Film Method at predicting five and fifteen row in-line and staggered η profiles will be discussed in this study.

Figure 22 Calculated Adiabatic Wall Temperature for Multirow Slot Cooling Scenario (Sellers, 1963)

Figure 23 Comparison of Experimental Results with Superposition Prediction (Mayle, 1975)

CHAPTER 3. METHODOLOGY

Section 3.1 Wind Tunnel

Figure 24 Coupled Impingement-Effusion Wind Tunnel

In order to perform high-resolution temperature measurements on complex geometries, a large open loop wind tunnel was constructed. The supporting structure for the tunnel had a footprint of approximately 10 m^2 and was 5 m tall. Two main flow paths are used to provide the coolant and mainstream flows, which combine at the point of effusion. See Figure 24 for a picture of the wind tunnel facility.

The primary components of the flow bench are a plenum and a cross flow duct, shown schematically in Figure 25. The cross flow duct represents the hot mainstream gases which the cooling system has been designed to protect against. The plenum provides a constant supply pressure for the impinging jets. Proper flow conditioning has been ensured through several

measures: (a) the plenum houses a splash plate which diffuses the flow at its inlet, preventing one large impingement jet through center of the plenum; (b) there are six alternating layers of honeycomb and mesh screens in the interior to produce a nearly flat, well-conditioned, velocity profile throughout; (c) the minimum plenum to jet array area ratio is 180 to ensure that this air velocity inside the plenum is negligible. The only driving factor for the impinging jets is the pressure ratio between the plenum and cross flow duct, in a similar fashion as a cooling system in a gas turbine.

Figure 25 Coupled Impingement-Effusion Wind Tunnel Schematic

The cross flow duct was designed with a span of three times that of the test specimen ensuring negligible wall effects in the cross flow velocity profiles. The height of the duct is three times the maximum effusion hole diameter. The size was determined from a balance of blower requirements and wall effects. An opening on the cross flow duct allowed the effusion plate to rest flush over the cross flow, so as to create a continuous, smooth surface between the test specimen and the duct wall. An acrylic window was positioned on the opposite side of the mainstream flow duct to allow optical access for temperature measurements. Light Emitting

Diode (LED) lighting, a Cooled Coupled Discharge (CCD) camera, and a Data Acquisition (DAQ) system were positioned below this window.

The plenum was pressurized by a vortex blower. Plenum flow rates were adjusted via gate valves to obtain the desired mass flow rate through the impingement-effusion system. The plenum air temperature was measured with three T-type thermocouples located near the test section, and the plenum pressure was measured with three static pressure ports positioned at different heights in the plenum near the test specimen. The cross flow was driven by a centrifugal blower under suction. This blower was controlled with a variable frequency drive so that the mainstream flow rate could be adjusted and thus allowing flexibility in test conditions.

Section 3.2 Test Specimen Geometry

Figure 26 Coupled Impingement-Effusion Cooling System Schematic

The impingement-effusion system is designed so that the impinging jets strike the cold side of the effusion plate. These jets are staggered with respect to the effusion holes in both the x- and y-directions, in the same manner as Rhee (2001). The nomenclature “cold side” and “hot side” is used to demonstrate that, in a gas turbine cooling system, the side of the effusion plate adjacent to the mainstream flow would be hotter than the reverse side of the effusion plate. The coolant enters the effusion array after impingement and subsequently is injected into the

mainstream flow. A schematic identifying the nomenclature used and the coolant flow through the system is provided in Figure 26.

Each impingement-effusion geometry was box-shaped and constructed of machined acrylic sheet. For each effusion plate made of acrylic, a counterpart was constructed from Rohacell foam for effusion testing. See Figure 27 for a picture of an assembled test specimen. The impingement plate and effusion plate had 24 and 35 holes, respectively, for coolant flow. These were interchangeable along with four acrylic walls which came in tall (“High Z”) and short (“Low Z”) heights. Impingement and effusion holes of different diameters were used to create a variety of geometries. A complete test matrix, with all non-dimensional geometrical and flow parameters identified, is presented in Table 1.

Figure 27 Acrylic Impingement-Effusion Test Specimen

Table 1 Coupled Impingement-Effusion Test Matrix

Geometry	<i>Impingement</i>					<i>Effusion</i>			
	X_i/D_i	Y_i/D_i	Z/D_i	Arrangement	Re	X_e/D_e	Y_e/D_e	Arrangement	M
A	4.0	3.6	2.4	In-line	10000	2.3	2.0	In-line Staggered	0.15
			10.5		20000				0.20
					30000				0.25
B	5.4	4.8	3.1	In-line	10000	3.0	2.5	In-line	NOT TESTED
			13.7		20000				
					30000				
C	6.8	6.1	3.8	In-line	10000	4.7	3.4	Staggered	NOT TESTED
			16.9		20000				
					30000				
D	7.9	7.0	4.4	In-line	10000	4.2	3.9	In-line	NOT TESTED
			19.6		20000				
					30000				
E	9.1	8.2	5.0	In-line	NOT TESTED	4.7	4.0	Staggered	0.15
			21.9						0.20
									0.25
F	9.5	8.5	5.3	In-line	10000	5.4	4.5	In-line	0.15
			23.4		20000				0.20
					30000				0.25

To enable a more detailed discussion of full coverage film cooling results, two additional film cooling arrays were tested. Both had larger hole spacing (X_e/D_e and Y_e/D_e) and more rows than the coupled impingement-effusion geometries. Rather than be fed by a impingement jet plate, a uniform pressure plenum was used as the inlet boundary condition. A wide range of blowing ratios were able to be studied. Also, the effusion holes, while still cylindrical in shape, were inclined with respect to the mainstream flow direction by inclination angle α and set at a compound angle β with respect to the spanwise direction. See Figure 28 for a schematic of the full coverage film cooling array test setup, and Table 2 for a complete test matrix.

Figure 28 Full Coverage Film Cooling Schematic

Table 2 Full Coverage Film Cooling Test Matrix

FC Geometry	x_e/D_e	y_e/D_e	M	Arrangement	# Rows	α	β
1	12.1	14.0	0.50	Staggered	15	30°	45°
			1.0				
			1.5				
			2.0				
2	12.1	14.0	0.25	Staggered	15	30°	0°
			0.50				
			0.75				
			1.0				
			1.5				
			2.0				

Section 3.3 Flow Measurement

As seen in Figure 25, the coolant supply is fed by a blower under pressure and the simulated hot gas (mainstream) flow is pulled under suction by a fan with VFD control. In order to control the coolant flow rate, gate valves by the blower were manually adjusted. The discharge coefficient (C_d) of each impingement geometry was determined separately, so that the coolant mass flow rate could be determined based on the pressure ratio across the impingement plate.

C_d is the ratio of the actual mass flow rate to an ideal mass flow rate based on pressure ratio. The use of a separate, simple rig (shown in Figure 29) to measure C_d helped to ensure no leaks were present. The actual mass flow rate was measured by a calibrated Venturi flow meter. All measured C_d were in the range of 0.76 to 0.82, or close to 0.8, the general guideline value for

flow through a sharp-edged orifice (Florschuetz, 1981). For a given geometry, C_d was constant over all Reynolds numbers being tested, a trend also identified by Florschuetz (1981). (15) shows the relationship between actual mass flow rate, discharge coefficient, and ideal mass flow rate through a given area, as a function of pressure ratio.

$$C_d = \frac{\dot{m}_{actual}}{\dot{m}_{ideal}} = \frac{\dot{m}_{actual}}{AP_0 \left(\frac{P}{P_0} \right)^{\frac{\gamma+1}{2\gamma}} \sqrt{\frac{2\gamma}{(\gamma-1)RT_0} \left[\left(\frac{P_0}{P} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}} \quad (15)$$

Figure 29 Discharge Coefficient Test Rig Schematic

The impinging jet Reynolds number, rather than the actual coolant mass flow rate, was set to the desired value through control of the gate valves. This Reynolds number may be calculated per (16). It is assumed that an equal amount of coolant flows through each of the 24 impingement holes, hence the factor of 1/6. This assumption was validated by measuring the static pressure distribution along the mainstream flow duct underneath the effusion plate. Only negligible pressure variations were recorded along the length of the test section, indicating a constant sink pressure and hence validating this assumption.

$$\text{Re}_D = \frac{\dot{m}_{ideal}}{6\pi D_i \mu} \quad (16)$$

The fan speed control allowed manual adjustment of the hot gas flow rate. This fan was equipped with a calibrated nozzle that allowed calculation of the mass flow rate through the fan inlet as a function of the static gage pressure at the inlet. The empirical formula for this flow rate calculation is shown in (17). Using (16) and (17), the film cooling blowing ratio (and hence the density ratio and momentum flux ratio) were able to be determined.

$$\dot{m}_{total} = 2.15 \rho \sqrt{\frac{1.204}{\rho} |P_{st}|} \quad \left[\frac{kg}{s} \right] \quad (17)$$

Section 3.4 Temperature Sensitive Paint

Figure 30 Jablonski Energy Level Diagram (Adapted from Bell, 2001)

The distributions of temperature on both the cold side and hot side of the effusion plate were captured using temperature sensitive paint (TSP). TSP is made by combining a luminescent molecule and polymer binder. Both the acrylic and Rohacell surfaces were coated with this paint and illuminated via LEDs with a 475nm wavelength, exciting the luminescent molecule. This molecule returns to its original energy state through the competing effects of emission of the

longer wavelength light and thermal quenching. With increased temperature, the probability of thermal quenching increases, thus the intensity is inversely proportional to the temperature. This process is indicated by the Jablonski energy level diagram shown as Figure 30. A full explanation on the experimental use of TSP used for heat transfer measurements is outlined by Liu (2006).

The intensity variation was captured by means of a CCD camera and high pass filter, and the camera takes images at a minimum rate of 200 to 350 ms to capture the fluorescence at steady state. A 1200x1600 resolution picture is taken with the camera. The pictures are then processed using a MATLAB code which took the raw image files and processed the corresponding temperature distributions. The calibration curve for TSP is based off of intensity ratio as a function of temperature difference. Two pictures are needed to gather a temperature distribution. One picture is needed as a reference (the “cold” image) with a known uniform temperature on the entire surface. The second is the test data (the “hot” image) containing the unknown temperature profile. This method of calibration versus intensity ratio is rather insensitive to lighting and paint variations. To reduce noise, four images were taken and averaged for both the hot and cold data.

Figure 31 TSP Calibration Experimental Setup

The TSP calibration curve was obtained through the experiment depicted by Figure 31. A 0.5" sheet of acrylic was painted with 6 layers of TSP as the calibration piece. This piece was placed on a copper block on top of a small electric heater which was used to heat the test piece to 10 different temperatures. With a thermocouple monitoring the surface temperature, thermal steady state was monitored and temperatures were recorded. This calibration allowed the use of the formula shown in (18), (19), and (20) below to obtain temperature as a function of an intensity ratio. The resulting TSP calibration curve is presented in Figure 32.

$$\frac{I}{I_{ref}} = C_1\theta^3 + C_2\theta^2 + C_3\theta + C_4 \quad (18)$$

$$Picture \quad Ratio = PR = \frac{I}{I_{ref}} \frac{Hot \quad Picture \quad Average}{Cold \quad Picture \quad Average} \quad (19)$$

$$T(x, y) = C_5PR^3 + C_6PR^2 + C_7PR + C_8 \quad (20)$$

Figure 32 TSP Calibration Curve

Sample images are shown in Figure 33 to demonstrate the relative intensity of the reference and test images. One can observe the uniformity of luminescent intensity on the reference image, relative to the sharp intensity gradients in the test image. The LEDs used to excite the fluorescent molecules in the TSP were diffused with opaque acrylic sheets to ensure a relatively uniform intensity distribution. To determine temperature, the ratio of reference to test intensity at each pixel of camera resolution is evaluated, and then converted to a temperature ratio via the calibration in Figure 32. This prevents small non-uniformities caused by uneven paint or light distribution on the test surface from affecting the result.

Figure 33 Image Processing with TSP

Section 3.5 Heaters

Thin foil heaters were manufactured for each geometry to provide a constant heat flux boundary condition for heat transfer measurements. These heaters are made out of $\delta = 5.08 \times 10^{-5}$ m thick Stainless Steel Type 321. The electrical resistivity of 321 SS is $\rho_e = 720 \times 10^{-7}$ Ohm-m. These strips were backed with double sided Kapton tape and then installed in between rows of effusion holes. Kapton tape was used because it is electrically insulative but thin enough to have a negligible temperature difference across its thickness.

Initial tests were carried out with foil heaters which covered the entire surface, with holes cut out at the effusion locations. Figure 34 shows this heater compared with the standard, constant cross-sectional width heaters. The variable heat fluxes were determined using Ansoft's Maxwell solver, and lateral conduction corrections were accounted for. However, due to the large variations in heat flux, the validation of these variable heat flux heaters did not produce acceptable results, and the decision was made to sacrifice surface area and use constant width heater strips.

Figure 34 Variable Flux vs. Constant Flux Heaters

To power these heaters, the ends were clamped to copper bus bars with leads attached to a DC power supply. The power supply used was a Xantrex DC with a maximum voltage of 7.5V and a maximum current of 300A. Voltages were varied depending on the heat transfer scenario to maintain the highest possible surface temperature that would not damage the acrylic or weaken the Kapton adhesive. As the heaters are relatively thin, variation in temperature through the heater thickness is assumed to be negligible. This is a valid assumption as the Biot number is 3×10^{-3} and therefore convection is dominant over conduction.

