HEAT-FLUX MEASUREMENTS FOR A REALISTIC COOLING HOLE PATTERN AND DIFFERENT FLOW CONDITIONS

Thesis

Presented in Partial Fulfillment of the Requirements for the Degree Master of Science in the Graduate School of The Ohio State University

By

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2011

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Abstract

For many years turbine designers have utilized advancements in film-cooling technology to allow for increased high-pressure turbine inlet temperature. Prediction tools, used to predict the cooling effectiveness of the representative cooling-hole and cooling-hole pattern designs have been successful in keeping the engines on wing for a large number of operational hours, but there is room for and a desire for improvement in the technology. Therefore, a study was undertaken at the OSU GTL to find a way to obtain basic data needed to help improve CFD prediction capability. The particular facility utilized for this work is a medium-duration blowdown facility to which significant improvements in the operational procedure of the cooling system have been made for the purposes of this work and both the facility and the improvements will be described in detail in this thesis.

In order to keep the CFD validation simple, a flat plate configuration with a realistic cooling hole pattern, representative of a high-pressure turbine blade for which measurements obtained as part of a full-stage experiment were obtained, was utilized along with flow properties of current interest to the industry. The measurements reported in this thesis yielded high response heat-flux measurements along the axial direction of the plate, including locations between the individual rows of cooling holes. The influence of Reynolds numbers on heat transfer to the plate was also explored. Lastly, the temperature of the main flow and the test section walls were varied to determine the effect of cooling on the local adiabatic wall temperature.
DEDICATION

Dedicated to my loving and supportive family,

For the joy of learning and will to succeed that you have given me.
ACKNOWLEDGMENTS

It is an honor to have been a part of the great group of people that make up the Gas Turbine Laboratory at The Ohio State University. Very few institutions can fulfill both the research interests of industry and that of pure academia with such success. I would first like to thank my advisor, Dr. Michael Dunn, not only for the opportunity, but also for his dedication to the laboratory and everyone in it. I am very grateful to Dr. Charles Haldeman for his patience and direction when assisting me with this research. Also, I would like to show my gratitude to Dr. Randall Mathison who was always around to assist with many aspects of this work, from experimental to computational and data analysis.

The Gas Turbine Laboratory would not be the fine facility that it is without all of its supporting members. I would like to thank Mr. Jeff Barton, Mr. Ken Copley, Mr. Ken Fout, and Dr. Igor Ilyin for their support in instrumentation, design, and machining all the necessary components for the experimental facilities. Also, I would like to thank Mrs. Cathy Mitchell for her help with all the administrative responsibilities at the AARL.

I would like to thank all the previous and current GTL graduate students whose previous thesis accomplishments have been beneficial to the success of this thesis. A special thank you goes out to Mr. Mitchell Parsons, with whom I shared the defeats and successes at the GTL. Also, to the other students at the AARL including Mike Crawley, Cameron DuBois, Casey Hahn, Chris Jensen, Martin Kearney-Fischer, and Kevin Yugulis: I appreciate your support and friendship. I can honestly say that
without this brilliant and unique collection of people my thesis would not have been possible.

Finally, I would like to thank Honeywell for their ongoing support in research at this facility at the GTL.
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<th>Description</th>
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<tr>
<td>AARL</td>
<td>Aeronautical &amp; Astronautical Research Laboratories</td>
</tr>
<tr>
<td>BLB</td>
<td>Boundary Layer Bleed</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>DAS</td>
<td>Data Acquisition System</td>
</tr>
<tr>
<td>FAV</td>
<td>Fast Acting Valve</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
</tr>
<tr>
<td>GTL</td>
<td>Gas Turbine Laboratory</td>
</tr>
<tr>
<td>HFG</td>
<td>Heat-Flux Gauge</td>
</tr>
<tr>
<td>HPT</td>
<td>High-Pressure Turbine</td>
</tr>
<tr>
<td>K</td>
<td>Degrees Kelvin</td>
</tr>
<tr>
<td>OSU</td>
<td>The Ohio State University</td>
</tr>
<tr>
<td>PSI</td>
<td>Pounds per Square Inch</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds Number</td>
</tr>
<tr>
<td>RTD</td>
<td>Resistive Thermal Device</td>
</tr>
<tr>
<td>SCF</td>
<td>Small Calibration Facility</td>
</tr>
<tr>
<td>St</td>
<td>Stanton Number</td>
</tr>
<tr>
<td>TC</td>
<td>Thermal Couple</td>
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</table>

\[ \dot{m} \] Mass Flow Rate  
\[ C_p \] Specific Heat  
\[ T \] Temperature  
\[ T_{Wall} \] Test Section Wall Temperature  
\[ T_{Main} \] Main Flow Fluid Temperature  
\[ T_{HexCool} \] Cooling Flow Temperature at the Heat Exchanger  
\[ h \] Convective Heat Transfer Coefficient  
\[ \rho \] Density
Chapter 1 Introduction

The current interest within the gas turbine industry to increase the efficiency of its engines to meet the demand of the aviation industry carries with it a significant increase in combustor exit temperature. Obviously, increasing fuel efficiency results in decreased operating costs for the user especially during these times of escalating fuel costs. The implementation of this desired increase in operating efficiency is restricted by the material limitations of the hot section components of the engine. Those components primarily influenced are the combustor and the high-pressure turbine vanes and blades, which are directly exposed to the hot airflow. Several new technologies have come into existence in an attempt to deal with this issue. Alternative materials alloys and various blade coating materials and processes have proven a key part of solving the problem.

An opportunity for potentially significant improvements in increased efficiency of the machine involves increased capability of turbine airfoil film cooling. This approach entails bleeding relatively cool air from the high-pressure compressor and directing it through interior passageways within the airfoils as well as through effectively placed cooling holes on the surface of the airfoils. The airfoil cooling holes provide a method for adding cooler fluid flow into the local boundary layer. The industry has effectively utilized film cooling for many years and has been successful in protecting airfoils sufficiently well as to allow combustor exit temperatures to far exceed those imaginable even 10-years ago.
Bleeding air from the early stages of the compressor allows increased efficiency of the turbine but decreases the mass of high-pressure air that the compressor delivers to the combustor and thus a penalty must be paid. The design of cooling hole configuration and location on a turbine airfoil is a balancing act designed to maximize the cooling effects while using the least about of bleed mass flow. This critical design aspect of turbine airfoils is limited by current heat transfer predictive abilities for these very complex flow field environments. The current heat transfer prediction methods used by industry work reasonably well as demonstrated by the relatively long life time on wing of present day commercial engines, but there is room for improvement and the interest in doing so is present within the industry.
During the past 70 or more years there have been literally thousands of papers published in the open literature dealing with film cooling for which the experimental configuration of choice was a flat-plate or a cascade and these papers are referenced in Kercher [6, 7]. The obvious question is then "Why in the world would anyone want to perform another flat-plate film cooling experiment?" The answer to the question is "Only if one designs an experiment that is very different than those that have been performed in the past in order to significantly improve the applicability of the results to the real-world problem of film cooling as seen through the eyes of industry." It is felt that this has been done with this thesis for which the primary goal is to provide a quality data set using a realistic cooling-hole pattern so as to help in the process of significantly improving CFD validation. With this goal in mind, the cooling hole pattern (number of rows of holes = 5, number of holes = 110, several different diameter of holes dependent upon specific location (see Table 1), and spacing of holes that is again dependent upon relative location) consistent with the pressure surface of a cooled high-pressure turbine blade has been reproduced on a flat plate and measurements performed for a range of flow conditions representative of parameters of interest to the engine designer. In doing so, the results of these measurements can be used for CFD validation and they can be compared directly to results obtained using real engine hardware (from which the cooling hole configuration was copied) for the pressure surface of the blade of a fully-cooled turbine operating at design corrected conditions. In performing this comparison, one must be conscious of the fact that the fluid dynamics associated with a flat plate (or a cascade for that matter) in no way represents the fluid dynamics for the pressure surface of a transonic turbine blade. However, it is these very flat plate and cascade experimental results that industry designers have used for many years to successfully design the film cooling hole patterns and distributions for these airfoils. The intent here is to provide yet another step in the prediction improvement ladder so that the current demands of
improved designs can be achieved.

1.1 BACKGROUND

A significant amount of time, ingenuity, and financial support has gone into improving prediction tools and testing designs to ascertain the most effective film cooling for the use in gas turbine engines. As noted in the Introduction, a plethora of journal papers and technical reports exist that provide the pathway to current film-cooling technologies, the most critical of which will be noted here. Dunn [8] provides a thorough discussion of the current and future issues the gas turbine industry faces, which include the intricate connection between computational fluid dynamics (CFD) progress and design tool gains.

Crawford [9, 10] completed a flat plate experimental and computational study in 1980. This yielded comparison data initially for the CFD code STAN5 developed at Stanford and for the improved code STANCOOL developed by Crawford while he was at the University of Texas at Austin. STANCOOL is currently used throughout the industry to help in the development of new turbine blade cooling designs. In the time since this study, efforts have been made to better understand the factors on which cooling effectiveness depends, for example the effects of turbulent flow, density ratio, blowing ratio, etc.

Likewise, studies have been conducted to advance film-cooling design guidelines for the current process. Studies have shown slanted holes produce better results than normal holes, and that staggered rows of holes protect the surface better than aligned holes. Also studies discovered that high cooling to main flow mass-flow ratio, known as the blowing ratio, compared to a medium or low blowing ratio seems to decrease efficiency of the cooling fluid in mitigating heat transfer. This is due to detachment...
of the cooling flow from the boundary layer close to the surface, allowing hotter main flow gas access to the surface of the metal.

