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An Experimental Investigation of Heat Transfer for Arrays of impingement Jets onto the Featured Surfaces with Cylindrical and Elliptical Raised Surfaces

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AN EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER FOR
ARRAYS OF IMPINGEMENT JETS ONTO THE FEATURED SURFACES
WITH CYLINDRICAL AND ELLIPTICAL RAISED SURFACES

by

MARC A. MEDINA

A thesis submitted in partial fulfillment of the requirements
for the Honors in the Major Program in Mechanical Engineering
in the Department of Mechanical, Materials, and Aerospace Engineering
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at the University of Central Florida
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ABSTRACT

This study focuses on multi-jet impingement for gas turbine geometries in which the objective is to understand the influence of the roughness elements on a target surface to the heat transfer. Current work has proven that implementing roughness elements for multi-jet impingement target surfaces has increased heat transfer ranging anywhere from 10-30%. This study has chosen to investigate three different roughness elements, elliptical in cross-section, to compare to smooth surface geometries for multi-jet impingement.

An experimental was taken for this study to extend the current knowledge of multi-jet impingement geometries and to further understand the heat transfer performance. A temperature sensitive paint (TSP) technique was used to measure the heat transfer on the target surface, in which the local temperature was measured to estimate area averaged heat transfer coefficient (HTC) and row averaged HTC.

In order stay consistent with literature, non-dimensional parameters were used for geometry locations and boundaries. For this study, the Reynolds number range, based on jet diameter and mass flux, is 10-15k. The X/D (streamwise direction), Y/D (spanwise direction), Z/D (channel height direction), L/D (thickness of the jet plate) constraints for this study are 5, 6, 3, and 1 respectively. From the local heat transfer distributions of the different roughness elements, it is concluded that the inclusion of these elements increases heat transfer by 2-12% as compared to a flat/smooth target plate. It is therefore recommended from this study, that elements, elliptical in shape, provide favorability in heat transfer for gas turbine configurations.

DEDICATION

To my family. For endless support throughout my entire life.

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NOMENCLATURE

A_c	Cross Sectional Area (m^2)
A_h	Heater Surface Area (m^2)
c_p	Specific Heat at Constant Pressure ($kJ/kg \cdot K$)
D_{jet}	Jet Diameter (m)
G	Mass Flux ($kg/s \cdot m^2$)
H	Feature Height (m)
h	Heat Transfer Coefficient (HTC) (W/m^2K)
$\bar{h}$	Average Heat Transfer Coefficient (W/m^2K)
I	Heater Current (Amps)
k	Thermal Conductivity ($W/m \cdot K$)
$\dot{m}$	Mass Flow Rate
N_h	Number of Impingement Holes
Nu	Nusselt Number
q	Heat Input/Output (kJ/kg)
q''	Heat Flux (W/m^2)
R_{el}	Total Heater Resistance (ohm)

Re	Reynolds Number
r	Jet Radius (m)
T	Temperature (K)
V	Voltage
w_{net}	Net Work
X	Streamwise Distance (m)
Y	Spanwise Distance (m)
Z	Impingement Height/Channel Height (m)
η	Thermal Performance Parameter
μ	Dynamic Viscosity

CHAPTER 1: INTRODUCTION

Turbomachinery, Power Generation, Gas Turbines

From the beginning of time, man has been clever enough to engineer technologies that create energy in order to increase the standard and quality of life. Harnessing the energy and work being produced by such mechanisms can come from many forms. As there are many different needs or demands for the transfer of energy, turbomachinery has found its place in today's existence for power generation. In particular, turbomachinery has had its major application in commercial aviation and city power generation, as its ability to produce work/power from a rotating shaft seems suitable for such fields.

Through the generations of their existence and the technological advances turbomachinery has made, it is inherent that these systems are not 100% efficient. In order to produce power, turbines follow a common thermodynamic procedure. In particular, the performance of gas turbines can be best described through a thermodynamic cycle, the Ideal Brayton Cycle.

Figure 1 - Ideal Brayton Cycle for Gas Turbines

Figure 2 - Ideal Brayton Cycle T-s Diagram

Here, the cycle describes a Gas Turbine by the following; fluid enters the system and travels through a compressor, kinetic and internal energy increases the fluid velocity and travels through the turbine and a combustion subsystem, the combustion subsystem then takes some of the incoming compressor fluid to mix at a combustion reaction where temperature of the fluid increases rapidly, fluid re-enters the turbine where it travels through a series of internal rotating geometries, such as blades, in order to rotate a central shaft to produce work/energy. Because the process can be complex, there are multiple opportunities for the mechanism to not perform at a high efficiency due to irreversible losses. However, an obvious variable that allows the system to perform at its maximum capability is the exit temperature leaving the combustor. The efficiency of the engine from the Brayton Cycle can be quantified and defined by Equation 1.

$$\eta_{th,Brayton} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)}$$

Equation 1 - Thermal Efficiency of Ideal Brayton Cycle

From this equation, efficiency is governed by temperatures and pressures at certain stages of the cycle. As mentioned, higher exit temperatures leaving the combustor can increase the efficiency of a turbine and is generally used as a maximum or optimal parameter for designers.

Furthermore, if achieving a higher (T3) is the objective, designers also must consider the maximum allowable temperatures the internals of a gas turbine can withhold. For example, ignition temperatures within the combustion process can reach up to 2800°C, whereas the maximum allowable temperatures for super alloys used in some areas of the turbine are 1200°C.

The balance of reaching high temperatures exiting the combustor without harming or ruining the integrity of the materials of the turbine geometries is crucial to the performance and can be accommodated with the use of advanced cooling schemes.