Section 3.6 Data Reduction

In a typical coupled impingement-effusion cooling system, the temperature of the target surface is governed by: (a) the cold side impingement heat transfer coefficient, h_i ; (b) conduction in the wall; (c) convection to the coolant as it passes through the effusion holes; (d) the hot side film cooling effectiveness, η ; (e) the hot side film cooling heat transfer coefficient, h_f ; and (f) radiation exchange between the surface and its surroundings. For the sake of simplicity, a 1-D heat transfer circuit is presented in Figure 35 that includes only (a), (b), (d), and (e).

Figure 35 Simplified Thermal Circuit for Effusion Plate

In order to measure h_i , h_f , and η , the effect of each must be isolated so that the other modes of heat transfer do not influence the results. In some cases, it is possible to reduce one of

these parameters to nearly zero; in others, the unwanted heat loss is estimated via analytical or experimental results. The method of heat transfer calculation and the assumptions made are described in this section.

Section 3.6.1 Impingement Heat Transfer Data Reduction

Figure 36 Impingement Heat Transfer Test Setup

In order to measure impingement heat transfer, the cold side of the effusion plate was painted with TSP and then affixed with double-sided Kapton and thin foil heaters. The TSP was then viewed by the camera through both the acrylic window and the acrylic effusion plate. See Figure 36 for a depiction of this arrangement.

The heaters would reach temperatures between 50°C and 80°C, while the impinging jets were typically 35°C. The resistance of each heater was calculated using the dimensions of the heater as shown in (21), where L is the length of the heater (in the direction of current flow) and w is the width. The voltage drop across each heater (ΔV) was measured in situ and the total heat generation could then be calculated using (22).

$$R = \frac{\rho_c L}{w \delta} \quad (21)$$

$$q_{in} = \frac{(\Delta V)^2}{R} \quad (22)$$

Figure 37 Result of Impingement Heat Transfer Heat Loss Test

Next the heat losses to conduction and radiation were considered. In order to estimate these, a heat loss test was run in which the plenum and test specimen were filled with insulation. Meanwhile the power supply and cross flow blower were run and the system was allowed to reach thermal steady state at several different set points. The resulting data was correlated and an expression for the lost heat flux, q''_{loss} , was determined. This data is provided as Figure 37.

Additionally, an estimate of the lateral conduction in the heater was determined using finite differencing methods and the local temperature profile. Figure 38 along with (23), (24), (25), and (26) describe this lateral conduction correction method. The final value of heat flux used in the calculation of heat transfer coefficient is then determined using (27).

Figure 38 Finite Differencing Corresponding to the Temperature at a Single Pixel

$$q_{x+\Delta x} = k\delta\Delta y \frac{T(x, y) - T(x + \Delta x, y)}{\Delta x} \quad (23)$$

$$q_{x-\Delta x} = k\delta\Delta y \frac{T(x, y) - T(x - \Delta x, y)}{\Delta x} \quad (24)$$

$$q_{y+\Delta y} = k\delta\Delta x \frac{T(x, y) - T(x, y + \Delta y)}{\Delta y} \quad (25)$$

$$q_{y-\Delta y} = k\delta\Delta x \frac{T(x, y) - T(x, y - \Delta y)}{\Delta y} \quad (26)$$

$$q'' = \frac{q_{in}}{\Delta x \Delta y} - q''_{loss} - q_{x+\Delta x} - q_{x-\Delta x} - q_{y+\Delta y} - q_{y-\Delta y} \quad (27)$$

The heat transfer coefficient is calculated using this heat flux, the local temperature profile $T(x, y)$ and the impinging jet temperature T_j as shown in (28). This heat transfer coefficient is converted to a dimensionless Nusselt number for comparison between geometries using (29).

$$h = \frac{q''}{T(x, y) - T_j} \quad (28)$$

$$Nu_D = \frac{hD_i}{k_f} \quad (29)$$

Section 3.6.2 Film Cooling Effectiveness Data Reduction

Film cooling effectiveness, η , is a dimensionless temperature ratio, as shown in (13). Measuring η requires a large temperature difference between the coolant temperature (T_c) and the mainstream, or cross flow, temperature (T_m). This was accomplished by heating the coolant and allowing the cross flow to remain at room temperature. (Heating the coolant does cause $DR < 1$, which would not be the case in practical cooling scenarios. Baldauf (2002) shows DR to have a second-order effect on film cooling performance; however this effect is outside of the scope of this study.) With T_c and T_m known via thermocouple measurements, the test surface was painted with TSP so that T_{aw} (and hence η) could be calculated. See Figure 39 for a schematic of this test setup.

The acrylic effusion plate was replaced with an identical Rohacell effusion plate for these tests. Rohacell is a stiff closed-cell foam, ideal to represent an adiabatic wall because of its low thermal conductivity. At $k \approx 0.026 \text{ W/mK}$, Rohacell is an order of magnitude more thermally resistive than acrylic. This yielded an error in T_{aw} (due to heat leakage) of approximately 0.3-0.5°C instead of 3-5°C as would be seen when using acrylic.

In order to account for the ever-present heat leakage that cause the TSP measurement to be slightly higher than the true adiabatic wall temperature, an estimate of heat leakage was determined using the 1D thermal circuit provided in Figure 35. The hot side heat transfer coefficient, h_f , was assumed to be unaffected by the addition of coolant, and flat-plate correlations (presented in detail in the following section) were used to provide an estimate. It can be shown that this is a good assumption, as the hot side convective resistance is relatively low,

and changing it by $\pm 50\%$ does has only a $\pm 0.05^\circ\text{C}$ impact on T_{aw} . Next, the conduction resistance was determined analytically, and the cold side convective heat transfer coefficient was taken from impingement heat transfer results. (An area-averaged value was used here as T_{aw} is relatively insensitive to fluctuations in h_i . Also, it should be noted that only High Z impingement geometries were used, in an effort to reduce the heat loss through the Rohacell Plate.)

Figure 39 Film Cooling Effectiveness Test Setup

Section 3.6.3 Film Cooling Heat Transfer Data Reduction

Film cooling heat transfer measurements required the use of an acrylic effusion plate, though not for its optical clarity (as was the primary justification in the case of impingement heat transfer): the double-sided Kapton tape that bonded the foil heaters to the surface did not bond effectively with Rohacell. Due to this, the heat loss through the test plate had a considerable impact on the uncertainty in the heat transfer tests. Conduction heat loss was determined analytically, using heater temperature T_w and previously measured values of h_i . See (30) for this relationship. Radiation heat losses were not accounted for due to the uncertainty in emissivity and view factor in the test setup; it is estimate that an error of up to 5% is caused by neglecting radiation.

$$q''_{loss} = \frac{T_w - T_c}{\frac{t}{k_{ACRYLIC}} + \frac{1}{h_i}} \quad (30)$$

The data collection system was set up in a similar manner as in the case of impingement heat transfer testing, but the heaters (rather than the acrylic test surface) were coated with TSP. The heaters were oriented normal to the mainstream flow, with gaps at each effusion row. See Figure 40 for a schematic of this test setup, and Figure 41 for a picture of an assembled effusion plate.

Figure 40 Film Cooling Heat Transfer Test Setup

Figure 41 Image of TSP-Coated Heaters Applied to Effusion Test Surface

To calculate the heat transfer coefficient h_f , it is necessary to either control the coolant temperature such that $T_c = T_m$ and R_η is effectively zero, or to apply a correction based on measured values of η . (This can be realized by noting the definition of film cooling heat transfer coefficient in (14), where T_{aw} must be equal to T_m , which can be measured.) It was not possible (without actively cooling or heating an air stream) to set the two flow temperatures equal, but they were held as close as possible to ensure the correction was minimal.

This correction is determined from the following rearrangement of (13) and (14):

$$T_{aw} = T_m - \eta(T_m - T_c) \quad (31)$$

$$h_f = \frac{q''}{T_{aw} - T_w} = \frac{q''}{T_m - \eta(T_m - T_c) - T_w} \quad (32)$$

Dubbed the “superposition method” by Gritsch (1999), this relationship is widely used to determine h_f or η based on the other. Often, a new dimensionless temperature θ is introduced, conveniently defined (as in (33)) to reduce the complexity of (32) to the expression in (34). Furthermore, a general heat transfer coefficient h may be defined as a function of θ , and calculation of h at two or more values of θ allows for extrapolation of both h_f and η . See (35) for this relationship.

$$\theta = \frac{T_m - T_c}{T_m - T_w} \quad (33)$$

$$h_f = \frac{q''}{(T_m - T_w)(1 - \eta\theta)} \quad (34)$$

$$h(\theta) = \frac{q''}{T_m - T_w} = h_f(1 - \eta\theta) \quad (35)$$

Once h_f is determined, it is necessary to compare this value to the heat transfer coefficient on an uncooled surface with the same mainstream flow conditions. Without this comparison, h_f alone is only applicable to a film cooling array with identical geometrical and flow parameters as the test specimen. (It is for this reason that we prefer to non-dimensionalize all results.) The ratio h_f/h_0 , hereby referred to as the heat transfer enhancement, was determined using an analytical method to determine the uncooled flat-plate heat transfer coefficient h_0 .

The flat plate heat transfer scenario exhibited by the heater setup on the effusion plate is more complex than standard correlations predict; however it may be represented by a flat plate with unheated starting length in an external flow, with succeeding constant flux and adiabatic portions whose behavior can be predicted using superposition of existing correlations. See Figure 40 for pictorial representation of this analytical problem. By using a flat-plate correlation, the

effect of the dimpled surface (i.e. effusion holes) is neglected, providing a realistic estimate of heat transfer without the presence of film cooling.

Figure 42 Schematic of Effusion Plate No-Film Heat Transfer Scenario

The Reynolds number is calculated based on the free stream velocity U and the fluid kinematic viscosity ν as shown in (36). The heat transfer Nusselt number as a function of position on a flat plate with a constant heat flux boundary condition can be calculated per (37). For turbulent flow and an unheated starting length, the relationship in (38) provides the Nusselt number amplified due to the unheated starting length ξ . Finally, superposition of multiple unheated starting length solutions is completed per (39) to achieve the heat transfer coefficient h_0 . In this equation N is the total number of transitions on the test surface from an adiabatic to constant flux boundary condition (in other words, the total number of heaters multiplied by two).

The solution for h_0 assumes lateral uniformity of heat transfer, as is expected for a boundary layer of significant width with uniform lateral velocity distribution throughout. The behavior of (39) is represented in Figure 42, where h_0 peaks at the thermal boundary layer restart at the beginning of each heater and trails off until the adiabatic portion (where it is not measured). It then peaks again, but to a slightly lower value, at the next heater.

$$\text{Re}_x = \frac{Ux}{\nu} \quad (36)$$

$$Nu_x|_{\xi=0} = 0.0296 \text{Re}_x^{4/5} \text{Pr}^{1/3} \quad (37)$$

$$Nu_{x,\xi_n} = \frac{Nu_x|_{\xi=0}}{\left[1 - \left(\frac{\xi_n}{x}\right)^{9/10}\right]^{1/9}} \quad (38)$$

$$h_0 = \left(\frac{k_f}{x}\right) \sum_{n=1}^N \begin{cases} 0 & x \leq \xi_n \\ (-1)^{n-1} Nu_{x,\xi_n} & x > \xi_n \end{cases} \quad (39)$$

Using the calculated value of h_0 , and the method for calculation h_f when θ is nonzero, the heat transfer enhancement due to the injection of film was quantified. First, as was the case in impingement heat transfer, a heat balance was completed to determine the convective heat flux (see (27)). Then, h_f was calculated as in (32) and h_0 was calculated as in (39). Finally, the heat transfer enhancement was determined as shown in (40).

$$\text{Heat Transfer Enhancement} \equiv \frac{h_f}{h_0} \quad (40)$$

Section 3.7 Experimental Uncertainty

Measurement uncertainty was determined from the using method presented by Kline (1953). Accuracies of all calibrated instruments and measurement techniques were used to obtain this uncertainty. Additionally, multiple thermocouple and pressure taps were used for each respective temperature and pressure measurement and an average value was taken. Total uncertainty in heat transfer coefficient is approximately 12%, while total uncertainty in film cooling effectiveness is 6%.

CHAPTER 4. RESULTS

Section 4.1 Experimental Validation

Section 4.1.1 Impingement Heat Transfer Validation

In order to validate impingement heat transfer measurements, a test specimen was constructed with two rows of impinging jets and a blank effusion plate with a slot at the downstream edge. The purpose of this test specimen was to create an impingement channel similar to Florschuetz's from the 1981 study in which a correlation was derived. The geometrical parameters corresponding to Florschuetz's diagram in Figure 13 are provided in Table 3. It can be observed that these parameters are within the limits of the correlation provided in (8).

Table 3 Validation Impingement Array Test Parameters

X_i/D_i	Y_i/D_i	Z/D_i	Re_D
4.0	3.6	3.0	52000

Figure 43 Florschuetz-Style Validation Impingement Array Test Setup

The distribution of h_i can be observed in Figure 44. The heat transfer distributions under neighboring jets are qualitatively similar, and there is a decay in heat transfer from the first row

to the second due to the building cross flow in the channel. It should be noted that, in order to mimic Florschuetz’s experimental conditions, two additional “dummy” jets were located above and below the limits of Figure 44 in order to reduce the effect of the wall and create periodic behavior that would be observed in an infinitely wide jet array.