Though there have been significant advancements in the area of film cooling, with the increased demands of current interest there is still a need for improvement. When surface heat-flux data obtained on the airfoils of a fully-cooled high-pressure turbine stage operating at design corrected conditions were obtained at OSU, the ensuing analysis of the data showed improvement in the prediction capability, but indicated specific regions where improvement could be helpful. The work reported herein was designed to provide a database that could be used in conjunction with various predictive codes to help improve the prediction capability.

1.2 Scope of the Current Study

This document serves as a culmination of over 5 years of graduate student research and achievements in the development of the Small Calibration Facility (SCF) at the Gas Turbine Laboratory (GTL) for use as an experimental tool to obtain film-cooling data of interest to the industry. The SCF is fundamentally a relatively small blowdown facility of medium duration that can be used to perform relevant fundamental studies. A variety of different tasks have been completed using this facility e.g., measurement of the recovery factor for total temperature and total pressure probes used as inlet and exit rakes for the full-scale turbine hardware typically run in the OSU very large Turbine Test Facility (TTF), both time-averaged and time-accurate heat-flux measurements for a flat-plate environment, measurements of turbulence intensity, and the measurements reported here. The work of past graduate students will be summarized throughout this document along with the most recent facility improvements that have been made to advance the cooled heat-transfer experimental techniques at the GTL. Many of the techniques developed here have migrated to the
As mentioned in the Introduction, if one is going to perform a flat-plate film cooling experimental program, it had better be something really different from the thousands of previous flat-plate film cooled studies reported over the past 70 or more years. The current work focused on using a very realistic geometric arrangement of film cooling holes to create a set of experiments to highlight the collective results of the effects of blowing ratio and initial test conditions on heat transfer distribution among and downstream of the five rows of cooling holes (for this particular geometry). Care has been taken to make it possible in future work to compare the data sets reported here with the previous work of Crawford [9, 10], which represents the data set that is used in most design work today.

The results presented in this thesis suggest that low blowing ratios produced the most effective cooling film in regions nearest the cooling holes compared with results of higher blowing ratios in the same region. Exploration shows cooling differences among ingestion into the cooling cavity, nominal cooling, low cooling, and high cooling runs.

Experiments with no cooling, room temperature cooling gas, and cold cooling gas illustrated the heat transfer differences for each scenario. The experiment draws strong parallels to Crawford’s cooled plate blowdown work completed in 1980 while he was doing his Ph.D. dissertation at Stanford University. The outgrowth of that work was a predictive code known as STAN5, which is still in use today within the industry.

Finally, this study includes the results of blowdown experiments using the SCF for which the adiabatic wall temperature was successfully measured. Experiments were conducted with various wall and main flow temperatures. Comparisons between
these experiments and the results from the room temperature test section experiments showed that changes in heat flux on the plate from different runs clearly trended with the temperature of the test section walls in addition to the main flow total temperature. These two temperature conditions make up the driving temperature for each experiment. Comparing this driving temperature to the heat flux for several runs uncovered the dependence of heat flux not only on the driving temperature, but also on the different Reynolds number associated with not only different run conditions, but also within a given run.
CHAPTER 2 DEVELOPMENT OF THE SMALL CALIBRATION
FACILITY FOR USE AS A FILM COOLING RESEARCH
TOOL

This chapter addresses the many facets of the facility used to conduct the experiments discussed in this thesis. The Small Calibration Facility (SCF) at The Ohio State University Gas Turbine Laboratory (OSU GTL) is a medium duration transonic blowdown facility. An associated cooling system blowdown technique specifically designed to eliminate the temperature spike associated with the starting process of the cooling system has enhanced the facility and made the current experiments possible. A portion of this chapter covers specifics of the test section configuration, which makes these experiments unique from other flat-plate experiments. Another portion lends insight into the problems associated with incorporating this new cooling system into the blowdown facility.

Fluid flow from a finite volume reservoir of high-pressure air to a low-pressure dump tank through a choked orifice is the configuration of the blowdown facility. In this type of facility the total pressure and total temperature of the working fluid decay with time, but can be well modeled as a function of time and initial conditions. The rate of decay of the total pressure and temperature is primarily dictated by the ratio of the volume of gas that can pass through the specific choke per unit time to the volume of the supply tank. However, the ratio of the test section area to the choke area dictates the test section Mach number, as described by the area-Mach number relation given in Equation 2.1 [11].
\[
\left( \frac{A}{A^*} \right)^2 = \frac{1}{M^2} \left[ \frac{2}{\gamma + 1} \left( 1 + \frac{\gamma - 1}{2} M^2 \right)^{\frac{\gamma + 1}{\gamma - 1}} \right]
\]  \tag{2.1}

The length of time for experiments performed using this facility is on the order of 1.5 to 3 seconds. The term medium duration refers to the fact that during the useful test time of the facility, the system is still relatively isothermal (consistent with what one sees for a short-duration facility), but unlike the short-duration facility, the inlet conditions for the medium-duration facility are changing. Because of the longer times associated with a medium-duration facility by contrast with a short-duration facility, one must be conscious of time-scales of the instrumentation as well as the influence of changing Reynolds number. The variation in Reynolds number during a single experiment can be exploited in a medium-duration facility to produce a more detailed experimental database, without adding more experiments. Simple changes in chokes and initial conditions yield a wide range of test conditions relatively inexpensively.

2.1 HISTORY AND OVERVIEW

The SCF was a part of the original transition of some of the current GTL facilities and personnel from CALSPAN to the OSU GTL in 1996. This transition included the TTF, a very large short-duration blowdown facility that is described further in Chapter 6. As previously note, in past years the SCF has been used to calibrate total pressure and temperature rakes prior to installation in the TTF, provide a platform for new instrumentation development, and to examine unsteady cooling effects through the use of a pulsating device that was designed to represent the unsteady flow experienced by a blade passing through successive vane wakes. In 2006 Bernasconi [2] completed an upgrade to the original pulsed cooling experiment through the design of a new test section, which allowed for finer resolution of the heat-flux data downstream of a cooling pattern that was more representative of the
typical turbine blade, and an updated cooling system.

Upon its arrival at the GTL, the SCF included a main supply tank and large dump tank connected to a test section via an upstream air powered fast acting main valve and an interchangeable choke downstream of the test section. This choke allows one to set the test section Mach number at a given value. Figure 2.1 shows a schematic of the SCF prior to installation of the cooling system.

![Figure 2.1: Schematic of the SCF before Installation of the Cooling System](image)

The flow path geometry for the gas leaving the main supply tank was modified from circular to rectangular at the test section and then to circular as the gas flows to the large dump tank. The main supply tank is supplied with dry air and has a mixing fan installed in the ceiling to ensure uniform temperature throughout the main supply fluid. Coil heaters on the exterior circumference of the tank connect to variable voltage transformers designed to control the supply fluid temperature. Fiberglass insulation surrounds the entire tank in order to maintain internal gas temperatures.
The facility has undergone several updates including an updated flat plate design based on the work of Bernasconi [2] (Section 2.2), an updated cooling system (Section 2.3) and improved instrumentation (Section 2.4). Combination of these upgrades allowed the development of a matched blowdown facility (Section 2.5). The resulting operations of the new facility are described in Section 2.6.

2.2 The Film-Cooled Flat Plate

The test-section used in the experiments for this research developed by Bernasconi [2] consists of 4 essential pieces: the boundary layer bleed plate, cooling hole plate holder, cooling hole plate, and an uncooled follower plate. The cooled flat plate is stainless steel and inserts into a holder in the test section between a boundary layer bleed wedge (BLB) and a following flat plate with no cooling holes. The separator is designed to house the cooled plate and allow an exit for instrumentation to connect to electronic equipment outside the test section. This creates a cooling plenum behind which the cooled fluid is fed into the back of the plate and to the cooling holes. This can best be understood using Figure 2.2 and Figure 2.3, which show the cross-section of the new test section and the CAD drawing of the cooling plate and holder.

![Figure 2.2: Schematic Cross-Section of the SCF Test Section [2]](image-url)
As previously noted, the cooling hole pattern, row and hole spacing, diameter, and number are representative of an industry designed cooling pattern for the pressure side of a turbine blade airfoil. The relative location of the double-sided Kapton heat-flux gauge instrumentation and the cooling hole geometry for the instrumented plate is shown in Figure 2.4. This cooling hole pattern (D, E, F, G, & H in yellow) is consistent with that of the pressure surface of a high-pressure turbine (HPT) blade used for the measurement program reported in Haldeman et al. [12, 13, 14, 15, 16, 17, 18].