Internal Cooling Schemes

As hot gasses exit the combustion chamber and enter the turbine, fluid temperatures can exceed the maximum allowable fatigue limit for some materials and geometries. Obtaining a higher combustion temperature for a turbine can be an objective for most turbine companies. As the combustion temperature increases from one design to another, close attention is brought to the

turbine blades and their cooling schemes. There are various methods to cool the blade externally and internally in order to accommodate such inlet temperatures of the turbine. Such methods include transpiration cooling, film cooling, impingement cooling, and augmented channel cooling. The objective for such cooling methods is similar with different applications for various locations of the blade. The main objective, for internal cooling, is to internally transfer cooled air, from the compressor, to the walls and outer layer of the blade, where temperatures are highest in those regions. Figure 4 shows a turbine blade with typical internal cooling schemes.

Figure 4 – Typical Internal Cooling Schemes for a Turbine Blade

A combination of various internal cooling methods is used in order to redirect the fluid to specific areas of the blade. As these methods have been advanced over the past decades, there are some challenges and goals in order to optimize the process of internal cooling. Some of these

challenges include; increasing heat transfer through the channels of the blade, finding a balance of pressure drop through the channels, redirecting the flow efficiently to critical areas, and providing a structurally sound geometry. For example, the inclusions of roughness elements (such as slanted ribs or slots) within the internal channels have been proved to increase heat transfer and manipulate flow to specific critical walls. However, if rib number, height, width, and orientation are not optimized to the ratio of channel dimensions, then the casualty of a higher overall pressure drop may exceed the benefit of the increased heat transfer. The inclusion of different roughness elements has also been implemented to support the inner walls of the blade with an additional benefit to increase heat transfer. As there are many other variables that must be considered, a compromise between all internal cooling challenges must be made. A critical cooling scheme still studied today is the impingement method, where internal coolant flow is directed to the outer channel and sleeve of the blade. Here, research endeavors are still being pursued on this topic in order to optimize the best design for cooling various geometries in a wide spectrum of applications and industries.

Multi-Jet Impingement

Jet impingement for a cooling scheme can become challenging for different applications. The idea of jet impingement is to direct cold flow through a jet hole, or an array of holes, in order to impinge, or target, on a heated surface. The application of this method is endless, as it's used in the electronics, automobile, aviation, and turbine industry. More specifically to the turbine industry, jet impingement is being used as a way to cool the outermost shell, internally, of blades from the compressor fluid. There have been other applications of impingement in the

turbine industry, as some companies have used this method to cool the outer surface of a combustor transition piece, another life critical component to the combustion cycle.

As a way to supply fluid to the outer-most channels of a turbine blade, there can be challenges similar to the ones mentioned earlier. As internal geometries of a blade can change from design to design, jet impingement can be influenced and will therefore be defined from the upstream fluid. Nevertheless, research endeavors are being made to accommodate most flow configurations.

First, a clear definition of impingement and its characteristics must be analyzed before discussing the benefits, challenges, and consequences. Figure 5 shows a schematic of a single fluid jet impinging on a target surface. There are three major areas of impingement that must be understood; free jet region, stagnation flow region, and wall jet region (as labeled in Figure 5).

Figure 5 - Single Jet Impingement Schematic

Fluid enters an orifice plate with a well-defined thickness and jet diameter, is directed in the jet hole direction and enters the impingement channel, becomes fully-developed shortly after entering the channel, impinges on another surface (defining a stagnation region), and then redirected to the adjacent open areas of the channel. The different regions of the fluid entering the channel and impinging on the target surface can be characterized best by discussing the interaction of the jet and its' surrounding. It is typical to describe its interaction with the surrounding influences by discussing the entrainment of mass, momentum, and energy [2]. Research studies have been performed in order to characterize and approximate the characteristics of this flow pattern. Some studies include the following examinations; the flow-development region, approximations of the velocity profile impinging onto the surface, turbulent vs. laminar regions, stagnation influences, boundary layer thickness development, and wall-jet area.

Fluid characteristics can become even more complex by implementing an array of impinging jets onto one target surface, as configurations can become endless by the creativity of designers. Before discussing which parameters and creative geometries are being used for blade cooling, an understanding of multi-jet impingement must be understood. Similar to single-jet impingements, flow enters an orifice plate, is directed in the hole orientation, and impinges onto the surface. If there are neighboring jets, flow within the channel can become complex, as now, the interaction from jet to jet must be considered. Figure 6 shows a schematic of a jet-to-jet interaction, with labeled components.

Figure 6 – Multi-jet Impingement Schematic

Similar to single-jet impingement, the fluid has the same components and characteristics, such as the free jet region, stagnation region, and wall jet regions. However, the difference now lies with two distinct interactions. The influence each jet has to the neighboring jet prior to impinging on the target surface and the collision of surface flow from adjacent jets [2].

Furthermore, the interactions are influenced by the spacing between the jets, jet diameter, the specific arrangement of the jets, jet velocity and much more. The interactions mentioned earlier will determine how well the flow influences heat transfer of the heated target plate as well as its collision with additional obstacles, which will be mentioned in this study. With the inclusion of multiple impinging jets for a target surface, there are various scenarios that can be predicted.

First, in a channel with multi-jet impingement, fluid will develop and influence neighboring jets inducing a crossflow. This crossflow can be induced in various ways depending on array arrangement, height of the channel, boundary side walls, and Reynolds number of the channel. Each jet will be influenced based off crossflow excitation and strength. This can influence jet

flow to change the stagnation zone to be offset from the jet hole region. Furthermore, crossflow can influence the level of turbulence in the vortice region and fountain flow region. Nevertheless, crossflow direction is typically defined by the geometry boundaries, where entering and exiting areas are known. Figure 7, shows some different crossflow schemes for multi-jet impingement.