Figure 44 Florschuetz-Style Impingement Array for Heat Transfer Validation

The data in Figure 44 was averaged over the span to provide the results of Figure 45. Due to the heater setup in Florschuetz’s study, only span averaged heat transfer coefficients were measured. Also, due to the heater width of two jet diameters, the span averaged values represent a spatial average with a length of two jet diameters. Accordingly, the validation data was averaged and compared directly to the values predicted by Florschuetz’s correlation, and a good agreement was observed. Florschuetz’s correlation predicted 6-10% higher heat transfer than observed in this study, but the discrepancy falls within the experimental uncertainty. It is possible that over-compensation for heat losses or a slight overestimate of jet Re_D led to this discrepancy; regardless, the impingement heat transfer test setup and data reduction methods were successfully validated.

Figure 45 Span Averaged Heat Transfer Compared With Florschuetz Correlation

Section 4.1.2 Film Cooling Effectiveness Validation

In order to validate effectiveness measurements, a Rohacell effusion plate was manufactured with a single effusion hole, $L/D_e = 1.33$. The L/D_e was chosen to match published data by Goldstein & Cho [4]. Resonator sidewalls were added to the effusion plate for structural support. The coolant mass flow rate was controlled by an orifice plate (impingement $Z/D_i > 20$) with four holes open upstream of the effusion location, shown in Figure 46. This ensured a plenum-like scenario within the box and no effect of impingement dynamics at the effusion hole. Tests were run at two different blowing ratios, $M = 0.22$ and $M = 0.57$. The local effectiveness data is presented in Figure 47.

Figure 46 – Effectiveness Validation Plate Test Setup

Figure 47 – Single Hole Effectiveness Validation Data

The span averaged data from each of these tests was compared to Goldstein & Cho's span average and many similarities were found, particularly at the higher blowing ratio. This comparison is provided in Figure 48. However, when comparing local data it was evident that lateral spreading was more prevalent in the experimental data, as the centerline effectiveness was lower than Goldstein & Cho while the $Y/D = \pm 1.0$ data was higher. Also, the $M=0.22$ span average indicated that the higher effectiveness than expected downstream.

Figure 48 – Span Average Effectiveness Data vs. Goldstein & Cho

Published effectiveness data from another source, Harrington [5], was only available as a local contour plot. This allowed for a qualitative investigation the effectiveness contours from both sources. These are provided in Figure 49. Table 4 compares the test parameters of both sets of published data to the validation data test parameters. It is immediately apparent that the peak effectiveness measured by Goldstein & Cho ($\eta \approx 0.8$) is much higher than Harrington's ($\eta \approx 0.5$). Goldstein's data also suggests a steeper gradient of decreasing effectiveness laterally. The differences between these two data sets prove that there is a great deal of measurement uncertainty near the effusion hole, mostly due to lateral conduction effects. When these two contours are laid over the validation data in Figure 50 they show that the validation data is clearly bounded by Goldstein on the higher end and Harrington on the lower end.

Figure 49 – Published Effectiveness Contour Plots

Table 4 – Validation Test Parameters

Data	α	M	L/D	Re_D
Goldstein & Cho (1995)	90°	0.22	1.33	14000
Harrington et al. (2001)	90°	0.25	1.0	4000
Validation	90°	0.22	1.33	~50000

Figure 50– Published Contour Plots Overlay on Validation Data

Section 4.1.3 Film Cooling Heat Transfer Validation

Film cooling heat transfer measurements were validated using existing correlations for non-film-cooled flat plates in an external flow, as relatively little literature exists predicting film

cooling heat transfer enhancement. Per these correlations, the heat transfer rate is infinite at the leading edge of the plate and drops as a power-law function.

A validation plate was constructed with a single wide heater and no effusion holes. A diagram of the validation plate is provided in Figure 51. The span averaged HTC is provided in Figure 52. Due to the way the heater was secured to the bus bars, some wrinkling of the heater took place during the experiment, causing a “wavy” appearance to the data. However, all the data points matched the validation estimate within the test’s 12% uncertainty.

Figure 51 – Hot Side HT Validation Test Plate

Figure 52 – Hot Side HT Validation Data

Further validation tests were run to investigate the accuracy of the superposition method for predicting the heat transfer coefficient over a flat plate with heated and unheated portions. Figure 53 shows a diagram of the test plate with multiple narrow heater strips and gaps between them. The heaters and gaps were placed in the configuration similar to the anticipated setup of Geometry A effusion plates, where X/D hole spacing was at its minimum (in other words, the gap distance between heaters was maximized). The results, shown in Figure 54, prove the validity of the analytical method.

Figure 53 – Modified Hot Side HT Validation Test Plate

Figure 54 – Superposition Method Validation Data

Given these results, the experimental methodology used in obtaining heat transfer and effectiveness measurements was determined to be validated, and full-scale testing was able to commence.

Section 4.2 Impingement Heat Transfer

A single impingement array was first tested with two different effusion geometries in order to evaluate the sensitivity of impingement Nusselt number with respect to effusion hole diameter. As seen in Table 1, Geometry A and Geometry A+ are identical except for effusion hole diameter. Both Low Z ($Z/D_i = 2.4$) and High Z ($Z/D_i = 10.4$) configurations were tested, each at three different jet Re_D . The purpose of this testing was two-fold: (a) to investigate whether effusion hole diameter would affect impingement heat transfer in such an impingement-effusion cooling system; and (b) to evaluate the necessity to construct a unique effusion plate for each impingement geometry for impingement heat transfer testing.

Figure 55 provides an illustration of the Geometry A and Geometry A+ heater configurations. For Geometry A+ the heaters ran perpendicular to the streamwise (with respect to the cross flow underneath the heat transfer test surface) direction, while the Geometry A heaters were aligned parallel to the streamwise direction. These heater orientations were chosen to investigate the feasibility of aligning the bus bars and electrical attachments in either orientation, so as to choose a preferred arrangement.

While the heaters were attached to the effusion plate from one end to the other, only a section of the plate was coated with TSP and therefore heat transfer data was only collected for this section. The coated region is denoted by the green box in Figure 55. It is recognized that a finite width array of jets will not produce a uniform periodic heat transfer distribution throughout because the walls of the test specimen will affect the flow distribution near the edges of the

array. Therefore, only the data shown in the red box of Figure 55 was used in determining the average Nusselt number for a given impingement configuration. This ensures the behavior of the center four jets is captured while the surrounding jets act as “dummy” jets (to set up the appropriate boundary condition on the center four).

Figure 55 Heater Configuration on Geometry A/A+ Effusion Plates

Section 4.2.1 Geometry A/A+ Impingement Heat Transfer

With these considerations in mind, local Nusselt number results for Geometry A and Geometry A+ are presented below. Figure 56, Figure 57, and Figure 58 show data for Geometry A with $Z/D_i = 2.4$. Data for each of three different impinging jet Re is provided. Similarly, data for Geometry A with $Z/D_i = 10.4$ is shown in Figure 59, Figure 60, and Figure 61. Geometry A+ results may be found in Figure 62, Figure 63, Figure 64, Figure 65, Figure 66, and Figure 67.

Note that the scale in each figure has been adjusted to best show the details of the heat transfer distribution; therefore, scales are not identical and the Nusselt number “color” cannot be compared among different jet Re_D . Some Nu_D scales have an origin at zero, while others do not. The black regions do not contain data; these portions indicate where no heater was attached. Effusion holes are drawn on each figure for reference. Impingement stagnation points are located at the points where $x/X_e = 1.5, 2.5, 3.5,$ and 4.5 intersects $y/Y_e = 0.5, 1.5, 2.5,$ and 3.5 .

Figure 56 Geometry A (Low Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 57 Geometry A (Low Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 58 Geometry A (Low Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 59 Geometry A (High Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 60 Geometry A (High Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 61 Geometry A (High Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 62 Geometry A+ (Low Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 63 Geometry A+ (Low Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 64 Geometry A+ (Low Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 65 Geometry A+ (High Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 66 Geometry A+ (High Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 67 Geometry A+ (High Z) Impingement Nusselt Number, $Re_D = 30000$

Through investigation of the local data presented above, it can be quickly observed that: (a) heat transfer increases with jet Re_D ; (b) the “Low Z ” geometries yield considerably higher heat transfer; and (c) the heat transfer distributions are highly non-uniform for $Z/D_i = 2.4$, yet highly uniform for $Z/D_i = 10.4$. The result in (a) is expected (and consistent with the reviewed literature) as the impinging jet strikes the target surface with higher momentum at higher Re_D , and is able to convectively extract more heat.

Both (b) and (c) are consistent with the velocity profiles for a free jet discussed in detail by O’Donovan (2005) (also presented in Figure 11). As the jet travels away from its circular origin, the fluid that is affected by the jet forms an ever-expanding truncated cone. Through air entrainment the momentum of the jet is either lost to viscous dissipation or distributed through the jet’s increasing cross-sectional area. By the point at which the jet reaches the stagnation zone (as identified in Figure 10), it has a much greater diameter for $Z/D_i = 10.4$ as compared to $Z/D_i = 2.4$. The distribution of Nusselt number reflects this enlarged stagnation zone; the magnitude of Nusselt number in this zone decreases as jet momentum decreases.

In the case of $Z/D_i = 2.4$, the heat transfer distribution highlights not only the stagnation zone, but also the formation of the wall jet. As fluid travels radially outward, heat must be convected through a thickening boundary layer. Also, wall jet momentum decreases with increasing radial position from the jet stagnation point, and heat transfer decreases accordingly. In all $Z/D_i = 2.4$ data (but most notably in Geometry A, $Re_D = 30000$ data) a secondary Nusselt number peak is formed along the intersection of neighboring wall jets. A similar result was reported by Cho (2008) in his investigation of impingement-effusion arrays, and he identified increased turbulent kinetic energy (and hence increased heat transfer) at the point where the wall jets collide. Note that this phenomenon is solely the result of the impingement array

configuration and not related to the “secondary peak” in heat transfer witnessed for a single impinging jet by Lytle (1991) and others.

Using the data enclosed by the red box in Figure 55, average Nusselt numbers were determined for each data set. The results are summarized in Table 5, below. Each result from Geometry A+ is compared with that of Geometry A, and it is shown that alternating the effusion hole diameter is not directly correlated to an increase or decrease in heat transfer. Rather, all results fall within the range of experimental uncertainty for heat transfer measurements, and therefore the differences between Geometry A+ and Geometry A cannot be conclusively identified through such an experiment. This fact drives the decision to use the Geometry A effusion plate for all subsequent impingement heat transfer testing, as this allows for a more efficient progression through the test matrix and heat transfer data covering a greater portion of the impingement target surface than would otherwise be possible.

Table 5 Area Averaged Nusselt Number, Compare Geometries A/A+

Z/D	Re	Average Nu		% Difference
		Geometry A+	Geometry A	
2.4	10000	91.9	81.5	-11%
	20000	129.1	119.3	-8%
	30000	159.1	166.0	+4%
10.4	10000	38.1	37.0	-3%
	20000	59.0	61.3	+4%
	30000	75.5	79.9	+6%

The data for Geometry A and Geometry A+ was also laterally averaged and plotted in Figure 68 and Figure 69. This allows for both direct comparison of the two geometries as well as a representation of the effect of jet Re on heat transfer. It can be observed that increasing jet Re_D from 10000 to 20000 generates a larger increase in Nu than is seen in the increase from 20000 to 30000. In other words, Nu_D is proportional to Re_D^m where the exponent m is less than unity; a similar trend was expressed by Goldstein (1986) for a single impinging jet.

Figure 68 also clearly shows that the local maximum in Nu_D created by interaction of neighboring wall jets becomes more significant (with respect to the global maximum) as jet Re_D increases. It may be concluded that higher jet Re (and hence higher wall jet momentum) leads to more turbulence generation upon collision of neighboring wall jets. It should be noted that, in single jet impingement studies (Lee (1999), for example) the magnitude (and, in fact, position) of the secondary heat transfer peak were functions of jet Re . The relative magnitude of this second peak increased with increasing Re as the turbulent kinetic energy in this ring increased with increasing wall jet momentum.

Figure 68 and Figure 69 allow for a quick quantitative analysis of the data displayed in Figure 56 through Figure 67, but due to the different heater alignment, one must view these with a few caveats. First, it appears the stagnation point behavior of Geometry A is significantly different than that of Geometry A+ in Figure 68; the peaks in Nu_D appear much less significant. In fact, the stagnation point behavior is very similar for the two cases, but as the lateral average for Geometry A captures more off-peak results than the lateral average for Geometry A+, the local Nu_D maxima is subdued by this additional data. Also, in Figure 69, some data near the edge of each heater on Geometry A suggests an increase in Nu_D away from the stagnation point. This result is partly erroneous due to measurement error at this point caused by lateral conduction in the acrylic effusion plate; however a local increase in Nu_D was observed near the effusion location, where coolant accelerated and changed direction while approaching the effusion hole. In summary, cautiously observe the laterally averaged results; the local and area average Nu_D data provides a better representation of heat transfer performance.

Figure 68 Laterally Averaged Nusselt Number, Compare Geometries A/A+ (Low Z)

Figure 69 Laterally Averaged Nusselt Number, Compare Geometries A/A+ (High Z)

Another way to view the Geometry A+ data presented in Figure 68 and Figure 69 is through a single plot in Figure 70. This allows for a clearer comparison of the relationship between jet Re_D and Nu_D . Also, it helps to see the effect of Z/D_i on Nu_D for this geometry. High Z/D_i leads to a very uniform heat transfer distribution (a positive quality, from the point of design of a part for an extended life). However, the difference in the magnitude of Nu_D at $Z/D_i = 2.4$ and 10.4 is critical to cooling system design. One could not get similar performance out of a $Z/D_i = 10.4$ geometry and a jet $Re_D = 30000$ that one gets from $Z/D_i = 2.4$ and jet $Re_D = 10000$ – even with three times as much coolant spent. Similarly, Hollworth (1980) identified peak heat transfer performance at a Z/D_i between 2 and 3 for such impingement-effusion cooling systems, regardless of jet spacing.