Figure 2.3: CAD Representation of Cooled Flat Plate and Plate Holder [2]
Figure 2.4 shows the instrumentation locations on the plate along with the numbering scheme used for them. The cooling pattern consists of five rows of holes and Table 2.1 gives the number and diameter of the holes in each row. This illustrates one novelty of the flat plate experiments reported in this thesis in that most other flat plate papers (some previously described) have only one or two rows of uniformly spaced rows and holes.
Table 2.1: Description of Cooling Rows on the Cooled Flat Plate

<table>
<thead>
<tr>
<th>Row</th>
<th>No. of Holes</th>
<th>Diameter of Holes (in.)</th>
<th>Area per Hole (in.)</th>
<th>Area per Row (in.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>D</td>
<td>16</td>
<td>0.018</td>
<td>2.54 \cdot 10^{-4}</td>
<td>4.072 \cdot 10^{-3}</td>
</tr>
<tr>
<td>E</td>
<td>15</td>
<td>0.018</td>
<td>2.54 \cdot 10^{-4}</td>
<td>3.817 \cdot 10^{-3}</td>
</tr>
<tr>
<td>F</td>
<td>32</td>
<td>0.012</td>
<td>1.13 \cdot 10^{-4}</td>
<td>3.619 \cdot 10^{-3}</td>
</tr>
<tr>
<td>G</td>
<td>31</td>
<td>0.016</td>
<td>2.01 \cdot 10^{-4}</td>
<td>6.233 \cdot 10^{-3}</td>
</tr>
<tr>
<td>H</td>
<td>16</td>
<td>0.014</td>
<td>1.54 \cdot 10^{-4}</td>
<td>2.463 \cdot 10^{-3}</td>
</tr>
</tbody>
</table>

All of the holes have a slant angle of 30° opposite the direction of the flow. The instrumented plate has a thickness of 0.100-in. and is installed onto the cooling plate holder block. This aluminum holder block is designed to evenly dispense heat through the cooling plate. This is critical for the adiabatic wall temperature experiments where the heaters are placed on the backside of the flat plate holder in the test section. Once they are turned-on the highly conductive aluminum distributes the temperature gradient evenly.

The heat-flux gauge instrumentation can be seen in Figures 2.4 and 2.5. Each number corresponds to two gauges, a top and a bottom, denoted throughout this
work as "#T" or "#B". The flat plate includes a total of 30 double-sided 0.002-in. thick Kapton with nickel heat-flux gauges identically aligned on each side (hence top and bottom) with significantly improved connecting wiring compared with the wiring technique that was used in previous SCF studies. In addition to these new "4th generation heat-flux gauges", the un-cooled plate placed behind the new cooled plate includes the previous generation of 0.001-in. thick Kapton double-sided heat-flux gauges. The heat-flux gauge instrumentation will be further described in Section 2.6.

2.3 COOLING SYSTEM UPDATE

2.3.1 INITIAL COOLING SYSTEM

A sketch of the initial cooling system design employed to supply cold cooling gas flow to the reservoir feeding the holes in the flat plate is shown in Figure 2.6. A secondary supply tank was added and supplied with nitrogen from a K-bottle. Nitrogen was chosen because dry nitrogen could easily be obtained and thus protect the delicate heat-flux gauge instrumentation from ice buildup. A manual valve and fixed choke was added into the plumbing after the supply bottle that led to a unique heat exchanger. The flow after the heat exchanger was directed through half-inch tubing to the cooling plenum of the flat plate.
A more thorough description of this system can be found in Kheniser [3].

2.3.2 Problems with the Cooling System

Upon completion of the cooling system assembly, experiments were performed that revealed issues with the blowdown matching techniques, the accuracy of the heat-flux gauge instrumentation, and the minimum temperature the cooling fluid could reach prior to injection into the cooling plenum. Kheniser [3] explains the issues with the heat-flux measurements experienced during his work and some techniques used to mitigate them. These problems included identical runs showing a high variance in temperature read by each gauge, and cooled-run gauge temperatures that were indistinguishable from uncooled runs. Additionally, the heat-flux gauges showed extreme delicacy resulting in loss of many data points across the flat plate and complete loss
of instrumentation in the interesting region between the cooling rows.

These complications were resolved by appropriate data processing, namely an error in the blowing ratio calculation was corrected, which led to the proper sample times being later than expected. Once samples were taken in the appropriate window with reference to the development of the cooling flow, the aforementioned concerns resolved themselves. The delay in the cooling flow development provided blowing ratios much too high to discover many relevant clues about the heat-transfer properties of the system.

In addition to the problems associated with the heat flux gauges, problems with the initial cooling system design were encountered. A compression heating temperature spike in the cooling system occurred when the valves opened to begin the main flow as shown in Figure 2.7.
This temperature spike occurred due to length of time the cooling system took to match the pressure produced by the start of the main flow and resulted in significantly higher cooling gas temperature than desired. The other main problem was that as a result of this compression of the gas a constant blowing ratio was never reached. This is exhibited in Figure 2.8.
Figure 2.8: Pressure Decay from Original System [3]

The respective pressure histories shown in Figure 2.8 made comparing data from separate runs next to impossible since the blowing ratio in each run changed dramatically throughout the run.

2.4 4th Generation Heat-Flux Gauges

The previously used flat plate was re-instrumented with the fourth generation of double-sided Kapton heat-flux gauges for the purposes of the measurements reported herein. The basic concept for these thin-film double-sided high response heat-flux gauges originated from a design initially reported by Epstein et al. [19]. This particular gauge design is important for film-cooling studies because it is essential to have a real-time temperature measurement on both sides of the gauge in order to effectively
use the one-dimensional heat conduction equation for analysis of the data. In this regard, it is extremely important that the gauge on the top surface of the Kapton film be identically aligned with its partner on the opposite side of the film. An explanation of the heat flux gauges used for the experiments contained in this document as well as a brief summary of the various steps taken at the GTL to significantly improve the performance and useful lifetime of these gauges follows.

2.4.1 Overview of HFG Technology at the OSU GTL

This experimental data set provides an insight into the true usefulness of the Kapton HFGs. The thin film onto which the nickel gauges are deposited for these measurements has a thickness of 0.002-in. and provide essentially no disturbance in the boundary layer flow. The double-sided gauges act as two separate temperature instruments directly aligned within one unit as shown in the schematic of Figure 2.9.

![Figure 2.9: Schematic of Double-Sided Heat-Flux Gauge][3](image)

The sensing element of this heat-flux gauge is made of nickel and the lead wires are made of copper. Each temperature gauge, top and bottom, have a known resistance determined on the basis of calibration measurements performed at individual measured temperature; this produces one of the calibration necessities of the gauges. The thermodynamic properties of the insulating layer provide the necessary information for heat-flux reduction from the two temperature measurements via the
one-dimensional heat conduction equation Weaver et al. [20].

2.4.2 Improvement in the 4th Iteration Kapton Heat-Flux Gauges

The most significant improvement associated with the application of the heat-flux gauges to the plate involved replacing the silver connections with copper wire. This process succeeded in making the gauges much more durable. The gauges used for the experiments reported here (see Figure 2.10) were installed and used for a set of 11-runs approximately one year prior to the main set of experimental runs described in this work. The HFGs did not receive any special treatment during that time and remained installed in the test rig at room temperature and pressure. A complete set of over 50 new runs was executed using these gauges over a year later, including runs with the heated test section wall, with minimal loses in instrumentation. The heated adiabatic wall temperature runs had no negative effects on the performance of the gauges. In the case of the gauge installation used previously, the heaters in the wall of the test section had caused a significant loss and damage to the gauges.

Figure 2.10: Photograph of the Kapton HFG’s [3]
2.4.3 Calibration of HFGs

Each of the heat-flux gauges undergoes two important calibration steps. The first calibration occurs prior to installation of the instrumented object into the test section. In recent research on the detailed behavior of these gauges, Hodak [21] determined a resistance-based calibration was necessary to maintain low offset errors among gauges. This calibration created correction factors for the voltages from each individual amplifier. Previously, only the voltage calibrations were completed. A detailed description of the optimization process for these calibrations is documented in Hodak [21] and lies outside the scope of this document. The set of gauges used for the work reported herein was calibrated using an oven heating technique to warm the instrumented plate and the RTD’s (accurate to 0.02 K) mounted on that plate. Reducing the data from this process yields both a linear and quadratic fit for the resistance as a function of temperature and the temperature as a function of resistance equations. The quadratic fits were used to determine the final calibration coefficients.

The second calibration step is crucial to understanding the attention to detail necessary to use a HFG-based system of instrumentation. This step occurs once the flat plate is installed in the test section. Using resistors of known value and the known temperature of the test section metal from one of the RTD devices noted in the previous paragraph, a series of measurements are taken through the DAS system. These tasks are used to calculate the line resistance, as well as all amplifier characteristics for each individual gauge. The offset, gain, and line resistance is installed into the processing file for the gauge measurements and accounted for in the calculation of the temperature. Once this calibration is complete one has a temperature measurement accurate to 0.2 K for each of the HFG’s.
2.4.4 Outside Dependancy of HFG’s

Although these initial measurements show a high level of accuracy, the dependence of the accuracy to the resulting measurements on the entire electronic circuit, of which they are a part, have become apparent. The amplifiers used in conjunction with the heat-flux gauges are independent constant current input amplifier (CCIA) boards designed and manufactured at the GTL. These units are wide bandwidth, low noise devices that allow for a known voltage input, offsets, and gains to be utilized and accounted for during the calibration processing. Since each heat flux gauge has its own amplifier and individual set of calibrations, the calibration process is quite involved and requires attention to detail. A number of factors go into the calibration coefficients of the heat-flux gauges, including a constant for the line resistance that changes as a function of temperature. Coping with the issue of drifting line resistance, in order to resolve heat flux, is discussed in the subsequent chapter.

