



**SCALING FILM COOLING PERFORMANCE FROM AMBIENT TO NEAR ENGINE
TEMPERATURES**

THESIS

Robert W. Ashby, First Lieutenant, USAF

AFIT-ENY-MS-16-D-036

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THESIS

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Abstract

The present study examines the importance of various nondimensional parameters in scaling film cooling performance. A large volume of film cooling research has been conducted at near ambient conditions assuming that results scale to engine representative conditions. However, thermal property variations over this temperature range prevent direct scaling of results. To understand the impact of these property variations, the Film Cooling Rig was operated at near ambient temperatures. A new technique, Pressure Sensitive Paint, was also implemented to expand facility capabilities for this and future film cooling investigations. Overall effectiveness measurements were obtained at engine representative conditions and then at matching near ambient conditions for matched freestream and coolant Reynolds numbers. A representative turbine airfoil leading edge model constructed of Hastelloy was used for both temperature regimes. Model geometry consisted of a quarter cylinder followed by a flat section and then a sloped after body. Engine representative freestream conditions were obtained using premixed propane fuel and air combustion products from a Well-Stirred Reactor with air supplied as film coolant gas. Near ambient conditions used heated air for freestream flow and carbon dioxide and argon as film coolant gases to augment coolant to freestream density ratio. While relatively good agreement was obtained for overall effectiveness magnitude for some cases, observed trends within the flow field did not agree well. Engine representative cases exhibited strong downward trends in overall effectiveness with axial position while near ambient cases displayed relatively flat profiles. This trend disagreement was attributed to less than ideal agreement in flow parameters such as freestream, coolant, and internal channel Reynolds numbers. Analysis of data obtained from each temperature regime revealed trends inconsistent

with published results, suggesting possible time dependent effects in the experimental rig. Adiabatic effectiveness data was obtained using the Pressure Sensitive Paint method with carbon dioxide, argon, nitrogen, and helium as coolant gases. Analysis of preliminary results revealed expected trends with blowing ratio and density ratio for carbon dioxide, argon, and nitrogen injection.

Nomenclature

A	=	area, constant
a	=	constant
ACR	=	advective capacity ratio
B	=	constant
b	=	constant
Bi	=	Biot number
C	=	concentration, chord fraction, constant
c_p	=	specific heat at constant pressure
D	=	leading edge diameter, mass diffusion coefficient
d	=	cooling hole diameter
D_h	=	hydraulic diameter
DR	=	density ratio
e	=	constant
H	=	freestream channel height
h	=	heat transfer coefficient
I	=	momentum flux ratio
J	=	intensity
k	=	thermal conductivity
L	=	cooling hole length
Le	=	Lewis number
M	=	blowing ratio
m	=	constant
$\dot{m}$	=	mass flow rate
Ma	=	Mach number
MW	=	molecular weight
N	=	number
n	=	number density, constant
Nu	=	Nusselt number
P	=	pressure, pitch
Pr	=	Prandtl number
q''	=	heat flux
R	=	universal gas constant
r_c	=	recovery factor
Re	=	Reynolds number
S	=	trench depth, system response
Sc	=	Schmidt number
T	=	temperature
t	=	thickness
Tu	=	turbulence intensity
U	=	velocity
VR	=	velocity ratio
W	=	trench width, freestream channel width

X	=	variable
x	=	length scale, axial position
y	=	lateral position
Z	=	parameter

Subscripts

0	=	without film cooling, reference
<i>air</i>	=	with air injection
<i>app</i>	=	apparent
<i>aw</i>	=	adiabatic wall
<i>b</i>	=	background, backside
<i>c</i>	=	coolant
<i>calc</i>	=	calculated
<i>cond</i>	=	conduction
<i>conv</i>	=	convection
<i>CP</i>	=	constant properties
<i>e</i>	=	cooling hole exit
<i>engine</i>	=	engine condition
<i>exp</i>	=	experimental
<i>f</i>	=	with film cooling
<i>fg</i>	=	foreign gas
<i>film</i>	=	film
<i>g</i>	=	gas
<i>holes</i>	=	film cooling holes
<i>i</i>	=	cooling hole inlet, into surface, initial condition, internal
<i>max</i>	=	maximum
<i>mix</i>	=	with foreign gas injection
<i>model</i>	=	model condition
<i>o</i>	=	out of surface
O_2	=	oxygen
<i>p</i>	=	predicted
<i>R</i>	=	reference
<i>rad</i>	=	radiation
<i>ref</i>	=	reference
<i>rms</i>	=	root mean square
<i>S</i>	=	signal
<i>s</i>	=	surface
<i>T</i>	=	cooling hole throat
<i>w</i>	=	wall
<i>x</i>	=	axial position
∞	=	freestream