Figure 70 Geometry A+ High Z / Low Z Laterally Averaged Nusselt Number

Figure 71 Geometry A High Z / Low Z Area Averaged Nusselt Number

For further comparison, average Nusselt numbers for Geometry A are plotted versus Re_D in Figure 71. As previously mentioned, the rate of increase of Nu_D slows with increasing Re_D . The trend may be approximated as a straight line on a log-log plot as shown in Figure 71. The Reynolds number exponents range from 0.65 to 0.71, very similar to that which Martin (1977) suggests for SRN performance throughout this range of Re_D in (4).

As previously mentioned, all remaining geometries (Geometry B through Geometry E as indicated by Table 1) are tested using the same heater configuration and effusion plate as Geometry A. In other words, the only parameters changed from one test to the next are impingement hole diameter, impingement height, and impinging jet Re_D . This allowed for rapid evaluation of a variety of different configurations.

The first of these configurations is Geometry B, with slightly higher impingement array spacing (X_i/D_i and Y_i/D_i) as well as slightly taller impingement height (Z/D_i) than Geometry A.

The local Nu_D profiles for Geometry B configurations can be seen in Figure 72 through Figure 77. The magnitude of Nu_D changes with each geometrical parameter variation, but, in general, the local distribution of Nu_D shares many features with that of Geometry A.

Section 4.2.2 Geometry B, C, D Impingement Heat Transfer

Similarly, the local Nu_D profiles for Geometries C, D, and E share these common characteristics. Therefore, rather than offer individual plots and discussion for each of Geometries B through E, all local data is introduced, and is followed by a general discussion of all impingement heat transfer data. From the comparison of all geometries, general conclusions are drawn.

The local Nu_D profiles for Geometry C are provided in Figure 78 through Figure 83. The local Nu_D profiles for Geometry D are provided in Figure 84 through Figure 89. The local Nu profiles for Geometry E are provided in Figure 90 through Figure 95.

Figure 72 Geometry B (Low Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 73 Geometry B (Low Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 74 Geometry B (Low Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 75 Geometry B (High Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 76 Geometry B (High Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 77 Geometry B (High Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 78 Geometry C (Low Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 79 Geometry C (Low Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 80 Geometry C (Low Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 81 Geometry C (High Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 82 Geometry C (High Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 83 Geometry C (High Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 84 Geometry D (Low Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 85 Geometry D (Low Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 86 Geometry D (Low Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 87 Geometry D (High Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 88 Geometry D (High Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 89 Geometry D (High Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 90 Geometry E (Low Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 91 Geometry E (Low Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 92 Geometry E (Low Z) Impingement Nusselt Number, $Re_D = 30000$

Figure 93 Geometry E (High Z) Impingement Nusselt Number, $Re_D = 10000$

Figure 94 Geometry E (High Z) Impingement Nusselt Number, $Re_D = 20000$

Figure 95 Geometry E (High Z) Impingement Nusselt Number, $Re_D = 30000$

Section 4.2.3 Comparison of Impingement Results for All Geometries

Data from all 5 impingement geometries is reduced to laterally averaged and area averaged Nusselt number results in the following plots, in order to evaluate the differences between the various configurations. Figure 96, Figure 98, and Figure 100 show the laterally averaged Nusselt number profile for all configurations with low impingement height. Each figure represents a different jet Reynolds number. Figure 97, Figure 99, and Figure 101 similarly show these results for the same configurations, but with high impingement height.

In general, for a given Re_D , Geometry A outperforms all others as it has the smallest hole-to-hole spacing and impingement height. Likewise, Geometry E, with the largest X_i/D_i , Y_i/D_i , and Z/D_i , consistently delivers the lowest Nu_D . In most cases, the other geometries fall in order, with Nu_D values in between these two extremes. As was the case with Geometry A, Nu_D generally increases with increasing Re_D and decreases with increasing Z/D_i .

Also noteworthy in the Low Z results is the disappearance of the local maxima in heat transfer between impinging jets. Though evident with Geometry A, this peak is only slightly visible in the laterally averaged result for Geometry B, and then missing as hole-to-hole spacing increases. Again, this is explained by the momentum of the wall jet: going from Geometry A to Geometry E, only the impinging jet diameter decreases. The distance from the impingement stagnation point to the location of wall jet interaction is a constant value of $\frac{1}{2}X_e$ for all geometries. The momentum of the wall jet decreases as the ratio of $\frac{1}{2}X_e$ to D_i (the dimensionless distance from the jet stagnation point) increases, and therefore the colliding wall jets have less momentum for Geometry E than they do for Geometry A.

Figure 96 Laterally Averaged Nusselt Number for All Impingement Geometries, Low Z , $Re_D = 10000$

Figure 97 Laterally Averaged Nusselt Number for All Impingement Geometries, Low Z , $Re_D = 20000$

Figure 98 Laterally Averaged Nusselt Number for All Impingement Geometries, Low Z , $Re_D = 30000$

Figure 99 Laterally Averaged Nusselt Number for All Impingement Geometries, High Z , $Re_D = 10000$

Figure 100 Laterally Averaged Nusselt Number for All Impingement Geometries, High Z, $Re_D = 20000$

Figure 101 Laterally Averaged Nusselt Number for All Impingement Geometries, High Z, $Re_D = 30000$

From a standpoint of heat transfer uniformity, Geometry A produces the “flattest” laterally averaged Nu_D profile at low impingement heights. For tall impingement, all geometries produce relatively “flat” profiles. In order to quantitatively examine heat transfer uniformity, the standard deviation of each profile was calculated and is presented in Figure 102. Standard deviation is a measure of variability in data, and has the same unit of measure as the data being analyzed. Therefore, the results plotted in Figure 102 are unitless, but refer to the magnitude of Nu_D . A standard deviation of 2 indicates that 68% of the measured data falls within ± 2 of the mean Nusselt number.

It is expected that the standard deviation of the High Z data is very low, in observing the profiles in Figure 97, Figure 99, and Figure 101, all of which are relatively flat with barely discernable peaks near the stagnation point. Such a uniform profile is, as discussed previously, created by the fact that the jet has a much wider, lower distribution of velocity by the time it reaches the target surface. Uniformity of the Low Z data, however, is much more variable, and trends in Figure 102 suggest that heat transfer uniformity decreases with increasing Re_D and increasing hole-to-hole spacing. The latter of these is explained in a similar manner to the disappearance of the second heat transfer peak at high hole-to-hole spacing: the relative size of the jet to the target surface causes a wide variability of heat transfer rate, from the stagnation point to the point of neighboring wall jet collision.

Figure 102 Standard Deviation of Laterally Averaged Nusselt Number, Used to Evaluate Heat Transfer Uniformity

As previously shown in Figure 71, the relationship between Nusselt number and Reynolds number is governed by an exponent m . SRN impingement correlations suggest this exponent varies from 0.6 to 0.8. Such a relationship has also been observed in impingement arrays as part of impingement effusion cooling systems, with Hollworth (1980) using a value of 0.8 and Cho (2008) identifying an exponent of 0.68 through correlation. Similarly, an exponent of 0.68 has been derived from the data presented, using a range of jet Re from 10000 to 30000. Some of the average Nu_D results are shown in Figure 103, with this correlation plotted for visual comparison. Regardless of hole-to-hole spacing and impingement height, this relationship between Nu_D and Re_D remains valid.

Figure 103 Effect of Jet Reynolds Number on Nusselt Number for Selected Geometries

These area-averaged Nu_D results could then be compared against the existing impingement array correlations of Florschuetz (1981) and Kercher (1970). Only the Low Z geometries are used, as neither correlation includes impingement heights exhibited by the High Z geometries within its range of validity. As cross flow is taken into consideration for these correlations, the cross flow mass flux was set to zero in their evaluation. This is a good assumption based on the consideration that fluid from each jet is extracted by the four surrounding effusion holes, and it is supported by the relatively uniform Nu_D distributions observed in the local results.

Data for all geometries is compared to the Florschuetz (1981) correlation in Figure 104. There is good agreement among the Geometry A results, but it is found that this correlation underestimates for the geometries with higher Z/D_i . As the data used to generate the correlation

was limited to $1 < Z/D_i < 3$, extrapolation of the correlation does not agree with the measured results.

Figure 104 Impingement Heat Transfer Results Compared With Florschuetz (1981) Correlation

Further comparison with the Kercher (1970) correlation shows better agreement than observed with Florschuetz (1981), particularly at high Re_D . Kercher (1970) used $1 < Z/D_i < 4.8$ and therefore is expected to capture the effect of increasing Z/D_i . Kercher's (1970) graphical correlation technique includes some uncertainty, however, and coefficients are plotted for two discrete ranges of Re_D . In general, the correlation of Kercher (1970) showed higher Nu_D results than that of Florschuetz (1981) for $Z/D_i > 3$; Gao (2003) noted a similar result. It is of the author's opinion that better agreement at intermediate values of Re_D (~10000 in this case) may be achieved if these correlation coefficients were presented as a function of hole spacing and Reynold's number, in a similar manner as Florschuetz (1981).

Figure 105 Impingement Heat Transfer Results Compared With Kercher (1970) Correlation

Next, Nu_D was normalized by the Reynolds number to the exponent of 0.68 to compress data for each geometry toward a single point. Then, all the data that has been presented thus far was plotted versus impingement height on a single graph, seen in Figure 106. As impingement height is a predominate variable of interest for impingement cooling, determining the relationship between Nu_D and Z/D_i in such a system was essential. Using Low Z and High Z configurations (ranging from $Z/D_i = 2.4$ to $Z/D = 23.5$) it is shown that heat transfer is inversely proportional to the square root of nozzle-to-target-plate spacing. This result compliments Hollworth's (1980) data that shows a monotonically decreasing trend in Nu for $Z/D_i > 2$. It also identifies Geometry A as the strongest performer, from a pure heat transfer point of view.

In the case of gas turbine cooling design, impingement cooling systems must not only be designed from a pure heat transfer point of view. Rather, such design requires consideration of heat transfer, aerodynamic losses, and coolant mass flow rate. Hollworth (1980) suggested that the coolant mass flow rate be considered when evaluating the performance of these systems by

introducing a new parameter, Re^* , shown in (41). This parameter is based on the coolant flow per unit area of target surface, G . Therefore, Re^* is proportional to G .

$$Re^* = \frac{4}{\pi} \frac{GD_i}{\mu} = \frac{Re_D}{\left(\frac{X_i}{D_i}\right)\left(\frac{Y_i}{D_i}\right)} \quad (41)$$

Using this definition, the impingement heat transfer results are again plotted in Figure 107. It is immediately evident that Low Z configurations outperform High Z configurations, as previously discussed, as (for a given impingement array geometry) both use an equal amount of coolant, yet Low Z yields higher heat transfer. The more subtle point on Figure 107 is the fact that arrays with large hole-to-hole spacing and higher jet Re_D (such as Geometry E, $Re_D = 30000$) produce superior heat transfer to geometries with small hole-to-hole spacing and low jet Re_D (for example Geometry A, $Re_D = 10000$) while requiring approximately half the coolant flow rate. Depending on the method by which impingement cooling performance is evaluated, it is possible to consider an array such as Geometry E to be more efficient than Geometry A. This conclusion was also arrived to by Hollworth (1980); however, he was quick to note that in a given design problem the source and sink pressure may be fixed, prohibiting the designer from achieving an elevated Re_D . One should also note that this method does not consider the significant aerodynamic losses associated with any impingement cooling system design.

Figure 106 Effect of Impingement Height and Jet Reynolds Number on Impingement Nusselt Number

Figure 107 Average Nusselt Number Plotted Versus Re^*

Section 4.3 Film Cooling Effectiveness

Coolant exiting the effusion holes interacts with the cross flow and generates a cooling film on the hot side of the test section. The temperature of this surface was captured in order to determine the film cooling effectiveness, η , for a variety of effusion geometries and blowing ratios, M . This effusion design was much like a perforated plate, in that the holes were oriented perpendicular to the target surface and the mainstream flow (e.g., $\alpha = 90^\circ$), the hole length-to-diameter ratio (L/D) was less than unity, and the holes were not “shaped” in any way to enhance film cooling performance. A sample of a typical perforated plate (or, “simple drilled sheet”) design is duplicated from Metzger (1973) and shown in Figure 108.

Figure 108 Typical Perforated Plate Effusion Cooling Design, from Metzger (1973)

Due to this, the boundary condition at the effusion hole inlet had a significant impact on the film cooling performance. Hale (2000) showed that for these short, normal (e.g. perpendicular to the mainstream flow) film cooling holes, the direction by which the coolant was fed into the hole greatly affected the resulting film cooling effectiveness. See Figure 109 to examine the different η profiles generated when feeding coolant parallel to the mainstream flow, either “co-flow” (in the x direction) or “counter-flow” (in the $-x$ direction).

Figure 109 Example from Hale (2000) Showing the Effect of Effusion Hole Inlet Boundary Condition

It was shown in the last chapter that the boundary condition downstream – the effusion array – could be changed without significantly altering the impingement heat transfer distribution. However, in the case of effusion performance, the upstream configuration – the impingement array – should be matched up as specified in Table 1, to avoid a significant change in film cooling effectiveness due to a significant change in flow distribution within the impingement-effusion test specimen. All impingement geometries used were in their High Z configuration.

Section 4.3.1 Geometry A Film Cooling Effectiveness

The first impingement-effusion geometry, Geometry A, was designed with two different effusion hole configurations: in-line and staggered. As previously presented in Figure 21, “staggered” configurations are staggered with respect to the mainstream flow direction, so as to fill in the spaces between the effusion holes with injected coolant. The inclusion of both designs for Geometry A allows for direct comparison of in-line and staggered effusion performance.