2.5 Developing a Matched Medium-Duration Cooling Facility

In order to properly understand the operational issues associated with this cooled facility some methodology specific to all blowdown facilities needs explanation. A general description of how blowdown facilities operate can be found in Epstein [22]. The primary isentropic blowdown equations are reproduced in Appendix A. These equations are given in Equations 2.2, 2.3, and 2.5 and show that the total pressure, total temperature, and density are a function only of the initial conditions, the time from the start of the blowdown (t), and a supply tank time constant (τ). For a facility that uses cooling flows to maintain relatively constant conditions at the test article, the two different parts of the facility (the main flow and the cooling flow) need to be configured so that they have similar time constants (τ).
\[ P_0(t) = \left[ P_0(t=0) \right] \left( 1 + \frac{t}{\tau} \right)^{\frac{\gamma}{\gamma-1}} \] (2.2)

\[ T_0(t) = \left[ T_0(t=0) \right] \left( 1 + \frac{t}{\tau} \right)^{-2} \] (2.3)

\[ \rho = \frac{P}{RT} \] (2.4)

### 2.5.1 Matching Blowdown Times

To obtain relevant data with the inclusion of a cooling system in the SCF required matching the two blowdown processes in order to control the cooling conditions. A number of variables go into characterizing a blowdown matching experiment, one of the most crucial being the blowing ratio (BR), which is a function of the velocity and density of the two fluid flows.

\[ BR = \frac{\rho_c U_c}{\rho_\infty U_\infty} \] (2.5)

Section 2.3.1 discussed the initial experiments for which the blowing ratio reached the desired point for only a few milliseconds and then continued to change. This undesirable trait made the conditions of each run very difficult to predict and control.

Facilities that produce a steady-state experimental system for an extended period of time can make short work of controlling this relation, but due to the short-duration nature of the SCF this is more difficult. The basic problem with this particular facility is that the pressure and temperature decay of the two processes are independent. In order to maintain a constant blowing ratio, these decays must be matched. Accomplishing this task opened the door to measuring heat flux at various Reynolds numbers in the time domain of one experiment. The system has this
unique ability as a result of the very thing that makes it difficult: the pressure decays through each sub-system. Since the Reynolds number changes throughout each individual experiment with time, this provides an abundance of data in a very short experimental window: one of the main advantages of the SCF. The details of how a constant blowing ratio was achieved will be discussed further in the following section.

2.5.2 Predictive Tools

The challenges of the short-duration experiment with so many initial conditions led to the need for a prediction tool to find the initial settings, specifically valve timings and choke areas, for any desired run conditions. A LabVIEW program was developed to fill this need. The solution requires a number of initial conditions including the volume of each component of the system, initial temperature of the main tank fluid, the cooling fluid temperature as it reaches the cooling plenum, and the initial pressures of both the main and cooling supply tanks. Since these temperatures and pressures cannot be known exactly prior to running the experiment, values close to the expected ones are used to produce anticipated results from the program. After the experiments have run, the actual initial temperatures and pressures determine the effective areas in the various valve positions, and the blowing ratio can be calculated using the program. Figure 2.11 is a schematic that shows how the program handles the breakup of the components of the facility. Each section is treated as a mass and energy balance-conservation problem.
2.6 **CURRENT SCF BEHAVIOR**

The interaction of the pressure ratios at the cooling holes has been completely reworked. Figure 2.12 illustrates the dramatic difference between the time at which fully developed cooling flow occurred in the previous experimental work (Figure 2.12a) and this work (Figure 2.12b).
Figure 2.12: (a) Previous Pressure Comparison (b) New Cooling System Pressure Comparison

The new system allows the pressure in the cooling plenum to build up completely prior to initiation of the main flow. This process eliminates the time for the cooling flow to adjust to the pressure spike in the main flow and helps stabilize the blowing ratio.

Figure 2.13 is a photograph of the SCF illustrating the main supply tank, test section, heat exchanger (housed within the insulated cooler chest), and the dump tank.
Figure 2.14 represents a schematic of the new and improved cooling system that was implemented for the first time on this set of experiments. The colors in the figure correspond roughly with the temperature of the fluid in that section where blue represents cooling flow (208-265K), pink represents room temperature, and red represents heated main flow (475-400K).
This system reduces the problems associated with matching two blowdown processes and trying to maintain a steady blowing ratio. A needle valve (A) and an ON-OFF valve (1) have been installed upstream of the heat exchanger. This allows for two critical things to happen. The needle valve allows for complete control over the mass flow of cooled fluid to the cooling plenum, a range in area from 0.0021 in\(^2\) (the fixed area of the small choke) to 0.03748 in\(^2\). Therefore, the blowing ratio can be controlled via the mass flow relation of the two blowdown processes. The ON-OFF valve in combination with the needle valve allows for elimination of the temperature spike by controlling the rate at which the pressure increase in the cooling plenum behind the cooling holes is allowed to occur and by leading the experiment with cooling gas flowing initially, followed by triggering the main flow. A complete description of the compression heating spike issue is documented in Boehler [4]. Figure 2.15 is a photograph of the new cooling system installed on the SCF to facilitate the measurements reported herein.
Two needle valves (B & C) and an ON-OFF valve (3) were added after the cooling plenum to serve as the blowdown-matching component of the system. The area can be controlled from closed ($0 \text{ in}^2$) to $0.0716 \text{ in}^2$ using the two needle valves. When valve 3 is opened it pulls some of the cooling fluid directly to the evacuated dump tank, allowing the cooling flow to decay at the same rate as the main flow and help maintain a constant blowing ratio. The new system helps significantly with the constant blowing ratio problem. The valve timing can be adjusted easily to align the shut off opening times with the blowdown of the main flow.

This unique cooling system setup makes it much more difficult to determine the total areas and total mass flow. Since the blowdown matching section pulls some of
the cooling fluid out of the cooling plenum before it is injected into the main flow, the total mass flow through the choke in the cooling system is greater than the total mass flow through the cooling holes. In order to determine the blowing ratio, one needs the total mass flow through the cooling holes. This, in turn, complicates solving for the Stanton number, or normalized heat flux. A later section discusses the process and methodology for accomplishing this task.

Instrumentation in this facility includes the necessary pressure and temperature sensors needed to calculate mass flow, Reynolds number, and therefore Stanton number. Figure 2.16 shows the additional instrumentation on the SCF that is used for this purpose.
Figure 2.16: Instrumentation on the SCF

Figure 2.16 requires some explanation. The five RTDs in the main supply tank were used to determine the main supply temperature. During the run, the main flow total temperature was gathered from the thermocouple located on a rake in the test section. This rake also contained a Kulite pressure transducer used to obtain a measure of the total pressure.

All the pressure transducers in the system are calibrated over the full range of pressures that the facility encounters, from 0 to 50 PSI. Each transducer is calibrated to a calibrated (National Bureau of Standard specifications) Heise pressure transducer that is located in the main supply tank. Upon completion of these calibrations the pressure transducers match one another to within fractions of a PSI. The RTDs in the facility are also calibrated to within 0.2 K of one another, which is the accuracy of each sensor.

This experimental work contains data from cooled runs with a cooling flow at least 20 K colder than the previously reported cooled runs of Kheniser [3]. A
much lower temperature was reached by eliminating heat gained by the fluid from the stainless steel tubing between the heat exchanger and the cooling plenum. Figure 2.17 is a photograph showing the internal components for the cooled heat exchanger and Figure 2.18 is a photograph of the current insulated plumbing.

Figure 2.17: Cooled Heat Exchanger
Previously, the cooling flow would reach a temperature of approximately 208 K in the heat exchanger, but by the time the cooling gas reached the cooling plenum it was at a temperature of 280-300 K (depending on the room temperature in the experimental facility). Insulating and cooling the connecting braided stainless steel tube by surrounding it with dry ice helped realize the new, lower cooling fluid temperature. The GTL has also conducted a study of cooling behavior, which shows promise in
future cooling fluid reaching a temperature of 160 K in the heat exchanger, and 230 K in the cooling plenum. The system envisioned to achieve these levels is described in Parsons [23]. Minimizing the difference between the ratios of the cooling flow temperature to the main flow temperature in the test facility with that of operating conditions ensures relevance of test data in design work and increases the likelihood of determining the correct defining characteristics of film-cooling effectiveness.
CHAPTER 3 DATA ANALYSIS & RESULTS

The results from this experimental work are organized around three long-standing goals. One of which is to discover the important characteristics of a cooling system for a facility to gather heat-flux data on a cooled flat plate. This has previously been discussed. The second is to discover if differences in heat-flux for representative flow conditions can be effectively resolved in this facility. And thirdly, can Reynolds number effects on the driving temperature be resolved in order to correct the adiabatic wall temperature information from the data. In the process of creating the data set described herein, the latter two issues are also addressed.

3.1 DESCRIPTION OF EXPERIMENTAL MATRIX

3.1.1 EXPERIMENTAL SEQUENCE

The main supply tank heater coils were activated a day prior to the time the experiments were scheduled to take place. This allowed ample time for the tank temperature to equalize and reach a steady-state condition. The temperature of the main supply tank fluid was kept between 450 and 470 K for all the room temperature test section runs. A complete table of the run specific main fluid temperatures is presented later. The mixing fan was allowed to run constantly until just prior to initiation of each experiment, equalizing the temperature of the main flow fluid. The dump tank, test section, and boundary layer bleed plumbing were evacuated to a few torr prior to each run.
The LabVIEW simulation program, described in Chapter 2, was used to determine the desired valve areas and timings for the three electronically controlled valves using the chosen quantity for the temperature of the main flow and cooling flow. The timing box used to sequence all of the valve activations was set for each of the three ON-OFF valves and the fast-acting main valve and a signal delay was accounted for in this programming for each valve.