Superscripts

*	=	nondimensional variable
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$\overline{(\dots)}$	= lateral or span average
$\overline{\overline{(\dots)}}$	= area average

Greek Letters

α	= thermal diffusivity, absorptivity, coolant injection angle
β	= cooling hole fan shape flare half angle
γ	= ratio of specific heats, cooling hole arc position
Δ	= change
δ	= layback angle, uncertainty
ε	= emissivity
η	= adiabatic effectiveness
μ	= dynamic viscosity
ν	= kinematic viscosity
ρ	= density, reflectivity
ϕ	= overall effectiveness
χ	= coolant warming factor

Abbreviations

AFIT	= Air Force Institute of Technology
AFRL	= Air Force Research Laboratory
CFD	= Computational Fluid Dynamics
COAL	= Combustion Optimization and Analysis Laser
CRVP	= Counter Rotating Vortex Pair
FCR	= Film Cooling Rig
GPM	= Gallons per Minute
HAL	= Heat-Transfer and Aerothermal Laboratory
IR	= Infrared
ISSI	= Innovative Scientific Solutions, Inc.
LEFCT	= Leading Edge Film Cooling Tunnel
PIV	= Particle Imaging Velocimetry
PSP	= Pressure Sensitive Paint
SLPM	= Standard Liters per Minute
TC	= Thermocouple
TBC	= Thermal Barrier Coating
TLC	= Thermochromic Liquid Crystal
WPAFB	= Wright Patterson Air Force Base
WSR	= Well-Stirred Reactor

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1. Introduction

The research presented in this document pertains to the film cooling of gas turbine engines. Film cooling is an integral part of modern gas turbine engines enabling firing temperatures exceeding the melting temperatures of the materials used for the hottest portions of the engine. Many film cooling experiments are performed at near ambient conditions to avoid difficulties with testing at realistic engine conditions. Due to variations in thermophysical properties, problems are encountered when scaling the results from these near ambient tests to realistic engine temperatures. This investigation sought to examine the importance of various parameters in scaling film cooling performance from near ambient to engine representative temperatures.

1.1. Film Cooling Motivation

The development of the gas turbine engine for aviation in the 1940s ushered in a new age. The gas turbine engine outperformed its predecessor, the reciprocating engine, in several key areas. Gas turbines delivered far superior power to weight ratios, were more efficient at high altitudes, and were more reliable for long flights [1]. These traits are highly desirable for aeronautical propulsion applications and have made the gas turbine engine pervasive throughout modern aviation. However, early gas turbine engines were not nearly as powerful or efficient as the engines of today.

Relentless demand for improved aircraft performance and reduced fuel consumption drove the need for higher power to weight ratios and increased efficiency, respectively. Two design parameters that were heavily targeted in achieving these improvements were the overall pressure ratio and turbine inlet temperature. Increasing the turbine inlet temperature increases the engine power to weight ratio and provides some benefit to turbine component efficiency.

Overall pressure ratio strongly impacts engine cycle efficiency, with higher pressure ratio values yielding more efficient engine operation. However, fundamental thermodynamic relations dictate that increasing overall pressure ratio also raises the turbine inlet temperature. As such, improvements in engine performance were accompanied by increasing turbine inlet temperatures.