Local film cooling effectiveness data is provided for Geometry A in Figure 110, Figure 111, and Figure 112 for the in-line configuration; results in Figure 113, Figure 114, and Figure 115 highlight the staggered configuration. In all local data, the mainstream flow direction is from bottom to top in the image. Results are plotted so that dark blue colors indicate uncooled ($\eta = 0$)

regions while oranges and reds indicate highly cooled ($\eta > 0.7$) regions. The effusion holes themselves can be seen in each image; several smaller holes around the border of the image indicate fasteners used in the assembly of the test geometry, and no data was collected at these points.

Figure 110 Geometry A (In-Line) Film Cooling Effectiveness, $M = 0.15$

Figure 111 Geometry A (In-Line) Film Cooling Effectiveness, $M = 0.20$

Figure 112 Geometry A (In-Line) Film Cooling Effectiveness, $M = 0.25$

Figure 113 Geometry A (Staggered) Film Cooling Effectiveness, $M = 0.15$

Figure 114 Geometry A (Staggered) Film Cooling Effectiveness, $M = 0.20$

Figure 115 Geometry A (Staggered) Film Cooling Effectiveness, $M = 0.25$

These results show a similar trend to that examined in single-hole and single-row film cooling data in the literature: when M is low enough that blow-off is insignificant, η is high immediately downstream of the hole, and decreases with distance away from the hole. Cooling effectiveness also gradually spreads laterally, both because the turbulence generated by the jet and cross flow interaction encourages mixing of the coolant and hot gas, and also because heat is conducting laterally in the boundary layer. Typically, the η profile at each hole was symmetrical and similar to that of the other jets in a given row. However, in a few cases (often near the edge of the impingement-effusion system) abnormal behavior was experienced. It is conjectured that the flow distribution between the impingement and effusion arrays yielded a non-uniform supply of coolant to all effusion holes. (Each test specimen was examined thoroughly to ensure that no blockages in the holes led to this behavior.) Also, some holes were fed with via an impingement wall jet that had enough momentum in the y -direction that it affected the angle of the coolant jet. Most notably, the two holes opposite each other in the first row of the in-line geometry (see

Figure 111) were an example of the former; the holes at the outside of the third row in the staggered configuration (see Figure 114) are an example of the latter. This conjecture is supported by: (a) the study by Hale (2000) that shows the coolant hole inlet boundary conditions affect the resulting η profile; (b) the observation by Hollworth (1984) in local impingement Nu_D results that identified an uneven distribution of coolant on the target surface, governed by the placement of the effusion holes; and (c) the fact that these abnormalities were observed at all blowing ratios, and were symmetrical with respect to $y = 0$.

It can immediately be seen from the local η results that the staggered configuration produced overall higher cooling effectiveness than the in-line configuration. The most significant difference between the two (at a given M) is found downstream of the last two rows of effusion. The increased magnitude of η , however, is not the only advantage held by the staggered configuration. As seen in Figure 116, the lateral distribution of η is much more even in the case of staggered effusion. This is a result of the staggered arrangement forcing coolant to “fill in the gaps” between effusion holes in each row and create a profile more likened to the idealized case of slot cooling. Immediately downstream of the 2nd Row, peak values of η are slightly higher for the in-line arrangement because cool fluid from two rows has compounded at this point. However, the “valleys” in between injection locations exhibit very low values of η . Such lateral uniformity is important in gas turbine film cooling design, as a difference in η of 0.2 to 0.3 often corresponds to a difference in T_{aw} of several hundred degrees Kelvin. Therefore, significant thermal stresses may be created in the cooled metal surface by uneven temperature distributions caused by severe non-uniformities in η . Downstream of the 5th (and last) Row, the staggered profile is slightly more uniform, and the magnitude of η is consistently higher by 0.15.

Figure 116 Lateral Film Cooling Effectiveness Profile, In-line and Staggered

In order to show the development of η with each additional row of injection, η was laterally averaged and plotted versus the x (streamwise) direction. (Data at each row was removed, as the averaging area changes and produces a misleading result.) All laterally averaged data for Geometry A is presented in Figure 117. The location of each effusion row is identified by a black circle on the x -axis.

The following observations were made from this graph: (a) varying blowing ratio (M) does not have a significant effect on η over the range tested (ignoring the first row behavior, which appears highly variable); (b) on the lateral average, staggered and in-line geometries produce a similar result downstream of the first three rows of injection; (c) after the third row, there are no significant increases in η with additional rows of injection for the in-line arrangement (such a condition is called “periodically fully developed” to indicate that the flow field is replicated on a periodic basis); and (d) elevated η may be experienced well downstream of the last row of injection.

Figure 117 Laterally Averaged Film Cooling Effectiveness, Geometry A

Each observation may be logically explained. For (a), the range of blowing ratios tested was relatively small, well within the range where jet blow-off should not have a significant effect on performance. Harrington (2001) and Cho (1995) witness no significant blow-off effects in $\alpha=90^\circ$ injection for $M < 0.35$. Furthermore, both studies identify that higher values of M not only require more coolant but also lead to lower η because of jet blow-off; the range tested is therefore more optimized for cooling performance. One must also consider that M is directly related to impinging jet Re_D in such a system, both flow parameters must be considered in design optimization. For (b), it was shown in Figure 116 that in-line arrangements lead to higher η than the staggered configuration immediately downstream of the 2nd row, but lower η at “valleys” in between effusion holes. Therefore, these values average out to the result that in-line and staggered configuration perform equally at the 2nd and 3rd rows, before the periodically fully developed condition hampers continued growth of the in-line laterally averaged η profile. For

(c), the periodically fully developed condition may be envisioned as a point where the amount of enthalpy being injected is equal to the amount of enthalpy being directed away from the wall through turbulent mixing and thermal conduction. This point is therefore reached more quickly in the case of in-line injection as each jet interacts with the jet immediately upstream; in the staggered case, each jet interacts most significantly with the jet two rows upstream. This behavior was also observed by Metzger (1973) in the comparison of in-line and staggered configurations; it takes approximately twice as many rows of staggered injection than in-line injection to reach the periodically fully developed state. For (d), this result is hardly surprising, as a large amount of coolant has been injected over a small surface at a blowing rate that allows the film to stay attached to the wall. A similar trend was observed by Mayle (1975) and Crawford (1980) in multi-row film cooling scenarios.

Section 4.3.2 Geometry E/F Film Cooling Effectiveness

Next, local η data for Geometry E is presented in Figure 118, Figure 119, and Figure 120. Geometry E used a staggered hole configuration, with larger hole-to-hole spacing X_e/D_e and Y_e/D_e . For a given blowing ratio, this meant that the same surface area must be cooled with a smaller amount of coolant. Accordingly, local distributions show a lower magnitude of η than was found with Geometry A. Also, uncooled or poorly cooled regions between holes are larger for Geometry E.

Figure 118 Geometry E Film Cooling Effectiveness, $M = 0.16$

Figure 119 Geometry E Film Cooling Effectiveness, $M = 0.22$

Figure 120 Geometry E Film Cooling Effectiveness, $M = 0.28$

In the following Figure 121, Figure 122, and Figure 123, film cooling effectiveness results for Geometry F are provided. This geometry uses an in-line configuration and the largest X_e/D_e and Y_e/D_e of all tested geometries. The poor performance of Geometry F, from a film cooling standpoint, is immediately obvious in comparing these results with the others. The lateral distribution of coolant is very poor, and uncooled portions remain in between the holes at the 5th row of injection. (NOTE: Results for the second to last column of holes, to the right of the test section, are invalid due to a shadowing issue in the test data. The entire test surface has been shown for completeness, but data on the right side in this region has been obstructed from view, and is not used for averaging.)

Figure 121 Geometry F Film Cooling Effectiveness, $M = 0.18$

Figure 122 Geometry F Film Cooling Effectiveness, $M = 0.24$

Figure 123 Geometry F Film Cooling Effectiveness, $M = 0.30$

The resulting laterally averaged η profiles for Geometries E and F are provided in Figure 142, below. Rather than the streamwise variable x being normalized by the hole diameter D , it was normalized by the streamwise hole spacing X_e . This ensures that the axis is scaled the same for either geometry (the hole diameter changes between geometries, while the hole-to-hole spacing remains the same). It is clear that higher η is achieved with Geometry E from the 1st row, as Y_e/D_e is smaller and hence the coolant coverage is greater for this configuration. Also, the magnitude of η grows more rapidly, as seen for the staggered version of Geometry A. Geometry F, though in-line, does not reach a periodically fully developed condition because the region between the holes was not filled in by the 5th row of injection. Rather, it is expected that this condition would be met after 8 to 10 rows of injection. It is worth noting that the η profile immediately downstream of the effusion hole is identical at all rows; in this region, a periodically fully developed condition already exists. It is apparent that the $\alpha = 90^\circ$, cylindrical hole film

cooling design is not conducive to jet spreading, a feature that is most commonly associated with shaped film cooling holes.

Section 4.3.3 Film Cooling Effectiveness, FC Geometry 1

Film cooling effectiveness results for FC Geometry 1 are provided in Figure 124, Figure 125, Figure 126, and Figure 127. This geometry showcased compound angle injection, leading to the lateral coolant trajectory seen in the local data. Despite a large variation in blowing ratio, the magnitude of film cooling effectiveness over the first half of the coolant array was comparable for all blowing ratios. This can also be examined in the laterally averaged data in Figure 128, as all except for $M = 2.0$ are within $\eta = 0.03$ throughout this region. Lower η witnessed for $M = 2.0$ is attributed to jet blow-off at such high blowing ratios. In fact, there is a clear trend between increasing M between 0.5 and 2.0 and decreasing η in this region. A similar trend was observed in the single row, compound angle film cooling effectiveness data of Goldstein (2001).

In addition to the entire film array, data was collected over recovery region 100 diameters in length. In this region it was found that film cooling performance was highly dependant on blowing ratio. At high M the significant lateral momentum of the coolant jets leads to an uneven distribution of coolant in the recovery region. At $M = 2.0$ the neighboring film cooling rows merge and enhance lateral non-uniformity. Furthermore, downstream of the holes located at $y/D < -50$, η decreases at a much faster rate for high M than low M because the coolant appears to be swept away from this region by its own momentum. The distribution of η with y is presented in Figure 129 for $x/D = 200$ and Figure 130 for $x/D = 250$. In these plots, the “sweeping” of coolant in the positive y -direction is evident as M increases, and it can be seen that this behavior is most pronounced downstream.

Figure 124 FC Geometry 1 Film Cooling Effectiveness, $M = 0.5$

Figure 125 FC Geometry 1 Film Cooling Effectiveness, $M = 1.0$

Figure 126 FC Geometry 1 Film Cooling Effectiveness, $M = 1.5$

Figure 127 FC Geometry 1 Film Cooling Effectiveness, $M = 2.0$

Figure 128 FC Geometry 1 Laterally Averaged Film Cooling Effectiveness

Figure 129 FC Geometry 1 Recovery Region Effectiveness, $x/D = 200$

Figure 130 FC Geometry 1 Recovery Region Effectiveness, $x/D = 250$

The geometry of FC Geometry 1 was modeled after a previous full coverage film cooling study performed by Mayle & Camarata (1975). One of the geometries studied by Mayle & Camarata is identical to FC Geometry 1, as shown by Table 6. This allows for additional validation of the experimental methodology while adding local detail to the effectiveness measurements so as to explain certain flow phenomena. As can be seen in Figure 131, Figure 132, Figure 133, and Figure 134, laterally averaged η for FC Geometry 1 closely follows that which was determined by Mayle & Camarata. One notable point where the two sets of data differ is in the recovery region for $M = 1.5$. Here, Mayle & Camarata reported a nearly flat (rather than gradually decreasing) trend of η with x . This trend was not seen for any other geometry or blowing ratio, and no further explanation was offered. Given the consistency among the other comparisons of FC Geometry 1 and Mayle & Camarata in the recovery region as well as the trend of η witnessed with increasing M , it should be expected that η will decrease in the recovery region regardless of M for this full coverage film cooling configuration.

Table 6 Full Coverage Film Cooling Test Matrix, Mayle & Camarata vs. FC Geometry 1

FC Geometry	x_e/D_e	y_e/D_e	M	Arrangement	# Rows	α	β
Mayle P/D = 8	6.9	8.0	0.50 1.0 1.5 2.0	Staggered	26	30°	45°
Mayle P/D = 10	8.7	10.0	0.50 1.0 1.5 2.0	Staggered	21	30°	45°
Mayle P/D = 14	12.1	14.0	0.50 1.0 1.5 2.0	Staggered	15	30°	45°
1	12.1	14.0	0.50 1.0 1.5 2.0	Staggered	15	30°	45°

Figure 131 FC Geometry 1 vs. Mayle & Camarata, M = 0.5

Figure 132 FC Geometry 1 vs. Mayle & Camarata, $M = 1.0$

Figure 133 FC Geometry 1 vs. Mayle & Camarata, $M = 1.5$

Figure 134 FC Geometry 1 vs. Mayle & Camarata, M = 2.0

Section 4.3.4 Film Cooling Effectiveness, FC Geometry 2

FC Geometry 2 was identical to FC Geometry 1 except for the compound angle β , which is set to zero. To enable direct comparison, blowing ratios of $M = 0.5$, 1.0 , 1.5 , and 2.0 were studied. Due to trends seen in the results, two additional blowing ratios of $M = 0.25$ and $M = 0.75$ were also included, so as to determine an “optimum” blowing ratio that leads to the highest η overall. The local film cooling effectiveness results are presented in Figure 135, Figure 136, Figure 137, Figure 138, Figure 139, and Figure 140. Note that the scale is identical to that of FC Geometry 1 so that direct comparison of the contours may be made. Performing this comparison, it is immediately obvious that the maximum η is greater for FC Geometry 1 than for FC Geometry 2, over the tested range of M .