In preparation for each experiment the main tank and cooling supply tank were filled with air and nitrogen, respectively, to the desired initial pressure. These pressures were consistent with the pressures used to predict the necessary timings for each run. Table 3.1 shows the target initial pressures for all runs in this data set.

<table>
<thead>
<tr>
<th>Cooling Supply Tank</th>
<th>Main Supply Tank</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure</td>
<td>Temperature</td>
</tr>
<tr>
<td>200 PSI</td>
<td>300 K</td>
</tr>
</tbody>
</table>

These pressure levels were monitored before each run using the pressure transducers in the system, specifically the Hiese and PX5500, in the main and cooling supply tanks. A timing method was used to match the initial pressures with the leak rate of the system. This explains why the initial pressures were not always exactly the target pressures, but were acceptably close.

The Data Acquisition System (DAS) began recording data 0.2 seconds prior to the flow being initiated. This provided baseline readings for all instruments in the facility. Next, the bypass valve (1) opens, filling the cooling plumbing and plenum. The FAV for the main flow opens as the bypass valve closes, allowing only cooling
flow thorough the small cooling flow choke. The blowdown matching valve then opens to decrease the mass flow of the cooling flow at the same rate as the mass flow in the blowdown of the main flow. The pressure histories for the various sub-systems of the facility given in Figure 3.1 exemplify this experimental timeline.
3.1.2 Matrix of Run Conditions

Figure 3.2 shows the temperature of the metal in the test section (red symbols) and the temperature of the main flow fluid (black symbols) for each run. The driving temperature for the heat transfer measurement is directly proportional to these two initial temperatures.
Figure 3.2: Initial Temperatures for Experimental Runs
As can be seen from the results presented in Figure 3.3, runs 11-46 had a room temperature test section and runs 47-61 had heated test section walls.

The desired valve timings for the first block of runs (11-42) encountered a problem related to the resolution of the valves. The ON-OFF valves had a timing resolution on the order of approximately 5 milliseconds. In order to make any irregularities in the valve firings negligible, the bypass area (1) was decreased, which led to longer fill times for the plenum (moving from 0.15 seconds to 1.5 seconds). This decreased the sensitivity of the peak pressure to any small changes in valve activation but increased the overall length of the experiments for the adiabatic wall temperature runs (43-61). Fortunately, these inconsistent delays in the bypass valve (1) led to a diverse sample set with many different blowing ratios. A complete table for all runs appears in Appendix A.
3.2 Heat Transfer Methodology

A brief discussion of heat transfer must be included in order to understand the data reduction processed used to gain heat-flux data. Newton’s Law of Cooling dictates that the heat flux, $Q$, depends upon the temperature of the object, the temperature of the fluid, and the heat transfer coefficient of the object, $h$ [24].

$$Q = h (T_\infty - T_{Wall})$$  \hspace{1cm} (3.1)

3.3 Heat-Flux Gauge Data Reduction

The initial raw data was reduced using the coefficients gleaned from the two calibrations previously described. Next, the true temperature of the test section metal immediately before each run was determined using the plate mounted RTD’s. The accuracy to which the temperature was known for each run was a function of the differences between the RTD’s, which are individually accurate to 0.2 K. The temperature of the test section varied due to the ambient temperature in the laboratory, and on how many experiments had been conducted on that particular day; clearly the hot main flow was leaving residual heat in the metal. This can be seen from the run-to-run temperature data previously presented in Figure 3.2.

The HFG temperature readings, at steady state throughout the experimental matrix, varied from the known RTD temperature anywhere from 0 to 20 K. The schematic shown in 3.4 shows a detailed classification of the HFGs both top and bottom and their relationship to the proper initial temperature.
Figure 3.4: Classification of Initial Temperature Readings from HFGs

Given the results presented in Figure 3.4, each gauge was then offset to the correct initial temperature. Using an average of the HFG readings from 0 to 50 milliseconds and taking the difference of that mean with the correct initial temperature yielded a constant offset. This was applied to the length of each run. Subtracting the top gauge temperature from the bottom gauge temperature and multiplying that number by the k/D of the insulating material used in the construction of the gauges provided an estimate of the steady-state heat-flux magnitude. A more correct calculation of the time-averaged heat flux was determined by use of a CFD technique (described in [20]) that is designed to solve the time-accurate heat conduction equation using iterative steps in the Kapton insulating layer and accounting for the time delay between the two gauges (top and bottom) due to the insulator. Figure 3.5 contains a graphical example of the heat flux generated by this program for a typical Kapton heat-flux gauge.
This iterative method is accurate to 1% so any variation in heat flux less than 1% is inconclusive [20]. The two methods of arriving at the heat flux agreed to within 5%, but obviously the solution described in [20] is the more accurate way to proceed. This difference in value seen by use of the two approaches comes from the time delay not accounted for in the steady state calculation.

3.4 COMPARISONS OF IDENTICAL RUNS

Runs with the same cooling plenum pressure and run timings were compared to ensure repeatability. A summary of this analysis is given in Table 3.2.
Table 3.2: Conditions of "Identical" Runs - Used to Check Repeatability

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>27</td>
<td>1379.0</td>
<td>207.97</td>
<td>229.8912</td>
<td>0.9871</td>
</tr>
<tr>
<td>28</td>
<td>1380.4</td>
<td>207.33</td>
<td>231.2448</td>
<td>0.9893</td>
</tr>
<tr>
<td>30</td>
<td>1377.0</td>
<td>206.27</td>
<td>229.9659</td>
<td>0.9886</td>
</tr>
<tr>
<td>16</td>
<td>1379.4</td>
<td>257.84</td>
<td>248.3744</td>
<td>1.0209</td>
</tr>
<tr>
<td>17</td>
<td>1379.0</td>
<td>251.30</td>
<td>253.4246</td>
<td>1.0338</td>
</tr>
</tbody>
</table>

Figure 3.6: Time History of Pressure Ratios for Similar Runs

Figure 3.6 shows the pressure ratio throughout Runs 27, 28, & 30. Clearly, they are a good candidate for repeatability comparison, as the pressure ratios fall exactly on one another. Here one sees that for a period of almost 100 milliseconds a constant pressure ratio is reached. An average of 15 milliseconds is used to determine the heat transfer at a given time.
Figure 3.7: Axial Distribution of Measured Heat-Flux for Runs with Similar Initial Conditions
Figure 3.8: Mean of 3 "Identical Runs" with Error Quantification for Repeatability
Figure 3.9 shows the heat-flux for each run. The heat-flux upstream of the cooling row represents the initial heat-flux due to the difference between the hot main flow and the room temperature test section metal. Each run trends the same throughout the axial distance of the plate. Heat-flux decreases by a factor of 1000 after the two largest diameter rows of cooling (D & E). Interestingly, the heat-flux drops 3 times the initial drop after row F even though it is composed of significantly smaller diameter holes, but twice as many holes as in rows D and E. Between rows F and G the heat-flux increases again but not to the magnitude of the upstream measurements, it reaches its local maximum 2/3 of the way between the two rows. The minimum heat-flux is realized between rows G and H. Row G has almost twice the number of cooling holes as either row D or E and the diameter of the cooling holes for row G is almost 90% of the diameter of the row D and E holes. After the last row of cooling holes the heat-flux increases gradually from this minimum back to
the same value as in the center of the rows F and G. If no cooling fluid were present, one would expect the heat-flux to only decrease slightly along the length of the plate due to the small changes in main flow temperature with axial distance. Clearly, the cooling is having some effect as we have peaks and valleys in the heat-flux, therefore the designation of "Minimal" Cooling will be given to runs with a nominal pressure ratio (near 1). Pressure ratios were used to classify these runs due to the nature of the new cooling system. The bulk of the problem stems from how cooling and main mass flows are measured. The new cooling system gets rid of the compression spike but makes determining the cooling mass flow very difficult due to cooling flow bleed through the blowdown matching system. It is very important to compare all of the results presented herein on the basis of blowing ratio, but in the absence of an accurate mass flow through the aggregate cooling holes, it is difficult to determine the local values of the blowing ratios. Ongoing work is directed at determining the defining choke areas, and therefore the mass flows, from known valve positions for these runs.

Figure 3.8 shows the heat-flux mean along the entire axial length of the flat plate for Runs 27, 28, and 30, which were designed to be essentially identical conditions within any experimental variation. There error bars denote the largest difference between the means and the individual runs. The error in the system at each location is taken to equal these differences throughout the rest of the thesis. The trends in the heat-flux are most easily seen in Figure 3.8. The effect of cooling appears in the heat-flux decrease immediately after the first cooling row. This effect is more than doubled after row F. Surprisingly, we see an increase in heat-flux after row H, compared to this minimum. This implies that the cooling from rows D-G is most effective in between rows G and H. The addition of cooling in row H does not increase this effectiveness near row H.
The repeatability analysis in Figure 3.9 shows the percent difference between Runs 27 and 28. Most locations reproduce each other to within 5%. The analysis revealed individual heat-flux gauge measurements accurate to approximately 3%. The gauges with the largest error were located downstream of all the cooling rows, with the exception of the gauge at an x/D of 32. There was no difference in the repeatability of the gauges in between cooling rows when compared to the gauges before or after all of the cooling rows.
The other two runs described in Table 3.2 that were designed to be for identical conditions were Runs 16 and 17. The pressure ratio comparison given in Figure 3.10 shows that Run 17 has more cooling flow than Run 16. Therefore, it should be anticipated that the heat-flux measured for Run 17 should be lower than that measured for Run 16.