Figure 7 – Crossflow Scheme Diagram

As a means to increase heat transfer, with the exclusion of other thermo-fluid goals, various schemes can be used for multi-jet impingement in order to make such accomplishment. Before trying to optimize the impingement geometry, an understanding of the specific application must be made in order to make further design choices. For jet impingement in turbine blades, some areas are defined by the spacing between the outside/exposed heated surfaces (from the hot gasses) to the orifice plate. Other areas are defined by the number jet columns allowable, where some applications focus on a single line of jet arrays [2]. Some design endeavors focus on the flow schemes directly prior to the impingement holes, where its characteristics also have an effect on the impingement channel.

The different design schemes for multi-jet impingement include the following standard parameters: jet diameter, distance between the each jet (spanwise and streamwise flow direction), arrangement of jets (inline vs. staggered), channel height, jet exit velocity, Reynolds number, thermal wall boundary conditions, orifice plate thickness, number of jet columns and rows, and crossflow direction. With this in mind, research efforts have been made to address every possible design scheme that can be used for this specific application in order to conclude which arrangements can best accommodate heat transfer and fluid flow challenges.

Literature Review

In order to add to past research studies on multi-jet impingement, a thorough literature review has been conducted in order to find novelty for progression on this topic. As discussed earlier, the multi-jet impingement application can vary on a case to case basis depending on the objective. Furthermore, applications will depend on some of the following; geometry for impingement, parametric bounds, flow conditions, and heat transfer objectives.

Many studies have investigated a large quantity of various parameters for smooth plate impingement cases, as this is typically a baseline for experimental and numerical setups. Extensive work has been conducted in order to investigate the flow and heat transfer characteristics of impingement cases [3]. Florschuetz conducted a variety of impingement cases in order to characterize flow and heat transfer distributions for complete crossflow geometries. More specifically, his experiments include a copper block isothermal technique in order to measure row averaged Nusselt Number on the target surface of the impingement channel.

Varying geometric parameters include; X_n/D (streamwise jet spacing), Y_n/D (spanwise jet spacing), Z/D (channel height spacing), inline vs. staggered jet arrangements, and jet diameter. Other parameters include; a defined range for Reynolds number cases, number of rows included for each test article, and number of copper blocks per geometry for spatial resolution. Florschuetz has also developed flow distribution correlations from a one dimensional model for multi-jet impingement. The correlation would create a design platform so that one may be able to predict heat transfer distributions as a function of jet and crossflow rates associated with spanwise jet rows, and different geometric parameters. It is important to note that the application for this correlation depends on the geometric and fluid characteristic of the impingement channel. Nevertheless, the experimental results, from the heat transfer distributions, describe various test articles which prove to be a good starting point for impingement investigations. This study will attempt to compare row averaged copper block heat transfer results, from Florschuetz, to laterally averaged temperature sensitive paint heat transfer results for smooth plate test articles.

Other studies have also been made in order to obtain additional information of the effect from crossflow to other impingement jets on a smooth plate [2]. Terzis has made some major contributions by obtaining heat transfer results for narrow impingement test cases, where its application is used for cast turbine airfoils. Here, Terzis has obtained heat transfer results for single row impingement cases with the use of a transient liquid crystal technique. Varying parameters include, jet to jet spacing, channel width and height, jet pattern and varying jet diameters. Results describe the heat transfer results on the target plate and side walls for varying

parameters. It is concluded that major contributors to the performance of heat transfer include crossflow, geometric dimensions, jet patterns and spacing, flow inlet conditions, and boundaries defined by the narrow wall channel. CFD simulations were also ran in order to compare experimental heat transfer results. From the experimental and numerical results, it is recommended to find impingement methods and cases to increase heat transfer downstream of the channels where jets are heavily affected by crossflow. A recommendation to implement surface elements is suggested in order to minimize crossflow which will inherently increase heat transfer for impingement cases.

Spring [4], has expanded upon this research by implementing raised surfaces on an impinging target surface. More specifically, Spring has implemented ribs with different orientations to crossflow schemes for multi-jet impingement geometries. Experimental and numerical approaches were taken in order to characterize the heat transfer distribution on the target surface. A comparison for flat vs. a roughened target surface was made in order to understand the benefit of rib elements with their respective arrangements and orientation. It is concluded that roughened elements on a target surface for impingement cases can increase heat transfer to downstream areas of the channel.

Other studies have progressed on the idea to implement elements to increase heat transfer for critical regions in the impingement channel [5]. Research endeavors and designers have attempted to optimize this implementation by investigating new and creative geometries within the impingement channel. Son, has investigated several geometries, in a staggered arrangement, and their performance on heat transfer to the target surface of an impingement channel. Some of

these elements include; cylinders, diamonds, augmented ribs, and hexagonal protrusions. Son has also identified that location of each element is crucial to the performance of impingement cooling. It is indicated that the lower heat transfer regions, typically between impingement jets, can contribute more effectively by becoming a space for roughness elements. As discussed earlier, a balance of flow and heat transfer must be made in order to optimize the impingement channel. As a means to address this topic, Son, has also quantified some of the flow disadvantages as he implements each element onto the target surface. Adding roughness elements can prove to have their advantages and disadvantages. Aside from collecting heat transfer results, from a TLC method, Son has also collected pressure measurements along the streamwise direction of the impingement channel. As discussed earlier, larger pressure drops can be detrimental to the overall design of a cooling channel. Son has quantified the pressure drop across each channel tested with their respective internal geometry and their inclusion of raised surfaces. As expected, results show that heat transfer can rise by more than 30% with the inclusion of roughness elements with the expense of a larger pressure drop as compared to a smooth case channel. However, the pressure drop penalty did not exceed 10% as compared to smooth target surfaces. This study will attempt to also compare to Son's TLC methodology for roughened surfaces in a multi-jet impingement channel by using temperature sensitive paint.