When gas turbine engine designers reached the limit of materials used to construct turbine components, a new solution was required to continue the trend of increasing turbine inlet temperature. This solution arrived in the form of cooled turbine blades. Cooling schemes generally consist of an internal component used to remove heat from the turbine blade material and an external component ejecting cool gas to reduce the heat load to the blade. However, the hottest portions of the turbine also experience some of the highest pressures in the gas turbine engine. To eject coolant for the external cooling scheme, the coolant must be at a higher pressure than the freestream flow in the turbine. This is accomplished by removing air from the highest pressure stages of the compressor, reducing the mass flow through the engine combustor [1]. Thus, the coolant air does not produce useful work in the engine system and is in fact a parasitic bleed on power production and engine efficiency. Engine designers must therefore balance the improved performance obtained with augmented turbine inlet temperature with the loss incurred from turbine cooling.

A large body of research on film cooling has been conducted. However, conducting research using true-scale models at engine conditions is very difficult and costly. As such, many studies have been performed at near ambient temperatures with scaled-up models. Because the fundamental flow physics dictate that not all of the relevant nondimensional parameters in a film cooling flow may be simultaneously matched between near ambient and engine temperatures, a

scaling problem exists. The nature of this problem requires near ambient studies to select parameters to match to the engine condition at the expense of not matching other parameters. Therefore, it would be of great benefit to characterize the relative impact of individual parameters in scaling film cooling performance. Knowledge of this sort would greatly aid researchers and engine designers in selecting which nondimensional parameters to match for near ambient testing.

1.2. Overall Objectives

This investigation aimed to accomplish three major objectives. The first was to investigate the impact of physical model size on adiabatic effectiveness by obtaining measurements at matched experimental conditions in two separate facilities using models of different physical size. The second was to determine the importance of matching various nondimensional parameters in scaling film cooling performance by making experimental measurements of overall effectiveness in two separate temperature regimes. The third and final objective was to assess the accuracy of predicted effectiveness values. Overall effectiveness would be predicted using measured adiabatic effectiveness values and adiabatic effectiveness values would be predicted using measured overall effectiveness values.

1.2.1. Physical Model Size

Two separate facilities were required to investigate the impact of physical model size on film cooling performance. The Film Cooling Rig (FCR) at the Air Force Institute of Technology (AFIT) and the Leading Edge Film Cooling Tunnel (LEFCT) operated by the Air Force Research Laboratory (AFRL) were used because the two facilities have utilized test articles of nearly identical geometry in the past. Pressure Sensitive Paint (PSP) was used to make adiabatic effectiveness measurements in both facilities. As such, functional PSP testing setups had to be

developed for each facility before work on this objective could progress. Using measured conditions from the LEFCT, test conditions in the FCR were set to match the tests conducted in the LEFCT. Because the two models are extremely similar, few difficulties were encountered in matching relevant flow parameters. However, the magnitude of the size difference between the two facilities necessitated a relatively low flow velocity for the LEFCT and a relatively high flow velocity for the FCR. For context, the FCR model was one-tenth the size of the LEFCT model, with model leading edge diameters of 0.95 cm and 9.5 cm, respectively.

1.2.2. Scaling Film Cooling Performance

Film cooling research is often conducted at near ambient conditions. These tests typically match several important nondimensional parameters governing fluid dynamics and heat transfer phenomena to accurately represent actual gas turbine engine conditions. Much less work has been conducted to validate the accuracy of this approach as compared to that relying on it. The AFIT FCR enables tests spanning from ambient to near engine temperatures. With this capability, the relative importance of various scaling parameters may be investigated experimentally. Comparing overall effectiveness distributions obtained at near ambient and high temperature conditions permits characterization of the relative importance of each nondimensional flow parameter studied in scaling film cooling performance.

Because a high degree of spatial resolution for overall effectiveness is desired, an image-based temperature measurement method, such as infrared (IR) thermography, is required. Obtaining accurate temperature measurements using an IR camera requires accounting for surface emissivity, reflected radiation, participating media, and transmission and absorption of radiation by an optical window. Fortunately, an in situ calibration may account for many of these effects as long as the test article features uniform surface emissivity. Determining a

sufficiently rigorous procedure to perform calibration of the IR camera for both the engine representative and near ambient conditions stands as a significant challenge to accurately measuring overall effectiveness distributions.