![Figure 3.10: Pressure Ratio Comparison for Two Similar Runs (16 & 17)](image-url)
Figure 3.11: Heat-Flux Repeatability Comparison for Two Similar Runs (16 & 17)

Figure 3.11 presents the heat-flux measurements that correspond with the pressure ratio time histories presented in Figure 3.10 and these results suggest that the heat-flux values are trending in the correct direction: at nearly all axial locations the measured heat flux values for Run 17 are lower than for Run 16. The exception to this is before any cooling takes place, where they line up within the estimated uncertainty, as they should since they have the same initial temperature conditions. The fact that all locations show approximately the same decrease in heat-flux for the run with more cooling, with the exceptions of x/D of 30 and 100, should be noted for a later portion of the discussion.
3.5 Comparison of Various Cooling Flows

3.5.1 Minimal Cooling v. Medium Cooling

Comparisons between the minimal cooling runs, classified by a pressure ratio near 1, and the medium cooling runs, classified by a pressure ratio between 1.1 and 1.3, shows the effects of film-cooling. A pressure ratio of unity indicated the two flows push on one another with equal force. Obviously, this is an ideal case and (as shown in the previous comparison) the minimal cooling cases do indeed release cooling flow into the test section. Any comparison to the minimally cooled runs should show that the higher pressure ratio runs, or in other word runs that emit more cold fluid from the cooling holes, protect the flat plate better and inhibit heat transfer assuming that all the cooling flow stays attached to the boundary layer protecting the plate. Figure 3.12 shows this, but also raises a few questions.
Figure 3.12: Comparison of Heat-Flux for Minimal & Medium Cooling

The x-axis is non-dimensional length, x/D, where D is the diameter of the holes in row D. Although the red triangles represent a run with higher cooling than the blue triangles, there is no apparent difference in heat transfer at several locations on the plate. The heat-flux between rows G and H is higher for the medium cooled case than for the minimal case, suggesting blow off. Also, all the locations between rows F and G have lower heat-flux for the minimally cooled runs. Clearly, the point with the least heat-flux is from the minimal cooling case run after row F. These results suggest the lower cooling protects the plate better nearest the cooling holes because the cold fluid is staying in the boundary layer close to the plate, rather than being injected off of the surface. Once the cooling mass flow is increased this benefit goes away as the cold fluid is blown further off the plate. It appears that the largest effects of cooling appear aft of the last row of cooling holes, and that the most effective run here is the medium cooling run. This is due to most of the cooling fluid reattaching.
to the boundary layer at this point downstream, and the medium cooling has more
total cold fluid in the test section at this point. The fact that blow off is suggested
in several locations lends itself to the explanation for why the best cooling happens
downstream, as all the blown off cooling flow comes back to the boundary layer at this
point. When comparing these results to that of Figure 30, it can be deduced that there
must be an optimal cooling amount for each section of the plate. In that comparison,
the slightly increased cooling flow produced better protection across the entire plate
with the exception of two places (x/D = 30, 100) where it still protected it better,
but not as effectively as the rest of the plate. These locations correspond to places
in this comparison where the increased cooling actually decreased the protection of
that section of the plate. While the medium cooling in this case protects the area
aft of the plate better, one can see that minimal cooling does a better job in these
particular locations. The gauge located at an approximate x/D of 135 was treated as
a single-sided gauge since the bottom gauge at this location failed during testing.
Figure 3.13: Further Comparison Between Minimal & Medium Cooling

Figure 3.13 is the same comparison but for two different runs. The medium cooling case in this comparison has an even higher pressure ratio than that of the previous comparison. Here the effects of blow off are visible between rows E and F whereas the previous medium cooling case (Run 15) did not have enough mass flow to cause a greater heat-flux in between these two rows. The remainder of the trends is consistent with that of Figure 3.12.

3.5.2 Minimal Cooling v. High Cooling
High cooling is defined as a run with a pressure ratio greater than 1.3. Figure 3.14 shows that the effects of high cooling show decreased heat-flux aft of all the cooling rows, but highly elevated heat-flux between both rows E and F, rows G and H, and in the initial points downstream of row F. This is consistent with the rest of the results from this section.

3.5.3 Minimal Cooling v. Ingestion

Figure 3.15 presents a comparison of the results obtained for a run for which the pressure ratio is less than 1 (labeled "ingestion"), approximately 0.75, and thus it is possible for some amount of hot air to be forced into the cooling holes. The ingestion run shows heat-flux values greater than the minimally cooled run, but still there is significant cooling provided as shown by the significant decrease in heat flux along the plate. The amount of cooling injected for the "ingestion" case is still significant.
Figure 3.15: Axial Distribution of Heat-Flux for Minimal-Cooled & Ingested Runs and results in significantly decreased heat-flux values along the plate

### 3.5.4 Medium Cooling v. High Cooling

Figure 3.16 compares the heat flux for two runs with cooling. Previous studies have shown higher cooling mass flow results in less effective cooling due to blow off interactions. Figure 3.16 exhibits the possibility of these same interactions. The heat-flux before the cooling holes is the same for both runs, but aft of this point the comparison of these runs shows difference in heat-flux due to the different cooling flows. Between rows E and F as well as between rows G and H, the heat-flux for the higher pressure ratio case is much greater than that of the medium pressure ratio. Table 3.3 shows the difference in the pressure ratios of the two runs.
Figure 3.16: Axial Distribution of Heat-Flux for High-Cooled Runs & Medium-Cooled Runs

Table 3.3: Pressure Ratios for Heat-Flux Comparison of Figure 3.16

<table>
<thead>
<tr>
<th>Row</th>
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<th>Pressure Ratio</th>
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<td>15</td>
<td>Medium</td>
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<tr>
<td>18</td>
<td>High</td>
<td>1.3438</td>
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</tbody>
</table>

Again, these results suggest blow off which causes hot air from the main flow to be entrained near the surface of the plate. Previous results showed this happened for the medium cooling case when compared to the minimal cooling case, and these results show that the effect is further exaggerated by the high cooling case.
3.5.5 Various Cooling Effectiveness Represented by Absolute & Percent Differences

Figure 3.17 presents a comparison of the heat flux obtained for four runs with different cooling properties. The measured heat flux for all four runs is in good agreement just upstream of the initial row of cooling holes. For all cases the heat-flux increases above the nominal cooling case in between cooling rows E and F. In the instance of the ingestion case, this is due to hot air flowing into the cooling passages resulting in increased temperature of the plate from the inside. The heat-flux also increases for the increased cooling cases, which seems counterintuitive. Cooling rows D and E produce the most mass flow due to the number of holes and much larger diameter than the remainder of the rows. The increased heat transfer here can be attributed to blow-off due to such large amounts of cooling injected here in the flow. A similar effect is seen when comparing the heat-flux between rows G and H and
after row H where the heat-flux difference is relatively small between all the cases. Further downstream of the cooling rows the effect of cooling becomes more dramatic. Most likely, the cooling flow is being blown off (significantly from the first 2 rows of cooling) and reattaching downstream.
Figure 3.18: Graph of Absolute Cooling Effectiveness
Figures 3.18 and 3.19 show the cooling effectiveness as a function of the difference among the three cases (medium, high, & ingestion) and the minimal cooling case. These figures include the error for each of the gauges. The effects of the different cooling cases are clearly real heat-flux changes. The same trends as previously discussed appear more quantitatively in these figures. The blow off after the cooling rows causes an increase in the heat flux anywhere from 5 to 22%. The most effective case downstream of all the cooling holes is the high cooling case, which decreased the heat flux by almost 30% when compared to the minimal cooling or ingestion cases.
Figure 3.20: Axial Distribution of Heat-Flux Percent Difference with Linear Interpolation

Figure 3.20 shows a linear interpolation between the HFG comparison points. These points are the same as in Figure 3.19 and provide an idea of the overall trend for each comparison. The ingesting case hovers around a 0% difference from the minimally cooled case, whereas the high and medium cooling cases shift significantly from higher heat-flux in between the cooling rows to much lower heat-flux aft of all the cooling holes.

3.5.6 Cooling Temperature Dependence
Figures 3.21, 3.22, and 3.23 clearly describe this system’s sensitivity to the cooling temperature. Previously the impact of cooling gas temperature on the local heat flux was not explored because low cooling temperatures (such as in Run 42) were not achieved. A change of 38 K in cooling fluid temperature changed the heat transfer between the cooling rows by approximately 30%. Although the effect diminishes downstream, one still sees a decrease in heat transfer upwards of 10% at the downstream locations. Figure 3.24 is the same data plotted to show the inverse of Figure 3.22 for clarity.
Figure 3.22: Absolute Heat-Flux Dependence on Cooling Fluid Temperature

Figure 3.23: Percent Difference of Heat-Flux Dependence on Cooling Fluid Temperature
Figure 3.24: Cooling Fluid Temperature Effect on Heat Transfer
Chapter 4 Reynolds Number Effects on Adiabatic Wall Temperature Calculations

For an ideal system with no changing boundary layer, the adiabatic wall temperature would equal the main flow temperature. As seen below from Equation 4.1, this makes the heat flux equal to zero.

\[ Q = h (T_\infty - T_{\text{wall}}) \]  \hspace{1cm} (4.1)

However, the facility in which these experiments were performed has associated with it a flow environment for which the Reynolds number changes significantly throughout each run. The effect that the main flow temperature, \( T_\infty \), has on the plate changes as the boundary layer thickens (the Reynolds number constantly decreases) during a run. The Reynolds number in the test section best represents these boundary layer changes. Equation 4.2 defines the Reynolds number; where \( \rho \) is the density of the fluid, \( U \) is the velocity, \( \mu \) is the viscosity, and \( L \) is the characteristic length of the test section.

\[ Re = \frac{\rho UL}{\mu} \]  \hspace{1cm} (4.2)

The characteristic length for the geometry used here is defined as the length in inches from the leading edge of the plate to the last HFG location.