Other studies have also indicated the same trend of heat transfer results with flow penalties[6]. Persson, has investigated the enhancement of heat transfer from turbulators for target surfaces with the elements in a staggered jet arrangement. Such elements include triangles, wings, cylinders, and portioned ribs. Other studies, [7],[8],[9],[10], [11], also investigated other

roughness elements and their effect on heat transfer for single jet configurations. Such elements include; pyramidal protrusions, square pedestals, cylinders, concentric rings, tall fins, annular rings and much more.

In order to find novelty in literature while also attempting to stay consistent with popular geometric trends, a spreadsheet of some of the contributing studies has been made. Table 1 shows a list of the major contributors and their investigations compared to this study.

From Table 1, it can be seen that multiple studies have been conducted to implement roughness elements on an impingement surface. Furthermore, numerous studies have investigated rib obstacles and cylinders. Various spacing parameters have been investigated with varying Reynolds numbers and jet diameters.

This study will continue research on roughness elements by implementing cylinders and, for the first time, elliptically shaped raised features to the target surface. This study will use a different measurement technique in order to measure the temperature distribution on the target surface, temperature sensitive paint. In order to stay consistent with literature, X/D , Y/D , Z/D , and Reynolds numbers will stay within the same range as past studies. In order to maintain a consistency for the raised surfaces being used for this study, volume for each feature stayed the same.

Table 1 – Literature Review Spreadsheet

Literature Review Comparison										
#	Author	Source	Geometry	Novelty	X/D	Y/D	Z/D	Reynolds Number	D _{jet}	Measurement Technique
1	Alexandros Terzis	Stuttgart University	Smooth	Smooth Single Row	5, 8	3, 4, 5, 6	1, 1.5, 2, 3	1-8.6 (x10 ⁴)	N/A	TLC (Local HTC)
2	Florschuetz	NASA	Smooth	Smooth Large Array	5, 10, 15	4, 6, 8	1, 2, 3, 6	5-52 (x10 ³)	.0635-.762 (in)	Copper Block (Area Avg.)
3	Son	Oxford University	Featured	Cylinders and Novel	3-3.7	3.75-4.8	1.88-3	17-41 (x10 ³)	10-16 (mm)	TLC (Local HTC)
4	Hoecker, Johnson	ABB Power Gen.	Featured	Cylinders	4.35, 6.5, 13	4.17, 6.227, 12.454	1.67-12.5	21k-80k	2.4, 4.8, 7.2 (mm)	Copper Block (Area Avg.)
5	Yan	Huafan University	Featured	Ribs and Augmented Ribs	6	3	3, 6, 9	1-5k	5 (mm)	TLC (Local HTC)
6	Andrews	Leeds	Featured	Slotted Ribs					Sqaure	Stainless Steel/Thermocouple
7	Andrews	Leeds	Featured	Interrupted Rib Obsstacles	1.86, 3.06, 4.66	1.86, 3.06, 4.67	1.1, 1.81, 2.75		Square 3.27, 4.98, 8.19 (mm)	Stainless Steel/Thermocouple
8	Persson	Lund Inst. Of Tech., Alstom	Featured	Triangles, Wing, Cylinder, Rib	5	5.77	.82, 1.73, 2.45	20k-65k	22 (mm)	IR (Local HTC)
9	Dongliang	Northwestern Polytech.	Featured	Cylindrical Pins and Film Holes	4.8	4.8	1		1 (mm)	IR (Local HTC)
10	Azad	Texas A&M University	Featured	Cylindrical Pins	2, 4	2, 4	3	4.8k-18.3k	0.635 (cm)	TLC (Local HTC)
11	Florschuetz Validation	UCF	Smooth	Smooth Large Array (Staggered)	5	8	2, 3	4k-30k	5 (mm)	TSP (Local HTC)
12	Son Validation	UCF	Smooth and Featured	Uniform Baseline Smooth and Cylinder Pins	3.7	4.6	2.3	20-31k	13 (mm)	TSP (Local HTC)
13	Marc Medina	UCF	Featured	Baseline Cylinder Pins (5x9)	5	6	3	10-15k	7 (mm)	TSP (Local HTC)
14	Marc Medina	UCF	Featured	Baseline Elliptical Feature Pin (5x9)	5	6	3	10-15k	7 (mm)	TSP (Local HTC)

CHAPTER 2: EXPERIMENTAL PROCEDURE

Temperature Sensitive Paint

As shown in Table 1, numerous methods have been used to characterize heated surfaces. The main objective, for heat transfer results, is to characterize a target surface with the use of a temperature measurement technique in order to quantify Nusselt Number, a non-dimensional heat transfer parameter. Each technique used from past and current studies has their advantages and disadvantages. Nevertheless, choosing the correct technique usually depends on resources available, heat transfer goals, and uncertainty.

In order to characterize each target plate on the test subjects used for this study, a temperature sensitive paint technique was used. Like other methods used in literature (Transient Liquid Crystals and Infrared), this technique measures temperature distribution locally. Obtaining local data for impingement geometries has its advantages, and even more so in multi-jet applications.