1.2.3. Effectiveness Predictions

Several commonly practiced near ambient testing techniques measure adiabatic effectiveness, effectively isolating the external cooling scheme from the internal cooling scheme. However, the internal cooling is a significant portion of the overall cooling scheme. Additionally, adiabatic effectiveness provides no direct information on turbine airfoil temperatures. Turbine airfoil surface temperatures may be presented in nondimensional form as the overall effectiveness. While accurately modeling the conjugate heat transfer effects occurring within the turbine airfoil presents a greater challenge necessary to measure engine representative overall effectiveness values, it directly provides information on turbine airfoil surface temperatures. With certain assumptions, a mathematical relationship between overall and adiabatic effectiveness may be formed. Knowledge of either the adiabatic or overall effectiveness and a handful of other flow parameters would enable prediction of the other effectiveness term. The validity of this relationship was examined at several experimental conditions. Adiabatic effectiveness measurements were made using the PSP technique and overall effectiveness measurements were made via IR thermography. Other relevant flow parameters such as heat transfer coefficients and the Biot number were measured to provide all the necessary terms in the equation relating overall and adiabatic effectiveness.

1.3. Progress Achieved

While the objectives set forth in Section 1.2 are the overall objectives for this work, achieving those objectives required the completion of several smaller objectives first. These

smaller objectives included developing an experimental method new to the AFIT FCR, characterizing a new FCR operating regime, and assessing FCR performance for expected trends and adequate data quality. The details of these objectives are discussed in the following sections.

1.3.1. Experimental Capability Development

In order to assess physical model size effects on adiabatic effectiveness, a new measurement method was required for use with the AFIT FCR. Due to the metallic nature of the FCR, thermal methods for determining adiabatic effectiveness were not feasible. As a mass transfer method, PSP enables measurement of adiabatic effectiveness regardless of the surface material. Before useful measurements with this new experimental method could be made, a functional experimental setup and experimental procedure had to be established. This was accomplished through research, training, and trial and error.

As with the PSP method development described above, operating the FCR in a new temperature regime required an understanding of the experimental method and characterization of the operating regime. Operating at near ambient temperatures required using equipment in new ways, selecting the correct equipment to minimize fluctuations and uncertainty, and understanding the rig's behavior in this new environment. Temperature control, thermal soak times required to reach steady state, and proper calibration of instrumentation such as the IR camera were of paramount importance.

1.3.1.1. Data Quality Assessment

While data quality assessment is not a research objective in and of itself, it does stand as a significant technical challenge to meeting the objectives laid out. Any time a new experimental capability is developed, that capability must be evaluated for adequate data quality. This includes verifying that performance trends are consistent with expectations and published results

and requires critical evaluation of all flow field facets. Unexpected challenges will always arise, and these challenges must be overcome before meaningful data can be recorded using a new experimental capability. Further, the experimental setup must be evaluated to ensure the phenomenon of interest can be observed within the data collected.

1.3.2. Matching Film Cooling Performance

Before the effect of various parameters on scaling film cooling performance can be investigated, correctly matched film cooling performance measurements must be made. This requires matching a multitude of parameters, some of which are notoriously difficult to measure accurately. Methods must be developed to measure or infer all the necessary parameters to ensure a good match between engine representative and near ambient conditions. Solutions will most likely require expanding FCR operating capabilities in terms of temperature and flow rate limits as well as measurement accuracy.

2. Literature Review

Film cooling was first applied to gas turbine engines in the 1960s to enhance engine performance. In this time period, combustion gas temperatures exceeded the material temperatures of the components used in engines. As turbine inlet temperatures and overall pressure ratios increased, the addition of film cooling became necessary. However, the addition of film cooling actually reduces power output and efficiency. This occurs because the gas utilized for film cooling must be compressed, requiring work output from the turbine section of the engine, and, in order to maximize the cooling potential from a unit of coolant, it cannot be used for combustion. Thus, a great deal of energy is expended to compress the coolant and sacrificially eject it from cooling holes in the turbine section, all without producing any useful work. Film cooling provides benefits by permitting higher turbine inlet temperatures and, indirectly, higher overall pressure ratios. As long as the addition of a film cooling scheme enables more additional power production than is required to implement it, film cooling should be added to an engine.