Figure 135 FC Geometry 2 Film Cooling Effectiveness, $M = 0.25$

Figure 136 FC Geometry 2 Film Cooling Effectiveness, $M = 0.5$

Figure 137 FC Geometry 2 Film Cooling Effectiveness, $M = 0.75$

Figure 138 FC Geometry 2 Film Cooling Effectiveness, $M = 1.0$

Figure 139 FC Geometry 2 Film Cooling Effectiveness, $M = 1.5$

Figure 140 FC Geometry 2 Film Cooling Effectiveness, $M = 2.0$

The laterally averaged cooling effectiveness for FC Geometry 2 at all blowing ratios is provided in Figure 141. This allows for careful examination of the effect of blowing ratio in both the injection and recovery regions. It is immediately apparent that $M = 0.5$ provides the highest η throughout the film array; here, higher blowing ratios lead to enhanced jet blow-off and subsequently lower η . However, after several rows of injection, so much coolant has been forced

into the boundary layer that η must increase, either as a result of coolant reattachment or simply thermal diffusion. Therefore, the rate of increase of η in the streamwise direction becomes greater as M increases.

In the recovery region, $M \leq 0.75$ yielded decreasing η while $M \geq 1.0$ yielded level or increasing η . Due to this, near $x/D = 275$, the highest η was seen at the highest M , the second highest η at the second highest M , and so on. (This however does not suggest that the highest M is necessary for the best overall film cooling performance; as observed in Figure 141, it is in fact quite the opposite. More discussion on the “optimal” blowing ratio is offered in a later section.)

Figure 141 FC Geometry 2 Laterally Averaged Film Cooling Effectiveness

Section 4.3.5 Film Cooling Effectiveness, Impingement-Effusion Geometries

Figure 142 Laterally Averaged Film Cooling Effectiveness, Geometries E and F

In order to appreciate the differences in the magnitude of η for the various geometries, laterally averaged data for each geometry at a single blowing ratio is combined in Figure 143. Again, all results are plotted on a scale that allows the location of each row to overlap. At the first row, the relationship between coverage and η is immediately apparent. “Coverage” is frequently used in film cooling to refer to the ratio of open area at a row of film cooling holes to the total area being cooled. For unshaped cylindrical holes, the coverage is simply the inverse of the hole-to-hole spacing (also called pitch), Y_e/D_e . The physical limit for laterally averaged film cooling effectiveness immediately downstream of the first row is equal to the coverage, because $\eta \leq 1$ immediately downstream of the hole’s open area and $\eta = 0$ elsewhere. The first row result for each geometry is very close to this limit: 0.5 for Geometry A, 0.25 for Geometry E, and 0.22

for Geometry F. Considering this initial value, one can understand how geometries with different hole spacings X_e/D_e and Y_e/D_e can produce drastically different η results.

Figure 143 Laterally Averaged Film Cooling Effectiveness, All Geometries

Using the laterally averaged η , which was determined using only the data from the center three rows to best simulate an infinitely wide array, a single “area averaged” η value was determined for each geometry. This allows quick comparison of the effect of geometrical and flow parameters on the magnitude of η . First, the area averaged results were studied with respect to blowing ratio (M), shown in Figure 144. As mentioned previously, over the small range of M studied, there was not a significant effect observed, except for a slight increase in η with M (attributed to the fact that more coolant was being used). This plot again allows comparison of the respective magnitude of η for each geometry.

Next, all results were compared versus streamwise spacing, X_e/D_e as shown in Figure 145. There was a clear advantage to staggered configurations over in-line in terms of both η

magnitude and lateral uniformity. This figure captures only the former, but shows a clear difference of approximately $\eta = 0.1$ associated with staggered configurations. (Note that all geometries had Y_e/D_e scale with X_e/D_e , so this plot represents not only streamwise spacing, but also spanwise spacing. Also note that values of $X_e/D_e \leq 1$ are unrealistic.)

Figure 144 Area Averaged Film Cooling Effectiveness Versus Blowing Ratio

Figure 145 Area Averaged Film Cooling Effectiveness Versus Row Spacing

Section 4.3.6 Film Cooling Effectiveness, Film Cooling Geometries

As FC Geometry 1 and FC Geometry 2 differ by only one geometrical parameter, β , a direct comparison of these geometries enables better understanding of the impact of compounding angle in full coverage arrangements. A comparison of laterally averaged effectiveness data at $M = 0.5, 1.0, 1.5$, and 2.0 is provided in Figure 146, Figure 147, Figure 148, and Figure 149, respectively. As previously observed, η is much greater for FC Geometry 1, especially at higher blowing ratios. This geometry sees a plateau in increasing η with blowing ratio at approximately $M = 1.0$, while FC Geometry 2 sees declining performance for $M > 0.5$. As can also be observed in these figures, the rate of increase of laterally averaged η with each row of injection is approximately doubled when $\beta = 45^\circ$. It is important to note that, at high M , η declines in the recovery region for FC Geometry 1 and increases for FC Geometry 2. It is reasonable to expect that, assuming the majority of the injected enthalpy remains in the boundary

layer, the adiabatic wall temperature far downstream will be nearly equivalent regardless of compounding angle β . This is typically captured (for single row configurations) via X/MS plotting, where all curves collapse to a single line far downstream (see Baldauf (2002) for an example). Laterally averaged η for FC Geometries 1 and 2 will likely merge downstream.

Figure 146 Compare FC Geometry Film Cooling Effectiveness, $M = 0.5$

Figure 147 Compare FC Geometry Film Cooling Effectiveness, $M = 1.0$

Figure 148 Compare FC Geometry Film Cooling Effectiveness, $M = 1.5$

Figure 149 Compare FC Geometry Film Cooling Effectiveness, $M = 2.0$

Section 4.3.7 Film Cooling Effectiveness Summary

To enable a brief comparison of impingement-effusion array geometries and full coverage film cooling geometries, area averaged η taken over the injection region is provided in Figure 150. With all geometries exhibiting different hole spacing, it is worthwhile to consider not only the mass flux of coolant introduced (i.e. via the blowing ratio) but also the total mass flow rate of coolant used in cooling the injection region of size A_{surf} . This consideration is important from a design perspective because as the total mass flow of coolant is a design parameter that must be controlled; a large mass flow rate in the secondary air system results in thermodynamic losses and decreased overall cycle performance. In short, the blowing ratio M is multiplied by the ratio of the flow area to the injection region surface area, where $A_{\text{hole}}/A_{\text{surf}}$ is determined via (42).

$$A_{hole} / A_{surf} = \pi/4 \left(\frac{X}{D} \right)^{-1} \left(\frac{Y}{D} \right)^{-1} \quad (42)$$

The resulting comparison is provided in Figure 150. Clearly, the more tightly-spaced geometries yield higher η but require considerably more coolant to do so. As expressed previously, staggered arrangements provide an advantage over in-line. In the injection region, it appears that normal injection at low M (impingement-effusion geometries) is more efficient than angled injection at a higher M (full coverage film cooling geometries). This plot does not account for recovery region effectiveness, however, where smaller inclination angles provide a significant advantage. Baldauf (2002), for example, shows that as α decreases, near-hole η decreases but far-field η increases, and recovery region performance is significantly better at high M when comparing normal and angled injection. All things considered, more extensive recovery region film cooling data would have to be available for the impingement-effusion geometries to assist in making a full evaluation. However, in the injection region, it is concluded that a staggered arrangement with large hole-to-hole spacing as well as normal injection at low M (i.e. Geometry E) is most desirable, if coolant flow rate is a primary concern.

Figure 150 Area Averaged Effectiveness for All Geometries

Section 4.3.8 Film Cooling Effectiveness Superposition

In the past, several researchers studying multi-row (i.e. full coverage) film cooling have made efforts to compare full coverage results with an analytical prediction based on single-row film cooling data. The development of such a film cooling superposition method would allow for better understanding of the flow physics associated with multi-row injection and help in optimizing full coverage film cooling design. The additive film superposition method, first proposed and demonstrated by Sellers (1963), has been used (with varied success) in the works of Metzger (1973), Mayle (1975), Sasaki (1979), and Harrington (2001) for comparison with full coverage film cooling effectiveness.

The additive film method essentially assumes linear superposition of the η profile with each row of injection, except that the mainstream temperature used to determine η is replaced by the adiabatic wall temperature resulting from upstream injection. The resulting formula (as developed by Mayle, 1975) for laterally averaged η is shown in (43, where n stands for the

number of rows of injection upstream of the streamwise coordinate, x . In (43), η_r is a function expressing the laterally averaged effectiveness of a single row of holes. (Note that X_e represents the streamwise spacing (a constant) while x represents the independent variable.)

$$\eta = 1 - \exp \left\{ \sum_{k=1}^n \ln \left(1 - \eta_r \left[\frac{x}{D} - \left((k-1) \frac{X_e}{D} \right) \right] \right) \right\} \quad (43)$$

Typically η_r is determined experimentally or extracted from single-row film cooling literature. A film cooling correlation may also be used, for the sake of simplicity, to evaluate the performance of the additive film method. Key to this method's performance is the prediction of effectiveness decay in the recovery region. Therefore, a common correlation method used in both literature and industry is selected to predict the η profile based on first-row full coverage data. When the streamwise coordinate x is non-dimensionalized by the blowing ratio and hole spacing, as shown in (44), families of curves for different blowing ratios collapse for large ξ (Baldauf, 2002). The correlation form shown in (45) was identified by Bunker (2005) and is used for its relative simplicity, with only a single undetermined constant. (Use of different correlation forms will undoubtedly lead to varying results, so the conclusions made in the following apply specifically to this formula.) A non-linear regression was performed on the first row data to determine the constant C , whose value is presented for each case in Table 7.

$$\xi = \frac{x}{MX_e} \quad (44)$$

$$\eta = \frac{C}{1 + \xi^{0.8}} \quad (45)$$

Table 7 Correlation Constant for Superposition Prediction

	Geometry A	Geometry E	Geometry F	FC Geometry 1	FC Geometry 2
C	1.61	0.695	0.424	0.0696	0.0391

Using this methodology, results have been plotted for each geometry at a single blowing ratio. First, the result for Geometry A is presented in Figure 151, with Geometry D and E following in Figure 152 and Figure 153. Data for both FC Geometries is available in Figure 154 and Figure 155. When viewing, keep in mind the limitations of the X over MS correlation method, which is accurate in the far-field (recovery) region but over-estimates near the hole, where coolant jet behavior dominates (Baldauf, 2002).

Figure 151 Film Cooling Effectiveness Superposition, Geometry A

Geometry A alone allows comparison of the additive film method applied to both staggered and in-line configurations. Downstream, at the 4th and 5th rows of injection, the superposition prediction over-estimates η for the in-line hole arrangement and under-estimates for the staggered arrangement. This trend is repeated in Geometry D and Geometry E results. Metzger (1973) also reported over-prediction by the additive film method for in-line effusion, but found it to be an accurate prediction for staggered arrangements. However, Metzger referenced a study using $\alpha=35^\circ$ for single row data and used it to compare with his $\alpha=90^\circ$ full

coverage array. The reduced injection angle leads to better jet attachment and coverage downstream, and thus an over-prediction of single-row performance in the far-field. It is reasonable to assume that Metzger's conclusions would have been very similar to those presented in this study if the single row data had been measured for $\alpha=90^\circ$. Regardless, the results show an adequate prediction of the average of in-line and staggered configuration η for as many as 5 rows of effusion.

Figure 152 Film Cooling Effectiveness Superposition, Geometry E

Figure 153 Film Cooling Effectiveness Superposition, Geometry F

With respect to inclined and compound angle injection results for FC Geometries 1 and 2, the additive film method significantly under-predicts η in the recovery region. While there is acceptable agreement following the first few rows of injection, the growth of film cooling effectiveness over the latter two-thirds of the array is not captured. Furthermore, it is unlikely that this method would be accurate at higher M , as first row effectiveness (off of which the single row prediction is based) is considerably lower due to jet blow-off, yet η is equal to or higher than the $M = 0.5$ value by the last row. In short, it can be concluded that this method (with the use of the chosen X over MS correlation) will likely under-predict for staggered arrangements. The discrepancy will grow with additional rows of injection. Likely, the best solution to this issue is to: (a) use experimental single-row data with the additive film method; or (b) use an analytical prediction for the single-row prediction, with coolant injection simulated as a point heat sink. The latter of these methods was claimed to be relatively accurate by both Mayle (1975) and Sasaki (1973) at certain blowing ratios.

Figure 154 Film Cooling Effectiveness Superposition, FC Geometry 1

Figure 155 Film Cooling Effectiveness Superposition, FC Geometry 2

Section 4.4 Film Cooling Heat Transfer

The third and final heat transfer experiment performed on the impingement-effusion cooling system determined the rate of heat transfer associated with film injection on a flat plate in a cross flow. If the results are to be non-dimensionalized (a typical goal of fluid mechanics experiments), the knowledge of the heat transfer coefficient in the presence of film as well as the heat transfer coefficient for the uncooled surface is required. The ratio of the two can be taken, and the result is a dimensionless quantity often referred to as the “heat transfer enhancement” associated with the injection of film.