Temperature sensitive paint (TSP) is a method that can be used for visualization and quantitative temperature distribution results. As a thermographic phosphor, a thin coating of temperature sensitive paint can be excited by narrow band incident light. Once the fluorescent molecule is excited within the binder, light is then emitted, defined by a certain wavelength, and then can be captured by a CCD camera. The wavelength emitted can be calibrated in order to relate intensity to temperature.

Figure 8 – Typical Temperature Sensitive Paint Experimental Schematic

Depending on the application, TSP chemical formulas will change and are typically defined by a narrow band temperature range, typically around 100K.. In order to excite the luminiphores in the TSP, a light source is needed. High energy LEDs or Xenon flash lamps are typically used to excite the molecules.

In order to capture temperature distribution on a target surface, a CCD camera was used. In order to obtain thermographic data from TSP, two snapshot images have to be taken; a reference image, where known temperature is recorded and used as a reference temperature for the paint, and then one after some temperature distribution has changed from the reference. In forced convection applications, reference images at constant surface temperature are usually taken when the wind tunnel is turned on and reaches steady-state flow conditions. Throughout TSP testing, light emitted has to be constant, as images will be taken when temperature distribution changes to that of the references images and any changes of emitted light can result

in error. Another contributing factor that can lead to error is any background noise that is captured in the images. It is usually a good practice to take background images with no emitted light so that one may subtract that from emitted surface images. If the change in temperature from the reference images is too small, then it may be difficult to recognize the difference between the images. This usually means it is good practice also to have a large temperature difference from reference images to the heat transfer images. Furthermore, in order to obtain local data after all images are taken, each pixel in the areas of interest from the image must be recognized. To do so, MATLAB was used with its image processing capabilities. Once each pixel can be recognized as a data point, a temperature distribution can be visualized and quantified over the entire surface area.

Heat Transfer Measurement and Quantification

Because temperature distribution can be quantified over an entire area of interest with the use of TSP, a constant heat flux test can be imposed in order to characterize the heat transfer to the surface. This technique is a semi-invasive way to characterize heat transfer for a target surface. The end goal, from a temperature surface distribution, is to calculate local Heat Transfer Coefficient and Nusselt Number. In Equation 2, heat transfer coefficient can be calculated, where q''_{eff} is the evaluated heat flux on the target surface, T_w is the surface temperature measured by the TSP on the target surface, and $T_{coolant}$ is the impingement coolant temperature measured from the plenum of the system.

$$h = \frac{q''_{eff}}{T_w - T_{coolant}}$$

Equation 2 - Local Heat Transfer Coefficient Formula

$$q''_{eff} = q''_{gen} - q''_{cond}$$

Equation 3 - Effective Surface Heat Flux

The challenge for this method is to impose a constant heat flux on the target surface without being invasive to the geometry. In order to apply constant heat flux to the target surface, thin conducting surface elements were used. Thin stainless-steel heaters with a thickness of, 0.002 inches, were used to overlay on top of the painted surface. The stainless-steel heaters were adhered by using a double sided insulating tape, Kapton tape. In order to provide constant heat flux to the stainless-steel elements, a simple series circuit was formed to provide current and power. Heat flux provided to each of the elements can be expressed in Equation 4.

$$q''_{gen} = \frac{I^2 * R_{el}}{A_h}$$

Equation 4 - Constant Heat Flux Provided to Stainless-Steel Element

With this method, we can calculate local heat transfer coefficient, which can then be used to evaluate local Nusselt Number.

$$Nu = \frac{hD_{jet}}{k_{coolant}}$$

Equation 5 - Nusselt Number for Multi-Jet Impingement

Small copper panels (bus bars) were manufactured to provide suitable conductivity to each element and to hold each heater to the target surface. Below in Figure 9, is a side view representation of the target surface coupons used in each experiment.

Figure 9 – Target Surface Coupon Schematic

Data Processing

With the utilization of the governing heat transfer equations and MATLAB image processing, a streamlined process can be used to import all raw images and output contours of temperature distribution, local heat transfer coefficient, and local Nusselt number with their

respective data arrays. Below, is an example of how a rendered temperature distribution contour may look after processing the raw images.

Figure 10 – Temperature Distribution Contour Example

Once each pixel is assigned a temperature value, the next objective is to obtain corresponding local Nusselt Number. Furthermore, to compare results, lateral averages were taken in the spanwise direction (Y/D). Below, is an example of how the lateral average data was evaluated as compared to a Nusselt number distribution contour.

Figure 11 – Nusselt Number Distribution Contour Example with Lateral Averages

From this, laterally average data points can be plotted vs. streamwise direction (X/D). The entire surface area can also be averaged in order to obtain total target surface heat transfer results. The objective, from the lateral average and total surface area average data, would be to visualize and quantify the augmentation of heat transfer for different test cases.

Experimental Setup for the Multi-Jet Impingement Rig

The test subjects used in this study can be comprised of three different areas; a Florschuetz validation geometry, a University of Oxford validation geometry, and one unique to literature. Validation cases were used in order to identify some critical differences from measurement techniques as well as to establish credibility for the temperature sensitive technique. All three cases are multi-jet impingement configurations with boundaries to induce complete/maximum crossflow. Furthermore, all impingement geometries will have a staggered

arrangement. Each of the test subjects were setup in a similar fashion. Below, is a schematic of the system used to test each case.