Since the inception of film cooling, a vast body of research has been conducted on a multitude of film cooling schemes under many different conditions and with several different experimental techniques. However, much of this research has been conducted at near ambient conditions. While this is highly favorable for reducing cost and increasing flexibility in experimental methods, it may not be representative of true engine conditions. Because temperature dependent property variations between near ambient conditions and engine conditions create significant difficulty when scaling film cooling performance, new research must be conducted to assess how to best account for these temperature-driven variations. Some recent work has examined the problem in much greater detail, but it has primarily been

computational in nature. This thesis intends to experimentally study the problem of scaling film cooling performance from near ambient to engine representative temperatures.

The following sections address several film cooling topics critical to the investigations conducted. Major sections include a discussion on film cooling fundamentals, geometric effects on film cooling, the effects of various flow characteristics, different experimental techniques used to measure film cooling performance, and issues associated with scaling film cooling performance from near ambient to engine representative temperatures. Film cooling fundamentals largely focuses on understanding the phenomena in a film cooling flowfield and the metrics used to quantify film cooling performance. The section on geometric effects discusses the impact of cooling hole spacing, injection angle, and shape on film cooling performance. Important flow characteristics discussed include the effect of Mach number, freestream turbulence, and internal coolant flow. Experimental techniques are divided into thermal and mass transfer methods with a brief discussion provided on the heat-mass transfer analogy. Finally, the problem of scaling film cooling performance is discussed with emphasis on experimental methods used to alleviate scaling issues and past work examining the importance of matching various flow parameters.

2.1. Film Cooling Basics

Film cooling entails the injection of a thin layer of cool fluid to reduce the temperature of a component. In gas turbine engines, film cooling is commonly used to prevent the temperature of a turbine blade from exceeding the melting point of the turbine material. This is generally accomplished by routing air from a high pressure stage in the compressor, through the engine, and ejecting it from holes in the turbine blades. Film cooling holes are typically placed in three regions on turbine blades. These regions include the leading edge region, also known as the

showerhead; pressure side; and suction side, illustrated in Figure 2-1 from Han et al. [2]. The relatively cool air forms a thin layer protecting the turbine blade from the intense heat of the gases exiting the combustor. Effective cooling schemes reduce the wall temperature, T_w , of the turbine blade and the convective heat flux, q'' , to the turbine blade.

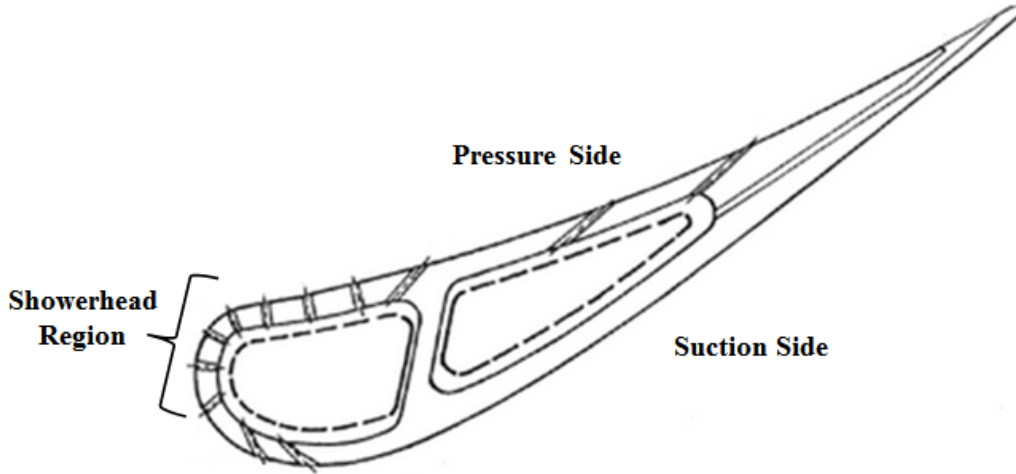


Figure 2-1: Sample turbine airfoil cooling scheme diagram (image from Han et al. [2])

Convective heat transfer in gas turbines is driven by conduction of heat from gaseous exhaust products exiting the combustor to the material of the turbine blade. This process is frequently modeled by Newton's Law of Cooling [3]

$$q'' = h(T_{ref} - T_w) \quad (2-1)$$

where q'' is the heat flux to the turbine blade, h is the convective heat transfer coefficient, and T_{ref} is an appropriate reference temperature. Bogard and Thole [4] note several difficulties arising from use of Newton's Law of Cooling. Two of the more prominent challenges include selecting a useful value for T_{ref} and measuring a value for h and T_{ref} . Figure 2-2 from Bogard and Thole highlights the locations of three temperatures relevant to the gas turbine heat transfer problem. In this case, T_g is the freestream gas temperature, T_c is the coolant temperature, and T_w is the temperature of the turbine blade surface.