As straight, constant-width heaters were used, data was collected between spanwise rows of holes. When possible, an additional heater was placed in the recovery region to investigate heat transfer enhancement downstream. Trends observed in the local data include a significant increase in h_f/h_0 through the first three rows of coolant injection, both for staggered and in-line configurations. Heat transfer coefficients were doubled (and sometimes nearly tripled) in the region immediately downstream of coolant injection. The interaction of a normally injected fluid jet in a cross flow leads to the development of vortical structures that significantly enhance heat transfer. See Figure 156 for the identification of these structures. This figure shows the horseshoe vortex leading to a “horseshoe”-shaped pattern of high h_f/h_0 immediately downstream of each hole; it also identifies the vortex pair in the jet itself that is responsible for hot gas entrainment and subsequent hampering of film cooling performance.

Figure 156 Vortical Structures Associated with a Jet in a Cross Flow (Margason, 1993)

Figure 157 Typical Heat Transfer Enhancement Profile for Inclined-Hole Injection (Gritsch, 1999)

Section 4.4.1 Film Cooling Heat Transfer, Geometry A

Local results for Geometry A are provided in Figure 158, Figure 159, and Figure 160 (for the in-line hole configuration), as well as Figure 162, Figure 163, and Figure 164 (for the staggered hole configuration). Laterally averaged heat transfer enhancement is presented in Figure 161 (in-line) and Figure 165 (staggered). Note the similarities between the local h_f/h_0 profiles and similar data from Gritsch (1999) in Figure 158. Gritsch's local data is for inclined hole injection ($\alpha = 30^\circ$), which typically generates lower h_f/h_0 than $\alpha = 90^\circ$ per Baldauf (2002).

Geometry A (In-line) Film Cooling Heat Transfer Enhancement, $M = 0.15$

Figure 158 Geometry A (In-line) Film Cooling Heat Transfer Enhancement, $M = 0.15$

Geometry A (In-line) Film Cooling Heat Transfer Enhancement, $M = 0.20$

Figure 159 Geometry A (In-line) Film Cooling Heat Transfer Enhancement, $M = 0.20$

Figure 160 Geometry A (In-line) Film Cooling Heat Transfer Enhancement, $M = 0.25$

The laterally averaged result for each blowing ratio shows an increase in h_f/h_0 with increasing M . It can be concluded that the increased coolant jet momentum leads to a better-developed horseshoe vortex structure and increased turbulence generation upon the jet and cross flow interaction. Heat transfer enhancement drops off quickly after the first row of injection, but remains pronounced as it enters the periodically fully developed region. Vortical structures dominate the flow regime near the cooled surface in this region. A few hole diameters downstream of the last row of injection, h_f/h_0 has fallen off considerably and would likely continue to fall as the increased turbulent kinetic energy dissipates throughout the boundary layer.

The local results from the staggered arrangement of Geometry A show a slightly different characteristic in the region of high heat transfer enhancement associated with the horseshoe vortex: rather than remaining at a near-constant width downstream of each hole, the horseshoe appears to converge in anticipation of the next downstream interaction. This does not have a

significant effect on the magnitude of h_f/h_0 , although this flow configuration does require more rows of coolant injection to reach a periodically fully developed condition. By the last row of injection, the laterally averaged heat transfer rate is approximately double that of the uncooled surface.

Figure 161 Geometry A (In-line) Laterally Averaged Heat Transfer Enhancement

Figure 162 Geometry A (Staggered) Film Cooling Heat Transfer Enhancement, $M = 0.15$

Figure 163 Geometry A (Staggered) Film Cooling Heat Transfer Enhancement, $M = 0.20$

Figure 164 Geometry A (Staggered) Film Cooling Heat Transfer Enhancement, $M = 0.25$

Figure 165 Geometry A (Staggered) Laterally Averaged Heat Transfer Enhancement

Section 4.4.2 Film Cooling Heat Transfer, Geometry E/F

Local heat transfer enhancement data for Geometries E and F is provided in Figure 166, Figure 167, and Figure 168, as well as Figure 170, Figure 171, and Figure 172, respectively. Laterally averaged results are provided in Figure 169 and Figure 173. More of the cooled surface was unaffected by the vortices associated with coolant injection, as the hole spacing on Geometries E and F was higher than Geometry A. The horseshoe vortex was less distinguishable in this case, but the same small region of low h_f/h_0 was observed immediately downstream of injection, indicating such vortices were present. Heat transfer rates for these geometries were approximately 1 to 1.5 that of the uncooled configurations. (Note: Data to the left and right edges of the effusion array was often affected by heater detachment near the bus bars, and only data from the center three rows was used when taking the lateral average.)

Figure 166 Geometry E Film Cooling Heat Transfer Enhancement, $M = 0.16$

Figure 167 Geometry E Film Cooling Heat Transfer Enhancement, $M = 0.22$

Figure 168 Geometry E Film Cooling Heat Transfer Enhancement, $M = 0.28$

Figure 169 Geometry E Laterally Averaged Heat Transfer Enhancement

Figure 170 Geometry F Film Cooling Heat Transfer Enhancement, $M = 0.18$

Figure 171 Geometry F Film Cooling Heat Transfer Enhancement, $M = 0.24$

Figure 172 Geometry F Film Cooling Heat Transfer Enhancement, $M = 0.30$

Figure 173 Geometry F Laterally Averaged Heat Transfer Enhancement

Section 4.4.3 Film Cooling Heat Transfer, Impingement-Effusion Geometries

To quickly evaluate the difference between h_f/h_0 for each of the tested effusion geometries, data at a single blowing ratio was plotted on a common graph for each. This graph is presented in Figure 174. Rather than use the full laterally averaged results, which are local in the x-direction, a single point was taken to represent heat transfer at each heater. A logarithmic trendline was drawn through the data to show the general trend of a quick rise with diminishing increases beyond the 3rd and 4th rows of injection. It is clear from this comparison that Geometry A, with the lowest X_e/D_e and Y_e/D_e of all geometries, maintained significant values of h_f/h_0 . This elevated heat transfer rate is critical to the designer, who must evaluate the advantage of the coolant film on adiabatic wall temperature while considering the adverse effect of an increased heat transfer coefficient. Another interesting note is that the difference between the staggered an

in-line configuration was relatively minimal for heat transfer enhancement. This is primarily due to the fact that the most significant enhancement occurs immediately downstream of injection, and the structures generated dissipate quickly downstream. Therefore it does not necessarily matter how the holes from one row to the next are oriented with respect to each other; the pitch (Y_e/D_e) and blowing ratio (M) appear to be the primary drivers of heat transfer enhancement.

Figure 174 Heat Transfer Enhancement for All Effusion Geometries

Section 4.4.3 Film Cooling Heat Transfer, FC Geometries

Film cooling heat transfer data was also acquired for full coverage film cooling geometries in both the injection and recovery regions. Local results for FC Geometry 1 $M = 0.5$, 1.0, 1.5, and 2.0 are provided in Figure 175, Figure 176, Figure 177, and Figure 178, respectively. Similar to the result seen for the impingement-effusion cooling system, the highest heat transfer enhancement occurs immediately downstream of the film cooling hole, and the

magnitude of h_f/h_0 builds with additional rows of injection. However, after the first three to five rows of injection, there does not seem to be a significant increase associated with additional rows of injection – h_f/h_0 reaches a periodically fully developed condition. In the recovery region, h_f/h_0 drops off dramatically, and approaches unity in the case of low blowing. This suggests that, when a relatively small amount of coolant is injected, its effect on boundary layer dynamics is predominately localized. At high blowing (for example $M = 2.0$), augmentation of heat transfer is exhibited well downstream. The large amount of coolant injected into the boundary layer, as well as the jet trajectory (which approaches the angle of injection as M increases), leads to an effect on heat transfer far downstream in the recovery region. A similar result was obtained by Crawford (1980) when studying film cooling arrangements with 6 to 11 rows of injection. In the local data it may also be observed that the compound angled injection causes the pattern of heat transfer enhancement to “drift” laterally with the coolant. Mayle (1975) noted a similar result when using $\beta = 45^\circ$ (though he claimed it to predominately affect geometries with a lower hole pitch). This lateral drift becomes more pronounced as M increases, a trend also seen in the film cooling effectiveness data.

Figure 175 FC Geometry 1 Heat Transfer Enhancement, $M = 0.5$

Figure 176 FC Geometry 1 Heat Transfer Enhancement, $M = 1.0$

Figure 177 FC Geometry 1 Heat Transfer Enhancement, $M = 1.5$

Figure 178 FC Geometry 1 Heat Transfer Enhancement, $M = 2.0$

To compare the magnitude of h_f/h_0 at various blowing ratios, laterally averaged enhancement is plotted in Figure 179. It can be seen that h_f/h_0 is greater than unity at all M , indicating an increased average surface heat flux as a result of film injection. (It should be noted that such an increase is undesirable as it corresponds to an increase in wall temperature.) As blowing ratio increases up to $M = 1.5$, h_f/h_0 follows suit. $M = 2.0$ shows slightly lower h_f/h_0 , but

as results for all blowing ratios are within the experimental uncertainty, no definite conclusion can be made as to the reasoning behind this result. The trends in the results also resemble those identified by Crawford (1980), in that the lowest enhancement occurs near $M = 0.5$, while the rate of increase with blowing ratio is diminished as M approaches 2.

Figure 179 FC Geometry 1 Laterally Averaged Heat Transfer Enhancement

Clearly, the locations of high heat transfer augmentation correspond to those with high effectiveness. To make a useful comparison of various geometries and blowing ratios it is necessary to combine the effect of both η and h_f/h_0 into a single factor. The goal of cooling is to reduce the wall temperature as compared to the uncooled configuration, which essentially requires a reduction in the heat flux through the wall. Accordingly, a parameter known as net heat flux reduction, Θ , has been introduced in film cooling research, for example Baldauf (2002). This parameter is calculated per (46), with the nomenclature described in Figure 180.

$$\Theta = 1 - \frac{q_w}{q_0} = 1 - \frac{h_{f,tot}}{h_{0,tit}} \frac{(T_{AW} - T_C)}{(T_G - T_C)} = 1 - \frac{h_f(h_w - h_0)}{h_0(h_w + h_f)}(1 - \eta) \quad (46)$$

Figure 180 Nomenclature Used for Θ Derivation, from Baldauf (2002)

The problem with this method remains that the quantity h_W is unknown, as it will be dependant on the material properties, structural design, and backside heat transfer condition. Essentially, a value of h_W must be chosen for comparison of the FC Geometry 1 and FC Geometry 2 data, and the value should be comparable to a realistic film cooling design. Using the simplification that h_W is proportional to h_0 by a factor ζ , the result of (46) is simplified as (47).

$$\Theta = 1 - \frac{(1 + \zeta)(h_f / h_0)}{\zeta + (h_f / h_0)}(1 - \eta) \quad (47)$$

The proportionality factor ζ is chosen to be 1, as was done in Baldauf (2002), citing typical gas turbine film cooling designs. The other factors of heat transfer enhancement and effectiveness are known. Using this, Θ was plotted for FC Geometry 1 at all M in Figure 181. This data may be interpreted as an absolute comparison of film cooling performance at various M for this geometry. High M leads to high heat transfer enhancement, but only marginally better film cooling effectiveness, due to jet blow-off. This combination leads to negative Θ at the first several rows of injection, indicating the use of film cooling in this configuration would actually make the cooled surface hotter than it would have been without film injection. Baldauf (2002) noted the same trend for inclined injection at high blowing rates. However, with the reattachment

of the film in the recovery region, high Θ is maintained well downstream of the last row of injection.

Figure 181 FC Geometry 1 Laterally Averaged Heat Flux Reduction

As FC Geometry 1 is identical to the full coverage array studied by Mayle & Camarata (1975), a comparison of heat transfer enhancement among the two allows for additional validation. Published data was for $M = 0.5$, 1.0 , and 1.5 only, and these results are compared versus FC Geometry 1 on a laterally averaged basis in Figure 182, Figure 183, and Figure 184, respectively. Error bars placed on the data represent the approximately 12% experimental uncertainty derived from the experimental methodology. It is seen that Mayle & Camarata's results clearly fall within this error band, and the trend with M is captured effectively in FC Geometry 1 data. In summary, for an inclined and compounded hole configuration such as FC Geometry 1, it may be stated that film injection causes a 20% to 30% increase in heat transfer in the periodically fully developed region.

Figure 182 FC Geometry 1 Heat Transfer Compared to Mayle & Camarata, $M = 0.5$

Figure 183 FC Geometry 1 Heat Transfer Compared to Mayle & Camarata, $M = 1.0$

Figure 184 FC Geometry 1 Heat Transfer Compared to Mayle & Camarata, M = 1.5

Local heat transfer enhancement results for FC Geometry 2 are provided in Figure 185, Figure 186, and Figure 187. Without the inclusion of compound angle, enhancement is localized primarily in the region immediately downstream of injection. The increasing blowing rate had a much smaller impact on the result than with compound angle injection, and the coolant did not appear to drift laterally as its momentum was aligned with the mainstream flow. One probable cause for the reduced effect of M and reduced overall h_f/h_0 is the method of jet-crossflow interaction. At low M (specifically, at low I), the compounded jet impacts the cross flow and leads to increased turbulent mixing as the coolant is turned into the mainstream direction. While the low momentum jet is being turned it is forced against the wall and high heat transfer results along this diagonal region of impact. High resolution data provided by Goldstein (2001) highlights this behavior. At high M and high I, the jet momentum dominates, and a similar interaction takes place although turbulence is being generated from the jets turning the direction of the crossflow near the wall. Successive interactions associated with multiple rows of injection

further perturb the boundary layer flow and exacerbate the heat transfer augmentation. More discussion of flow phenomena associated with compound angle injection is available in the heat transfer study by Goldstein (2001).