Figure 12 – Rig Schematic

The experimental rig was setup in an open loop system. The blower used for the rig was located upstream up of the test section where it would be turned on in suction. Flow would be brought into the test section from the impingement hole array, impinge onto the target surface, where heat transfer measurements were taken, and then exit the test section where it would then leave the entire system. A venturi was also placed upstream of the test section where mass flow of the system was measured. Mass flow rate would be altered with an upstream valve in order to reach the desired Reynolds number. Average Reynolds number for a multi-jet configuration can be described by the equation below.

$$\overline{Re} = \frac{\bar{G}D_{jet}}{\mu_{fluid}}$$

Equation 6 - Reynolds Number Mean

Where, G is mass flux based on jet diameter.

$$\bar{G} = \frac{\dot{m}}{N_h A_c} = \frac{\dot{m}}{N_h \pi r^2}$$

Equation 7 - Mean Mass Flux for Multi-Jet Impingement

The experimental procedure would remain similar for all test cases. Once each test section was placed in the system, the CCD camera and LED light would be optimally positioned for heat transfer measurements. Below is a picture of the positioning of the CCD camera and the LED to that of the test section.

Figure 13 – Experimental Setup with Camera/Light Position to Test Article

After positioning the LED spotlight and CCD camera, electrical wires were connected to the end bolts protruding out of the test section where a simple series circuit was completed in order to provide current to the heater elements. As mentioned earlier, in order to acquire heat transfer measurements using the light intensity method, reference images and temperature difference images have to be taken. Reference images were taken when the blower was turned on and reached steady-state conditions at the desired Reynolds number without any power being supplied to the heater elements. Temperature distribution images were taken when current was provided to the heater elements. The images were taken when power provided reached steady-state. For all cases, current provided to the electrical elements depended on the maximum surface

temperature, where, the desired temperature difference from surface temperature to coolant temperature was approximately 40°C. This process would be used for all test cases and for all desired Reynolds numbers.

Florschuetz Validation Geometric Parameters and Test Conditions

From all the impingement cases Florschuetz tested, one particular case was chosen to be included in this study. Because this study will focus on impingement arrays in a staggered arrangement, a staggered geometry was chosen from Florschuetz. Indicated in Table 2 is the geometric configuration for the Florschuetz validation case. This case is a smooth wall test subject where there are no protruding pin fins or obstacles in the crossflow direction.

Table 2 – Florschuetz Validation Configuration

Jet Arrangement	Staggered
Target Plate Geometry	Smooth
Djet [mm]	5
X/D	5
Y/D	8
Z/D	3
Reynolds Number	5,000 & 20,000

University of Oxford Validation Geometric Parameters and Test Conditions

The second validation case that was used in this study comes from published work done by the University of Oxford. This validation case includes two different test specimens with the same boundaries and jet plate configuration. The difference between the two specimens is the target surface, where in one case, the target surface is smooth, and in the other, the target surface includes extended surfaces (cylindrical elements) which indicate a rough target surface. The cylindrical elements were 3D printed out of plastic and therefore are non-participating to the heat transfer on the target surface. In order to be consistent with previous works in literature, blockage of the extended surfaces is defined as the following:

$$\text{Blockage} = \frac{H_{\text{feature}}}{Z}$$

Equation 8 - Multi-Jet Impingement Blockage Definition

Below in Table3, is the geometric configuration and test condition for this validation case.

Table 3 – University of Oxford Validation Configurations

Jet Arrangement	Staggered
Target Plate Geometry	Smooth & Featured
Feature	Cylinders
Blockage	40%
Djet [mm]	13
X/D	3.6
Y/D	4.6
Z/D	2.3
Reynolds Numbers	21,000, 26,100, 31,800

University of Central Florida (Current Study) Geometric Parameters and Test Conditions

The experimental setup in the current study is similar to that of the previous validation case. The current study test cases include a smooth target surface as well as three other featured target surfaces. The features that will be added to the target surface will be similar to that from the University of Oxford. It will include a cylinder case, and two other elliptical shaped element cases. As mentioned before, all features used were 3D printed, out of plastic, and are non-participating elements. Some geometric variables remained the same for all featured cases such as volume and blockage. Table 4 lists all the geometric parameters for the cases mentioned.

Table 4 – Current Study Configurations

Jet Arrangement	Staggered
Target Plate Geometry	Smooth & Featured
Feature	Cylinders, Ellipse, Oval
Blockage	50%
Feature Volume [mm^2]	747.02
Djet [mm]	7
Total Number of Jets	49
X/D	5
Y/D	6
Z/D	3
Reynolds Number	10,000 & 15,000

Table 5 shows all features, designed in CAD, used for the current study.

Table 5 – Raised Surface Elements

Cylinder	Ellipse	Oval

The features were all oriented in such a way that their longest side was parallel with the crossflow directed. Below is an image of the crossflow to feature orientation.

Figure 14 – Crossflow to Feature Orientation

Repeatability

Because this heat transfer technique is not common for multi-jet impingement studies, repeatability was implemented for the University of Oxford validation case. The validation case chosen was the smooth target surface test article. Below, in Table 6, is the day-to-day repeatability schedule used. As seen, different repeatability methods were used, where, the same

tests were conducted over several days. In addition, the testing sequence for testing different Reynolds numbers changed, as well as the occurrence for each Reynolds number.

Table 6 – Day to Day Repeatability Test Schedule

Date	10/21	10/22	10/27	10/29	10/30
Reynolds Number ($\times 10^3$)	21.4	21.4	21.4	21.4	31.8
		26.1	26.1		26.1
		31.8	21.4		21.4

CHAPTER 3: RESULTS AND DISCUSSION

Florschuetz Validation Results

Florschuetz represents his experimental results by row average, depending on spatial resolution, and total area average from his copper block technique. For comparison, lateral average data was averaged to compare to his row average experimental results. All data points on the target surface for each test case, obtained using the TSP method, were also averaged to compare to his total area average results. Figure 15 shows the total area average comparison for the two Reynolds number cases tested for the staggered Florschuetz geometry.