Analyzing the driving temperature provided the unique ability of the SCF to measure changes due to Reynolds number decay during each run. The maximum
Reynolds number, occurring at the beginning of the main flow (where density times velocity is highest), is used to normalize the Reynolds number throughout the run.

\[ Re_{Correction} = \frac{Re}{Re_{max}} \quad (4.3) \]
Figure 4.1 demonstrates that the main flow remains consistent between runs, as shown by the minimal difference in the two Reynolds number corrections. The variation in the FAV firing creates a small offset in the start times. Figure 4.1 also shows that the correction factor diminishes as the boundary layer thickness increases, as expected, over the time span of each run. This is a direct result of the Reynolds number in the test section decreasing with time.

\[ T_{Driving-Absolute} = (T_\infty - T_{Gauge}) \]  

\[ T_{Drive-Corrected} = (T_\infty - T_{Gauge}) \cdot Re_{Correction} \]
Figure 4.2: Driving Temperature with and without Re Correction (H1)
Figure 4.3: Driving Temperature with and without Re Correction (H2)
Figure 4.4: Driving Temperature with and without Re Correction (H16)
The driving temperatures shown above in Figures 4.2, 4.3, 4.4, and 4.5 come from the four heat-flux gauges located upstream of the cooling rows, shown in the flat plate setup in Chapter 2. Neglecting any effect from the downstream cooling rows, the driving temperature logically should be the only variable that influences heat flux for these gauges. A linear relationship between the driving temperature and the heat flux was expected. As shown in the Figures 4.2, 4.3, 4.4, and 4.5, before any corrections are made, this is not the case. The driving temperature was calculated here using the total temperature rake in the test section and the bottom temperature reading for each HFG. Upon discovering the non-linear relationship, the first suspect was the total temperature rake. Using an idealized gas temperature calculated from the previously discussed blowdown relation and recalculating the driving temperature created no change in the shape. Accounting for the Reynolds number change throughout each run, however, did result in a linear relationship between the heat flux and the driving temperature.
Accounting for this Reynolds number effect permits a linear fit to determine the driving temperature at which the magnitude of the heat transfer is equal to zero. Using this driving temperature and the main flow fluid temperature one can calculate the temperature of the test section wall, also known as the adiabatic wall temperature, for all points along the length of the plate. The cooling flow will have an effect upon this temperature; that is, locations and runs with more effective cooling will have a lower adiabatic wall temperature.
Figure 4.6 shows the adiabatic wall temperature correction due to the Reynolds number for each gauge along the axial direction on the plate. This important correction is explored thoroughly in determining adiabatic wall temperatures in the following chapter.
Chapter 5 Adiabatic Wall Temperature Experiments

5.1 Experimental Process

Three Minco™ silicone rubber thin-film heaters provided heat for the adiabatic wall temperature experiments. The back of the test section plate housed each of these heaters. The first four runs were taken with a room temperature test section to provide a baseline for the experimental matrix. Next, the test section temperature was increased using several power supplies to control the power output of the thin-film heaters. The remainder of the adiabatic wall temperature experiments progressed in the same fashion as the previously described experiments.

5.2 Driving Temperature Variation

Variation in the test section wall temperature and the temperature of the main flow for the experiments discussed in this section created runs with significantly different driving temperatures. Appendix A contains a table of these initial wall temperatures and main tank temperatures for further reference. Appendix A also shows the driving temperature, defined here by the mean initial temperature in the main supply tank less the initial test section metal temperature.

The aluminum holder for the cooled plate successfully distributed the heat from the heaters to the cooled plate evenly. Analysis showed that the stainless steel cooled plate contained no temperature gradient due to the heaters prior to any of these runs. Therefore, a new wall temperature was determined from the RTD’s and used for processing the temperature from the heat flux gauges.
5.3 Heat-Flux Comparisons

Ideally, an adiabatic wall temperature would have a heat flux of 0 between the main flow and the flat plate. The temperature increased by the heaters on the facility was incremented slowly to protect the heat-flux gauges on the plate. This allowed for a myriad of runs with different driving temperatures. As the driving temperature decreases, or in other words as the test section wall temperature increases in comparison to the main flow temperature, one would expect the heat flux to decrease significantly. The cooling flow temperature and mass flow were the same for all runs in this section. Therefore, all changes in heat flux are due to the varying test section metal and main flow fluid temperature.
For the results shown in Figure 5.1, the driving temperature difference between run 50 and run 61 was 84 K. Clearly, the increased metal temperature associated with Run 61 combined with a 50 K decreased main flow fluid temperature created a significant decrease in heat transferred to the plate. This can now be used to pursue comparisons between runs in separate temperature regimes (e.g., room temperature) that have the same driving temperature difference.

Applying the Reynolds number correction to a heat flux versus driving temperature graph, and interpolating to where the heat flux equals 0, yields the adiabatic wall temperature at any point along the flat plate. This adiabatic wall temperature reflects the cooling effectiveness along the length of the plate. Figure ?? exemplifies this feature.

This is the same data discussed in Chapter 3 but graphed as a function of adiabatic wall temperature. Here we see where the cooling is actually increasing the adiabatic
Figure 5.2: Adiabatic Wall Temperature for a Minimally Cooled Run

wall temperature near the holes. Next, the adiabatic wall temperature is compared to see where cooling effectiveness changes when the cooling flow is changed from nominal to high. The Reynolds number correction for Runs 29 and 42 is shown in Figure 5.3.
These two runs had room temperature test section wall temperatures. Figure 5.3 shows no difference in the main flow boundary layer development behavior, since the Reynolds number does not change. The runs have a difference in main flow fluid temperature of 7.75 K. The results for the adiabatic wall temperature are offset for this difference (which directly relates to the driving temperature).
Figures ?? show clearly that the space between the cooling holes has the greatest influence on the adiabatic wall temperature. The locations upstream of the holes show no difference in adiabatic wall temperature. This is expected since increased cooling is the only difference between the runs and no cooling has been introduced at this point. The temperature of the main flow was different but this was accounted for by a constant offset. The adiabatic wall temperature shows a slight decline aft of the cooling rows for the case with the higher cooling flow. Also, in between the cooling rows the adiabatic wall temperature decreases more for the high cooling flow, this follows since the high cooling flow would pump more cold fluid into the test section.
Figure 5.5: Difference in Adiabatic Wall Temperature due to High Cooling

Figure 5.6 shows how the Reynolds number correction effects the increased wall temperature runs differently than the room temperature wall runs, which were exemplified in Figures 4.2, 4.3, 4.4, and 4.5.
The result shown in Figure 5.7 is consistent with what one would expect. No change in the adiabatic wall temperature throughout the cooled section of the plate. Since the run has minimal cooling each location should have the same adiabatic wall temperature.
Figure 5.7: Adiabatic Wall Temperature for Hot-Wall with Minimal-Cooling

Figure 5.8 compares Run 49 with Run 61, which has a higher Reynolds number than Run 29. This explains the disagreement about the adiabatic wall temperature before any cooling begins. The higher Reynolds number case will take longer for the boundary layer to develop/thicken and therefore the heat transfer to the metal at the beginning is larger. They each have the same cooling flow. Table 6 shows the conditions for each of these runs.
Figure 5.8: Adiabatic Wall Temperature for Different Temperature Conditions

Table 5.1: Run Conditions for Adiabatic Wall Comparison

<table>
<thead>
<tr>
<th>Row</th>
<th>$T_{\text{Main}}$ (K)</th>
<th>$T_{\text{Wall}}$ (K)</th>
<th>$RE_{\text{Max}}$</th>
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<tr>
<td>29</td>
<td>416</td>
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<td>2,450,660</td>
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<td>61</td>
<td>371</td>
<td>344</td>
<td>2,118,840</td>
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</tbody>
</table>

The very small changes in the adiabatic wall temperature calculations should be noted. The error in the heat-flux gauges technically masks the effects due to cooling on the adiabatic wall temperatures, as the change in temperature is very small. The trends do still appear and further exploration could reveal that another
factor increases these differences.
Chapter 6 Heat-Flux Comparisons to Fully-Cooled, Full-Stage High-Pressure Turbine Experiments at the Gas Turbine Laboratory

Completing a comparison between the full-stage high-pressure turbine data and the flat plate data of similar cooling plenum versus main flow pressure ratios will open the door to cross-facility comparisons. This type of comparison can yield an opportunity for previous and future flat plate work to inexpensively advance the crusade for an improved predictive design tool. The main obstacle for this comparison remains the variations of Stanton number calculations used in facilities of such different test section design. Although both facilities seek to answer the same questions, the simple physics of the different mass flows and test section area breakdowns, make comparing data from the two facilities complicated.