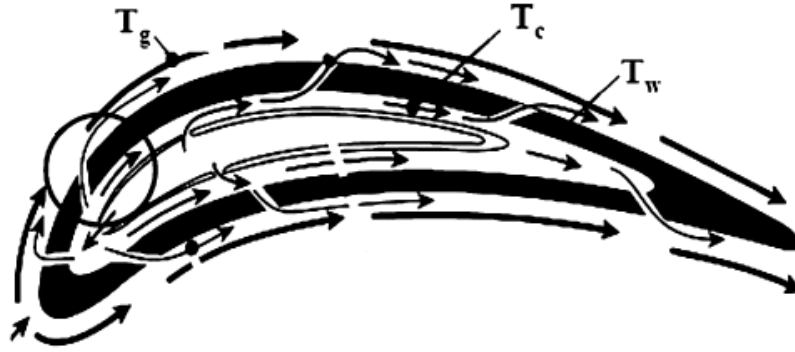


Figure 2-2: Turbine airfoil fluid flow diagram (image from Bogard and Thole [4])

The two fluid temperatures depicted in Figure 2-2 create a dilemma in selecting the correct reference temperature for Eq. (2-1). Coolant injection not only causes variation in local fluid temperature, but it also alters the momentum transport and heat transfer characteristics of the flowfield. Bogard and Thole [4] note that for h to be independent of the flowfield and T_c , which is highly desired, T_{ref} must be the driving temperature for heat transfer between the fluid and the turbine blade wall. This driving temperature should therefore be the temperature of the fluid immediately above the surface. In the case of an uncooled surface, the driving temperature is T_∞ . For a surface with film cooling, the driving temperature is the temperature of the fluid film immediately adjacent to the wall, T_{film} . T_{film} corresponds to the temperature reached by the surface if it were adiabatic. Thus, T_{film} is also commonly referred to as the adiabatic wall temperature, T_{aw} . Using T_{aw} for T_{ref} , Eq. (2-1) is rewritten as [4]

$$q_f'' = h_f(T_{aw} - T_w) \quad (2-2)$$

where q_f'' is the heat flux to the turbine blade and h_f is the heat transfer coefficient with film cooling present.

Incropera et al. [3] highlight that h varies spatially, generally decreasing in the downstream direction. Other factors affecting h include flowfield geometry, flow velocity, and

freestream turbulence. However, the introduction of coolant to the flowfield introduces a new level of complexity. Thus, Ammari et al. [5] note that the ratio of coolant to freestream fluid density, coolant injection angle, and coolant injection rate further affect the heat transfer coefficient.

Because T_{aw} reflects the temperature of the fluid adjacent to the surface, T_{aw} varies with position in the flowfield. This positional dependence is due to mixing of the coolant gas with the freestream flow and the discrete nature of film cooling holes. While a distribution of h_f values may be applicable for various flow temperatures, a distribution of T_{aw} values is specific to a given cooling scheme at a given temperature and velocity. T_{aw} is further affected by the freestream velocity, U_∞ , via the recovery factor, r_c . The recovery factor expresses the rise in static temperature experienced by a fluid through the process of being brought to rest. For a constant property, laminar boundary layer, T_{aw} may be calculated using the recovery factor [6]

$$T_{aw} = T_\infty + r_c \frac{U_\infty^2}{2c_p} \quad (2-3)$$

Alternatively, T_{aw} may be expressed in terms of the freestream Mach number, Ma_∞ [6]

$$\frac{T_{aw}}{T_\infty} = 1 + \frac{1}{2} r_c (\gamma - 1) Ma_\infty^2 \quad (2-4)$$

Kays et al. [6] note that $r_c \approx Pr^{\frac{1}{2}}$ for a laminar boundary layer and $r_c \approx Pr^{\frac{1}{3}}$ for a turbulent boundary layer.