The critical factor with respect to FC Geometry 2 is the lack of such a significant jet-crossflow interaction. Much like the normally-injected jet in a crossflow model shown in Figure 156, inclined injection leads to vortical structures formed on either side of the jet and increased turbulence associated with the mixing of flow streams of unequal momentum. However, this interaction does not encourage jet spreading or intense mixing as were seen with the compounded jet, and hence the effects are much more localized. However, as periodic injection continues, these vortical structures compound upon one another and become well-developed. As a result, heat transfer is increased slightly (10-20% overall) as compared to an uncooled boundary layer and the effects of injection protrude far downstream. In the recovery region for each FC Geometry 2 data set, the presence of these structures is obvious, and their intensity does not decrease significantly, even as the coolant effectiveness diminishes.

Figure 185 FC Geometry 2 Heat Transfer Enhancement, $M = 0.5$

Figure 186 FC Geometry 2 Heat Transfer Enhancement, $M = 1.0$

Figure 187 FC Geometry 2 Heat Transfer Enhancement, $M = 2.0$

In a similar fashion to FC Geometry 1, laterally averaged heat transfer enhancement and net heat flux reduction are presented in Figure 188 and Figure 189. While h_f/h_0 was reduced as compared to compound angle injection, Θ was lower for FC Geometry 2 than FC Geometry 1 and therefore compound angle injection would be more effective at protecting the cooled surface in the configuration tested. Increased M for purely inclined injection led to high jet blow-off and

very low cooling effectiveness, and the Θ results show that the advantage of coolant addition is almost negligible at $M = 2.0$, except in the region well downstream of the initial row of injection. Designers may choose a lower blowing ratio to conserve coolant and produce overall better performance when dealing with inclined cylindrical hole injection.

Figure 188 FC Geometry 2 Laterally Averaged Heat Transfer Enhancement

Figure 189 FC Geometry 2 Laterally Averaged Heat Flux Reduction

Section 4.5 Impingement-Effusion Cooling System Performance

Critical to the performance of an impingement-effusion cooling system model tested are impingement Nusselt number (Nu_D), film cooling effectiveness (η), and film cooling heat transfer enhancement (h_f/h_0). The use of these three, along with appropriate thermodynamic and aerodynamic parameters, allows for approximate solution of steady-state metal temperature in such a system. In order to illustrate such a calculation, a simplified, arbitrary heat transfer model will be utilized. All geometries for which impingement and effusion performance was investigated will be compared.

First, a steady-state one-dimensional heat transfer model is used, as shown in Figure 35. Various simplifying assumptions are made, such as:

- Realistic values of impingement hole diameter (D_i) and mainstream flow uncooled convective heat transfer coefficient (h_0) are assumed, for demonstrative purposes.
- The conduction resistance in the wall is insignificant compared to the convective resistances, as thermal conductivity of metal is relatively high and wall thickness may be relatively small in a gas turbine cooling system.
- Similarly, local resolution of wall temperature would be possible but unrealistic, as lateral conduction in the wall typically “smoothes out” the effects widely varying convective boundary conditions. Accordingly, Nu_D , η , and h_f/h_0 are area-averaged and a single, average wall temperature value is obtained.
- Coolant temperature does not change from impingement hole inlet to effusion hole exit. Note that this is an unrealistic assumption and will cause wall temperatures to appear lower than would exist in a real cooling system.
- As indicated by the 1-D thermal circuit, effects of radiation are ignored.

Using the methods of Gritsch (1999), film cooling heat transfer is redefined so as to incorporate the effects of η and h_f as a single heat transfer coefficient, $h(\theta)$, as shown in (48). Using this nomenclature, a solution for the dimensionless wall temperature parameter θ may be obtained through rearrangements shown in (50) and (51). The result in (51) is most easily written in terms of the dimensional h_i and h_f , but it has also been rewritten with the non-dimensional parameters presented in this text. (Note that some texts use the inverse of θ as the overall cooling effectiveness, for example Landis (2009). θ will be used in this case as reduced θ equals lower target wall temperature and therefore the superior cooling. This is acceptable for the context of this discussion, and it prevents the unnecessary introduction of more dimensionless parameters.)

$$h(\theta) = h_f(1 - \eta\theta) = \left(\frac{h_f}{h_0} \right) h_0(1 - \eta\theta) \quad (48)$$

$$\theta = \frac{T_m - T_c}{T_m - T_w} \quad (49)$$

$$q'' = \frac{T_m - T_c}{\frac{1}{h_i} + \frac{1}{h(\theta)}} = h(\theta)(T_m - T_w) \quad (50)$$

$$\theta = \frac{h_i + h_f}{h_i + \eta h_f} = \frac{Nu_D \frac{k_f}{D_i} + \frac{h_f}{h_0} h_0}{Nu_D \frac{k_f}{D_i} + \eta \frac{h_f}{h_0} h_0} \quad (51)$$

Given this relationship and all the assumptions that have been presented thus far, trends of θ are presented in Figure 190 and Figure 191 for Geometries A and E. Note that the parameter θ must always be greater than unity by definition, and that low θ (in other words, low temperature) is often desirable, as the purpose of cooling is to reduce the metal temperature. Two different cases have been presented: in the first (Figure 190), the hot gas convective cooling

coefficient (without film addition) is much greater than the impingement heat transfer coefficient; the data in Figure 191 assumes the opposite. For comparison, the y-axis gridlines are equally spaced on both plots; y-axis values (other than 1) have been removed, as arbitrary values of h_0 and D_i were used. Gas turbine hot gas path flows maintain a high velocities and turbulence intensities (a result of combustion and wake effects). These, in addition to thin boundary layers, typically result in high h_0 . Therefore, the results in Figure 190 should be representative of a realistic system, while the trends seen in Figure 191 would only apply in a hypothetical case when impingement heat transfer coefficients dominate.

In Figure 190, it can be observed that the magnitude of impingement heat transfer has a much more significant effect on wall temperature when film cooling effectiveness is relatively low. However, in this case film cooling effects clearly dominate, and geometries with high film cooling performance (i.e. Geometry A) provide a significantly higher level of cooling.

Figure 190 Average Dimensionless Wall Temperature θ , $h_i \ll h_0$

In Figure 191, the wall temperature is primarily dependant on impingement height. Film cooling only produces a secondary effect as hot gas heat transfer coefficients are relatively low. Geometry E, with a generally lower Nu_D than Geometry A, produces a favorable result only because the geometries are being compared versus impingement Re_D . With smaller impingement jet diameter D_i , jet velocity must be higher than that of Geometry A, and therefore impingement heat transfer coefficients are comparable between the two geometries. Another key observation from Figure 191: as the y-axis gridlines are spaced equally in this and Figure 190, it is observed that θ is very close to unity for all cases, regardless of impingement-effusion geometry.

Figure 191 Average Dimensionless Wall Temperature θ , $h_i \gg h_0$

These summary results are arbitrary as no concrete boundary conditions are available, but the methods used could easily be combined with appropriate values of h_0 and D_i to evaluate these impingement-effusion geometries in any design scenario. Consideration must be given by the designer to the ability for the given pressure ratio to maintain the coolant flow rates presented. In

essence, this pressure ratio will govern the design because it helps drive the selection of D_i , while an estimate of h_0 is already available. From here, values of θ may be evaluated. An understanding of the relative magnitudes of heat transfer in such a system is important when weighing the positive and negative aspects of different geometrical configurations and flow parameters.

CHAPTER 5. CONCLUSION

Section 5.1 Summary of Results

Results for impingement Nusselt number, film cooling effectiveness, and film cooling heat transfer enhancement in a coupled impingement-effusion cooling system have been presented. A wide variation in jet array configurations and coolant flow rates has allowed for some general conclusions to be developed regarding these parameters' effect on cooling performance. Some new parameters and data analysis methods were generated to allow for additional evaluation of heat transfer uniformity, film cooling effectiveness superposition accuracy, and overall cooling capability. These methods, along with various comparisons to the available literature, provide unique insight into coupled impingement-effusion cooling system design.

With regard to impingement heat transfer distributions, a variety of different jet-to-jet (X_i/D_i and Y_i/D_i) and jet-to-plate (Z/D_i) spacings were studied. Local impingement Nu_D results were presented, allowing observation of a high level of non-uniformity for Low Z ($Z/D_i < 6$) configurations and a relatively uniform Nu_D distribution for High Z ($Z/D_i > 10$) configurations. The greatest heat transfer was observed at the low jet-to-jet spacing, but it was shown that high jet-to-jet spacing and high Re_D is most efficient on the basis of coolant spent per unit surface area cooled. Low jet-to-plate spacing produces significantly higher heat transfer than taller impingement configurations, as the jet's momentum is greater at the point of impact. However, the comparison between short and tall impingement array performance had been overlooked in the open literature and thus this work helped to alleviate this deficiency. Also, it was shown that

the Nu_D was approximately inversely proportional to the square of Z/D_i , and therefore increasing impingement height had little effect on heat transfer for very tall impingement ($Z/D_i > \sim 15$).

Local distributions of film cooling effectiveness on the hot side of the impingement-effusion target plate were very useful in identifying jet trajectories and interactions. The advantages of staggered over in-line arrangements were viewed both qualitatively and quantitatively through local data and laterally averaged effectiveness distributions. Jet interaction in in-line arrangements led to a portion of the coolant being directed away from the target surface. Furthermore, jet spreading was insufficient to adequately “fill-in” the regions between injection locations, creating large lateral non-uniformities in η . Staggered arrangements, however, offered a more even lateral coolant distribution.

Five rows of injection were used in this full coverage film cooling study, and the results were compared with 15 row full coverage data for comparison. These large full coverage arrays utilized very high hole-to-hole spacing (to minimize coolant usage) and also were unshaped cylindrical coolant holes, but at an inclination to the mainstream flow. One configuration involved inclined and compounded holes. Significantly higher η was generated at equivalent M for the compound angle arrangement due to the physics of the jet-cross flow interaction. Again, jet spreading and the subsequent uniform lateral coolant distribution played were critical to the success of this arrangement. In the recovery region, performance was correlated to the amount of coolant injected; low M led to a rapid decline in η , while high M allowed for cooling well downstream of the last row of injection.

While the similarities among the five and 15 row configurations were few, it was observed that, in general, the magnitude of η was tightly correlated to the quantity of coolant injected. However, η was highly sensitive to hole arrangement, injection angle, and blowing

ratio. In most cases, cooling effectiveness increased with every row of injection. The exception to this was predominantly the in-line effusion geometry, where a periodically fully developed condition was reached as the rate of enthalpy injection was equal to enthalpy loss into the bulk flow. Attempts to predict full coverage film cooling performance via the additive film superposition method showed promise, but consistently under-estimated performance of staggered hole arrangements. In general, this superposition technique is a good model if no other data is available – but its performance is highly dependant on the method by which the single-row η data is determined.

Film cooling heat transfer enhancement data showed that full coverage film cooling injection generates higher heat transfer coefficients than an uncooled flat plate. The effect of this is parasitic to the cooling system, generating higher wall temperatures. Heat transfer enhancement is primarily the result of turbulent mixing of coolant jets and mainstream gas. This effect is magnified as the coolant jet momentum increases and therefore h_f/h_0 generally increases with blowing ratio. Enhancement effects are primarily located immediately downstream of the injection location, and h_f/h_0 typically returns to values near unity in the recovery region. Heat transfer enhancement associated with compound angle injection was higher than the non-compound case due to the intensity of the jet-cross flow interaction. Regardless, it was shown that the net heat flux reduction for the this case remained superior due to the increased effectiveness associated with compound angle injection.

Finally, some system-level considerations were made to analyze the effect of impingement and film cooling heat transfer on the simulated wall temperature in a gas turbine cooling application. With some simplifying assumptions and the choosing of a few arbitrary parameters, the dimensionless wall temperature ratio was calculated for several impingement-

effusion configurations at different coolant flow rates. It was shown that wall temperature was relatively insensitive to coolant flow rate in the case of high heat load configurations with small jet-to-jet spacing. These geometries could effectively be used to maintain the material at a temperature well below that of the mainstream flow, but at the expense of high coolant usage. Conversely, configurations with large spacing provide only moderate cooling, but conserve coolant. The effect of impingement height is only substantial in conjunction with low film cooling effectiveness on the hot side; in most full coverage film cooling scenarios, the opposite will be true, as η increases with the buildup of film.

Section 5.2 Recommendations for Future Work

This research brings about many additional items of interest. First and foremost, while a wide range of parameters in the impingement-effusion array were varied, future studies may focus on the system design from an optimization point of view. A wider range of coolant flow rates should be tested, especially to see the effects on film cooling performance at higher M . Lower impingement heights should be investigated to evaluate the optimal jet-to-plate spacing. Film cooling with angled injection should be applied to the tighter hole spacings of the impingement-effusion geometries to evaluate the effect of inclination and compound angle.

In addition, there is a great deal of research potential involving full coverage film cooling. Superposition methods continue to need further validation and evaluation; to begin, single row data should be experimentally measured for these configurations and used in the additive film method calculation. By its nature, the dynamics of full coverage film cooling are considerably more complex than those of single row film cooling, this topic could easily be the focus of future investigations.

The potential for life analysis work exists as well, as the convective heat transfer boundary conditions on a three-dimensional surface have been established on a local basis. The interaction of thermal and mechanical stresses due to cooling effectiveness and effusion hole size and configuration needs to be evaluated. Such an analysis would provide a new perspective on the practical design of an impingement-effusion cooling system.

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