6.1 Fully-Cooled URETI Experimental Description

The Turbine Test Facility (TTF) at the GTL is a short duration shock tunnel that can be operated in blowdown mode or shock-tube mode. For the URETI program, and for all fully-cooled turbine stage experiments, the facility is operated in blowdown mode.

As seen Figure 6.1, the TTF consists of a 100-ft long tube (18.5-in inside diameter), a yellow expansion nozzle, and a very large dump tank. The inlet of the dump tank contains a high-pressure turbine stage. What one cannot see is that for every
full-stage cooled turbine that is installed in the yellow nozzle, a combustor emulator constructed as described in reference [25] provides heat to the main flow prior to reaching the turbine stage and has the capability of creating any desired turbine inlet temperature profile. Heat flux data was taken on the pressure and suction side of the turbine vanes and blades while operating at design corrected conditions. In addition, the airfoil internal cooling path temperature and pressure are routinely measured during these experiments. Comparing the two data sets for the blade and the case of a uniform vane inlet temperature profile reveals the differences in heat flux due to wakes, rotation, and curvature of the blade.

The largest challenge of this comparison is the largely unexploited Stanton number comparison. The current experimental society publishes its results as a facility-
dependent Stanton number, due to what measurements are available in each unique experimental setup. This causes difficult issue when comparing results from one experimental facility to another, including in this study. The GTL typically uses the vane inlet Reynolds number and compares the heat transfer results via the Stanton number. Since the vane Reynolds number will not be the same as the blade, which is the section of the turbine that the SCF experiments attempt to reproduce, gaining a direct comparison requires more detailed information about the flow at different locations within the turbine stage.

6.2 RESULTS OF HEAT-FLUX COMPARISONS

6.2.1 MINIMAL COOLING/PRESSURE RATIO = 1

Figure 6.2: URETI Heat-Flux for Minimally Cooled Run
Figure 6.2 shows the measured heat flux at instrumented locations downstream of the first two rows of cooling holes, then between the third and fourth row, and downstream of the fourth row. The non-dimensional length (x-axis) is consistent between this figure and Figure 6.3.

The magnitude of the heat-flux measured for the two sets of data clearly do not agree and one would not expect them to do so. However, by converting the heat flux to Stanton number and accounting for the different Reynolds number associated with each experiment, a direct comparison could be completed. That comparison is currently being done, but it was not possible to completed that work in time for inclusion within this thesis. Previously, Abhari and Epstein [26] compared the results of a fully rotating experiment with those of a cascade experiment and found a lack of correlation. Focusing on the SCF data we see the trend in the first half between the cooling rows F and G matches that of the URETI data. Then the SCF data
decreases where the URETI data continues its upward trend. The cooling rows on a flat plate have been shown in the past to block one another, which could be the case here, causing the increase in cooling effectiveness right before the holes, whereas for the case of the URETI rotating blade the secondary flow effects that are not present in the flat plate flow field would significantly impact this upstream cooling effect.
Chapter 7 Computational Investigations Using FINE/Turbo

7.1 Structured Grid Cooling Predictions

This study continued CFD work using the Numeca FINE/Turbo code initially set up by Wishart [5]. The particular version of the code used for this work employed a structured grid and modeled the cooling holes using point sources with a set mass flow and temperature. The predictions were run on a cluster of 5 processors and took approximately 2 days to reach convergence and complete each prediction. The structured grid and cooling hole injector sites are shown in Figures 7.1 and 7.2.

Figure 7.1: Flat Plate Grad [5]
In order to help the solutions converge a spreading technique was used to ensure the abrupt mixing of flows between grid spaces did not cause the solution to diverge. The right side of Figure 7.2 illustrates this spreading.

Figures 7.3 and 7.4 compare the SCF data from Run 4, taken one year ago, with various predictions computed by FINE/Turbo. The model requires the input of mass flow allocations for each cooling row or each individual cooling hole. It also needs the initial cooling flow temperature, test section wall temperature, absolute total pressure and temperature for the main flow, and the mass flow through the test section.
Figure 7.3: Run 4 CFD Predictions Altering Wall and Cooling Temperatures
Clearly, as seen in the Figures 7.3 and 7.4, the prediction seems to overplay the effect of the cooling flow. Interestingly, the peaks seen between the cooling rows seem to correspond to the previously discussed data. The SCF data plotted in the figures above (Run 4) was completed prior to the insulating advances in the cooling system. The most recent data in which the same trend as the CFD is visible had a 20 K colder cooling flow. This suggests that where the temperature in the cooling system is determined (on the plenum before the flow goes through the cooling holes) is insufficient to determine actual cooling flow temperature. Temperature gain from the metal is most likely causing a higher temperature aft of the cooling passages. This would explain why the predictions for the run conditions with higher cooling temperatures produced results with trends similar to the colder, more recent runs. Unless the loss due to the cooling passage is accounted for, the temperature input that is used in FINE/Turbo, as measured in the cooling plenum, is actually too cold.
The predictions seem to miss how the heat transfer acts before any cooling begins. This should be the most straightforward portion of the prediction. The first figure shows that at low cooling temperature (255K) small changes in test section temperature have little effect on the prediction, this is shown by the red and black-dotted line. The change in the predictions due to a 30-degree change in the input cooling temperature is represented by the change from the black dotted line and the magenta line, completed with the same wall temperature. Next, the difference due to changing the wall temperature input with high cooling temperature (285K) is presented by the difference between the magenta and green lines. We see that with a higher cooling temperature that changing the test section wall temperature has a much more significant effect than changing the wall temperature with a lower cooling temperature.

The comparison also yields information about how the predictions handle turbulence. Increasing any turbulence coefficient changes the predicted effect of cooling dramatically. The peak between the rows E and F disappears and the size of the second peak between rows G and H shrinks. Turbulence measurements were previously conducted in the SCF and indicated flow turbulence was fairly low. The results of the experimental & preliminary CFD work support this conclusion since the peaks are still seen in the data and disappear in the predictions for higher turbulence. The increase in the turbulence input also causes the prediction to miss the data downstream of all cooling, whereas before, at low turbulence, it fit reasonably well.

7.2 **NEW UNSTRUCTURED GRID**

Future work with personnel at Numeca will yield a new unstructured grid in an attempt to improve the resolution of predictions near the cooling hole injectors in the model. This data set provides the perfect opportunity for exploration of what
current technologies can accomplish, as opposed to what they noticeably cannot.
Chapter 8 Conclusions

8.1 Conclusions

The Gas Turbine Laboratory at The Ohio State University has improved and validated the previously used SCF facility that can now be effectively used to obtain fundamental data for state-of-the-art advancements in film cooling heat transfer modeling important to the gas turbine industry. This medium duration facility can discern Reynolds number effects and minute changes in heat flux. The cooling system now operates in a predictable manor for a large range of initial conditions.

Repeatability and reliability of the heat flux gauges has been proven to be very good. The measurements described in this thesis involved detailed temperature measurements and known insulator qualities necessary to calculate the heat flux at each point in time and space along a heavily instrumented flat plate that had a cooling hole pattern modeled after real engine hardware. The heat-flux gauges used in this experimental work withstood over a year of installed functionality, as well as increased test section wall temperatures without any significant gauge damage.

The test section flat plate has a much more relevant cooling pattern than many other flat plate experiments that have been reported in the open literature. Results of separate experiments within this facility showed that runs with cooling decreased the heat transfer by up to 30% when compared to runs with nominal cooling. The results also indicated a high sensitivity to cooling fluid temperature with respect to cooling effectiveness. High blowing ratios produced some blow off, most prominent
near the largest cooling holes. Ingestion increased the heat flux nearest to the cooling rows, where hot air warmed the plate from inside the cooling passages.

Examination of the driving temperature revealed Reynolds number effects. Accounting for these Reynolds number effects allowed for proper determination of the adiabatic wall temperature. This is crucial in the design world for determining cooling effectiveness and the best design strategies.

The current structured-grid CFD predictions show some correct heat flux trending. The largest contributors to changes in the predictions were the initial temperature of the cooling fluid and turbulence in any of the fluid flows. This set of experiments has provided the unique data set for a realistic geometry that can be used to help in the validation of the next state-of-the-art prediction tools for the design and optimization of film-cooled turbine stages.

8.2 Future Work

The next big step in advancing understanding of heat flux calls for development of a "Universal" Stanton number relation. In order to make optimal use of data from all facilities (flat plate, cascade, and fully rotating) comparisons need to be made possible for test sections with different mass flow treatments.

Current CFD capabilities are still in need of improvement in order to be able to predict heat-flux distributions for fully cooled high-pressure turbine experiments. Further applying current technologies to predict this repeatable heat flux data set from its initial conditions is a next logical step in helping to solve some of the difficulties. Close work with Numeca will yield a new unstructured mesh for use in FINE/Turbo in hopes of improving the resolution of the effects near the cooling holes.
The preliminary CFD results showed high sensitivity to any one turbulent flow (cooling or main flow). In order to designate this as a real effect or a misunderstanding in the CFD logic, experimental work needs to be completed with different levels of turbulence. The SCF already contains a modification for adding a turbulence grid and conducting these experiments. Work will continue on these tasks at the Gas Turbine Laboratory at The Ohio State University.
BIBLIOGRAPHY


## Appendix A Initial Temperature Conditions

Table A.1: Initial Temperatures for Cooling Run Comparisons

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Table A.2: Initial Temperatures for All Heated Wall Runs

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