2.1.1. Performance Metrics

To allow comparison between different cooling schemes, temperatures, and velocities, T_{aw} is generally presented in a nondimensional form known as adiabatic effectiveness [4]

$$\eta = \frac{T_{\infty} - T_{aw}}{T_{\infty} - T_{c,e}} \quad (2-5)$$

where $T_{c,e}$ is the coolant temperature at the film cooling hole exit. Values of η range from zero to unity with larger values representing better coolant coverage of the wall. η is one of the most commonly presented performance parameters in film cooling literature because it allows direct comparison of the performance of different film cooling schemes. As applied to gas turbines, η represents how well the coolant protects the wall from the hot freestream gases.

Because turbine cooling schemes are composed of an internal and external component, evaluating a cooling scheme using η essentially neglects the contributions of the internal cooling. Real turbine airfoils are made of highly conductive materials, so the airfoil surface temperature depends on the contributions of both the internal and external components of the cooling scheme. Another useful performance metric for turbine cooling that accounts for the internal cooling component is the nondimensional wall temperature, also known as overall effectiveness, defined as [4]

$$\phi = \frac{T_{\infty} - T_w}{T_{\infty} - T_{c,i}} \quad (2-6)$$

where $T_{c,i}$ is the coolant temperature at the entrance to the coolant hole. Values of ϕ also range from zero to unity with larger values desired. While engine designers can and do use η in conjunction with other parameters to estimate T_w [7], ϕ provides a more direct path to estimate T_w . As will be shown later, correctly matching certain flow parameters potentially enables direct scaling of the turbine wall temperature from experimental to engine conditions through ϕ [7,8].

2.1.2. Characteristic Parameters

As is common in fluid dynamics, film cooling is governed by nondimensional parameters. These parameters are used to ensure the relevant flow physics are correctly matched

between experiments. The most common parameters found in film cooling describe the fluid dynamics and heat transfer phenomena that occur in complex film cooling flowfields. Some of these parameters are frequently reported for many types of fluid dynamics investigations, such as the Reynolds number, Re , and Prandtl number, Pr .

$$Re = \frac{\rho U x}{\mu} = \frac{U x}{\nu} \quad (2-7)$$

$$Pr = \frac{\mu c_p}{k} = \frac{\nu}{\alpha} \quad (2-8)$$

Where ρ is the fluid density, x is an appropriate length scale, μ is the dynamic viscosity of the fluid, c_p is the specific heat of the fluid, k is the thermal conductivity of the fluid, ν is the kinematic viscosity of the fluid, and α is the thermal diffusivity of the fluid. It is important to note that these flow properties may be applied to either the freestream or coolant flow. The Reynolds number describes the relative importance of inertial and viscous effects for momentum transport within a flow, while the Prandtl number describes the relative importance of viscous and thermal diffusion effects for the transport of energy within a flow.

The Biot number, Bi , is also of great importance for film cooling flows.

$$Bi = \frac{hx}{k} \quad (2-9)$$

The Biot number relates the rate of convective heat transfer into a solid to the rate of conductive heat transfer through it. This is especially significant for gas turbine engines because the highly conductive materials used to construct turbine airfoils enable a high rate of conductive heat transfer, resulting in typical Biot numbers ranging from 0.3 to 0.6 [9]. Martiny et al. [10] note that Bi may be defined using heat transfer coefficients from the freestream, internal coolant passage, or coolant hole flows. Of these, they analytically determined that Bi based on the freestream heat transfer coefficient exerted the greatest influence on film cooling performance.

Defining Bi as such ultimately determines the relationship between convective heat flux into the turbine blade and conductive heat transfer through it. As will be shown in Section 2.6, correctly matching Bi is critical for properly scaling ϕ to engine conditions.