Figure 15 – Florschuetz Validation, Total Area Average Comparison

Following the area average data is the lateral averaged data comparison. Each comparison will be made with their respective Reynolds number. In addition to the row average comparison, a plot of the laterally average data is represented.

Figure 16 – Raw Lateral Average UCF TSP Data, Re= 4,961

Figure 17 – Florschuetz Validation, Row Averaged Comparison 1

Figure 18 – Raw Lateral Average UCF TSP Data, $Re=19,857$

Figure 19 – Florschuetz Validation, Row Averaged Comparison 2

It can be visualized in both sets of raw lateral average plots, that peaks and valleys are identifiable due to jet stagnation and wall jet regions. Peaks identify areas where impingement jets are directly hitting on the target surface, whereas, valleys indicate a low heat transfer region between the jets. Once the lateral average data is averaged further for row averaged comparison,

the peaks and valleys are not as identifiable, where, trends for both Reynolds numbers appear to be more linear. From the Figure 17 comparison, both trends appear to be linear until segment number 9, where the Florschuetz data begins to oscillate. This also occurs for the higher Reynolds number case as shown in Figure 19. However, in the higher Reynolds number case, both experimental trends tends to cross more often than in that of the lower Reynolds number case. From the total area average comparison, it can be seen that from both Florschuetz and validation test results, overall Nusselt number increases accordingly with Reynolds number, which can be expected. Overall, the validation test articles tested in this study seems to experience lower heat transfer results as compared to Florschuetz data. From these results, it was concluded as acceptable to move onto the next validation case.

University of Oxford Validation Results

This study will attempt to validate another multi-jet impingement case from literature. This validation case includes smooth and roughened target surfaces, where quantification for heat transfer augmentation is desired. The University of Oxford used a triple thermochromic liquid crystal technique (TLC), where local temperature distribution can be obtained. Furthermore, the data is represented similar to that of Florschuetz, area average Nusselt number and jet row average Nusselt number. For this validation case and from the data acquired from the TSP method, lateral averaged data was averaged for jet row average comparison. Figure 20 represents the total area average data as compared to the University of Oxford data.

Figure 20 – Oxford Validation, Total Area Average Comparison

It can be seen that the validation experiments tested in this study performed higher in heat transfer for all Reynolds number cases. Though area average Nusselts numbers do not match with their respective Reynolds numbers, heat transfer augmentation from smooth to featured seem to be similar. Similar to that of the Florschuetz validation case, we see heat transfer increase accordingly with an increase in Reynolds number for all cases. Heat transfer augmentation of smooth vs. featured is tabulated below in order to compare.

Table 7 – University of Oxford Heat Transfer Augmentation, Smooth to Featured

Feature	Re	Nu Percent Increase
Cylinder	21,065	13.86%
Cylinder	26,017	8.26%
Cylinder	31,851	12.21%

Table 8 – University of Central Florida Heat Transfer Augmentation, Smooth to Featured

Feature	Re	Nu Percent Increase
Cylinder	21,409	11.02%
Cylinder	26,075	9.7%
Cylinder	N/A	N/A

As shown in the heat transfer augmentation data, there are similar percent increases for respective Reynolds numbers. For the lowest Reynolds number case, we see a percent increase above 10% but less than 15%. For the Reynolds number of approximately 26,000, we see a percent increase, in both cases, just below 10%.

A comparison for row averaged Nusselt number was made in order to understand the validation trends further. Below are all the comparisons made with lateral average data as well as jet row averaged data. For each Reynolds number, a plot of UCF lateral average data is represented with imposed row averaged data from the University of Oxford. Also below, for the respective Reynolds number, a graph of row averaged data is represented for trend comparisons.

Figure 21 – Oxford Validation, Jet Row Averaged vs. Lateral Averaged Comparison 1

Figure 22 – Oxford Validation, Jet Row Average Comparison 1

Figure 23 – Oxford Validation, Jet Row Averaged vs. Lateral Averaged Comparison 2

Figure 24 – Oxford Validation, Jet Row Average Comparison 2

Figure 25 – Oxford Validation, Jet Row Averaged vs. Lateral Averaged Comparison 3

Figure 26 – Oxford Validation, Jet Row Average Comparison 3

As expected, the experimental results obtained from this study appear to be higher across all cases tested when compared to the University of Oxford data. Trends obtained from the experimental results for this validation case seem to decrease slightly in the crossflow direction for both smooth and featured. On the contrary, trends seem to increase from that of the University of Oxford in the crossflow direction for smooth and featured. However, the trend for augmentation from smooth to featured is consistent for this validation case.

Repeatability Results

As mentioned earlier, a day to day repeatability experiment was conducted in order to identify any methodology mistakes. The repeatability experiments were conducted using the smooth target surface test section from the University of Oxford validation case. Table 6 lists the sequence of tests that were conducted over a course of 5 days. As mentioned earlier, the testing sequence as well as the occurrence for respective Reynolds numbers experiments is detailed in Table 5. Below are the results obtained from the repeatability experiments.

Figure 27 – Repeatability Experiment Comparison, $Re=21.4k$

Figure 28 – Repeatability Experiment Comparison, $Re=26.1k$

Figure 29 – Repeatability Experiment Comparison, $Re=31.8k$

From the repeatability results, it can be seen that result trends match for every Reynolds number case. There is slight Nusselt number deviation from certain cases, however, it can be concluded that test results can be duplicated on different days using the same experimental procedure.

Current Study Results

The final results come from the geometry configurations described in Table 5. To preface, the results include smooth and roughed target surface test articles for two different Reynolds number cases. There were three different roughened target surface test articles, where, the implemented feature would change. The three different features tested were cylinders,

ellipses, and ovals. Their shape and constraints are described in Tables 5 and 6, as well as their orientation, described in Figure 14. The heat transfer results are represented similar to previous result plots, where, area average and lateral average is of interest. The objective from the results is to understand the augmentation and heat transfer benefit of roughened target surfaces to that of smooth target surface test article. Below are the area average heat transfer results as well as an augmentation comparison table.

Figure 30 – Marc Medina Novelty, Area Average Comparison

Table 9 – Marc Medina Novelty, Heat Transfer Augmentation, Smooth to Featured

Feature	Re	Nu Percent Increase
Cylinder	9,990	11.90%
Cylinder	14,992	5.25%
Ellipse	9,981	11.06%
Ellipse	15,013	3.99%
Oval	10,014	2.51%
Oval	15,023	2.35%

From the area average test results, heat transfer increases accordingly with an increase in Reynolds number. Furthermore, results indicate that cylinders performed best, with the highest heat transfer results for both Reynolds numbers, with a percent increase ranging from 5-12%. Following the cylinder case augmentation, ellipses provide the next best heat transfer results, then followed by oval. None of the featured cases provided lower heat transfer results as compared to a smooth target surface, which is to be expected.

To compliment the area average heat transfer results, lateral average data will also be compared. Below are two plots for lateral average data for the two Reynolds number cases tested.

Figure 31 – Marc Medina Novelty, Lateral Average Comparison 1

In Figure 31, augmentation in heat transfer for implemented features is well represented, where the cylinder trend seems to be the highest. Trends tend to match in the crossflow direction, and tend to decrease as well. Peaks and valleys are well represented, illustrating high heat transfer regions and low heat transfer regions.

Figure 32 – Marc Medina Novelty, Lateral Average Comparison 2

Similar trends are found in Figure 32, where cylinder does appear to perform best. However, the deviation from each case is much smaller than in the lower Reynolds number comparison. Like the previous comparison, trends still match and decrease accordingly along the crossflow direction. From the results obtained in this study, many conclusions and recommendations can be made for continuation of multi-jet configurations.

CHAPTER 4: CONCLUSIONS AND FUTURE WORK

The focus from study was to understand the augmentation of heat transfer by implementing raised surfaces to a target plate for multi-jet impingement configurations. Aside from the focus, many subsidiary goals were also obtained. Some of the secondary goals include; comparison of measurement techniques via validation cases and repeatability for the temperature sensitive paint technique.

From results obtained, there can be some clear conclusions that are apparent across all experiments tested. It can be concluded that heat transfer augmentation does occur with the implementation of raised surfaces. From augmentation data, it can be concluded that an increase of overall heat transfer can rise up to approximately 12% with the use of cylindrical pins as compared to smooth target surfaces. Ellipses and ovals, oriented in the crossflow direction still provide higher heat transfer results as compared to smooth target surfaces. In all experiments tested, results show that an increase in heat transfer can be obtained with an increase in Reynolds number.

From the data comparisons in the validation cases, there still seems to be deviation with heat transfer results. Because the validation attempts were done using a different measurement technique as done from the original experimentalist, it is recommended to investigate the differences in techniques. Because the discrepancies from the validation cases are not apparent or obvious, further studies need to be conducted in order to understand where the problem is. The

repeatability experiments conducted in this study proves as a good starting point, for temperature sensitive paint, to understand duplication of results by varying the day-to-day test schedule.

As mentioned in Chapter 1, multi-jet configurations can become complex depending on the application and objective. Furthermore, multi-jet geometries can become creative by varying the hole arrangement, impingement channel, and changing the non-dimensional parameters. Because this study only focused on the heat transfer results for different multi-jet configurations, it is recommended to also understand the flow field within the impingement channel, where, more useful information can be utilized for comparisons and conclusions. To understand the flow field for impingement cases, one can conduct several experiments or setup different numerical domains. More specifically, a recommendation to understand flow fields for roughened surfaces is of value, where most of the flow field studies have focused on smooth target surfaces for multi-jet impingement applications. As mentioned in Chapter 1, an increase in heat transfer for cooling schemes may not always be the objective. In some applications, pressure drop or redirection of coolant within turbine geometries may be of interest instead of augmentation of heat transfer.

Some of the final conclusions and recommendations stem from the heat transfer results represented in Figures 31 and 32. The challenge for multi-jet impingement is alter the impingement channel in such a way that benefits both the flow structures and heat transfer characteristics. Specifically about heat transfer, the challenge is to alter the channel so that there provides uniform heat transfer across the entire target surface. This can be accomplished by implementing unique features on the target surface, that won't be detrimental to the flow

structures, but still provide efficient heat transfer results. Some parameters that can be varied include blockage of feature, feature density within the impingement channel, shape of the feature, location of the feature, and orientation of the feature to crossflow.

Overall, the study on multi-jet impingement still needs to be continued, where characterizing the cooling scheme is still being defined while the demand for such cooling is still high for turbine geometries. Optimization of multi-jet configurations are still being discovered one experiment at a time, where challenges to meet the demands of fluid flow and heat transfer is endless. By continuing the work for multi-jet impingement configurations, the hope is to find unique geometries that may be of practical use for turbine industry standards and limitations.

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