Other parameters are generally only found in film cooling literature, such as the density ratio, DR ; velocity ratio, VR ; blowing ratio or mass flux ratio, M ; and momentum flux ratio, I [4].

$$DR = \frac{\rho_c}{\rho_\infty} \quad (2-10)$$

$$VR = \frac{U_c}{U_\infty} \quad (2-11)$$

$$M = \frac{\rho_c U_c}{\rho_\infty U_\infty} = DR \times VR \quad (2-12)$$

$$I = \frac{\rho_c U_c^2}{\rho_\infty U_\infty^2} = M \times VR = DR \times VR^2 = \frac{M^2}{DR} \quad (2-13)$$

Where the subscripts c and ∞ refer to the coolant and freestream fluids, respectively. Bogard and Thole [4] note that each of these parameters attempts to scale a different phenomenon in the film cooling flowfield. Traditionally, M has been thought to scale the coolant thermal transport capacity while VR scales the shear layer between the freestream fluid and coolant jet, ultimately scaling turbulence production by the coolant jet. I scales the interaction of the coolant jet and freestream fluid, indicating the ability of the jet to penetrate into the freestream. Careful inspection of the parameters in Eqs. (2-10)-(2-13) reveals that if DR and any other parameter are simultaneously matched to another test condition, all of the parameters in Eqs. (2-10)-(2-13) are matched.

2.1.3. Coolant Jet Dynamics

Ideal cooling schemes generate a uniform layer of cool fluid immediately adjacent to the cooled surface. To ensure the coolant flow remains attached to the surface, fluid injection is desired to be parallel to the cooled surface. However, it is more practical for the coolant holes to be oriented at some angle to the surface. Thus, jet dynamics dictate that the coolant flow will eventually separate from the surface if the jet momentum becomes too great. Eberly and Thole [11] measured this effect by performing particle image velocimetry (PIV) measurements on a flat plate with a single row of five circular holes inclined 30 degrees to the surface for various values of M and I . They found that the jet remains well attached for $M = 0.6$ for both low and high density cases ($DR = 1.2, 1.6$). As M was increased to 1.0, the jet began to create a region of flow blockage upstream of the cooling hole, indicating jet separation from the wall. For $M = 2.0$, the jet fully separates and does not reattach to the wall. The low density case creates a noticeably larger flow blockage upstream of the hole than the high density case due to the increased jet momentum for the low density case ($I = 3.3$ for low density compared to $I = 2.4$ for high density). This suggests that the jet momentum, and thus I , is the most important parameter in predicting coolant jet separation. Eberly and Thole further found that I scales the turbulence intensity of the flowfield. As will be discussed next, high levels of turbulence may be detrimental to film cooling performance, so lower values of I may be desired. This can be achieved for a given M value by attaining a higher DR value.

The process of ejecting fluid from a jet in crossflow is highly turbulent and generates considerable vortical motion and mixing in the flow. Fric and Roshko [12] used flow visualization and hot-wire anemometry techniques to characterize the major fluid structures in a film cooling jet flowfield. Utilizing a flat plate with a circular hole normal to the plate surface,

they found that the flow is dominated by four separate structures: wake vortices, horseshoe vortices, jet shear-layer vortices, and a counter-rotating vortex pair (CRVP). Figure 2-3 shows these flow structures as they appear near a coolant jet.

These structures arise from different phenomena within the coolant jet flowfield. Fric and Roshko [12] note the jet shear-layer vortices are most prominent in the initial portion of the jet. These vortices emanate from the shear layer between the coolant jet and freestream flow due to the Kelvin-Helmholtz instability of the shear layer. Horseshoe vortices appear because the boundary layer flow encounters an adverse pressure gradient upstream of a bluff body, such as a cylinder or a coolant jet in crossflow. This adverse pressure gradient causes the flow to separate and form horseshoe vortices. Fric and Roshko argue that wake vortices appear due to an adverse pressure gradient imposed on the wall, causing separation events in the wall boundary layer. They found that these separation events lead to periodic eruptions of boundary layer fluid, forming the wake vortices which are then convected downstream. Haven and Kurosaka [13] found that the CRVP results from vorticity developed along the hole sides prior to the hole exit. Fric and Roshko [12] further state that the CRVP begins forming near the coolant hole and eventually becomes the dominant fluid structure further downstream.

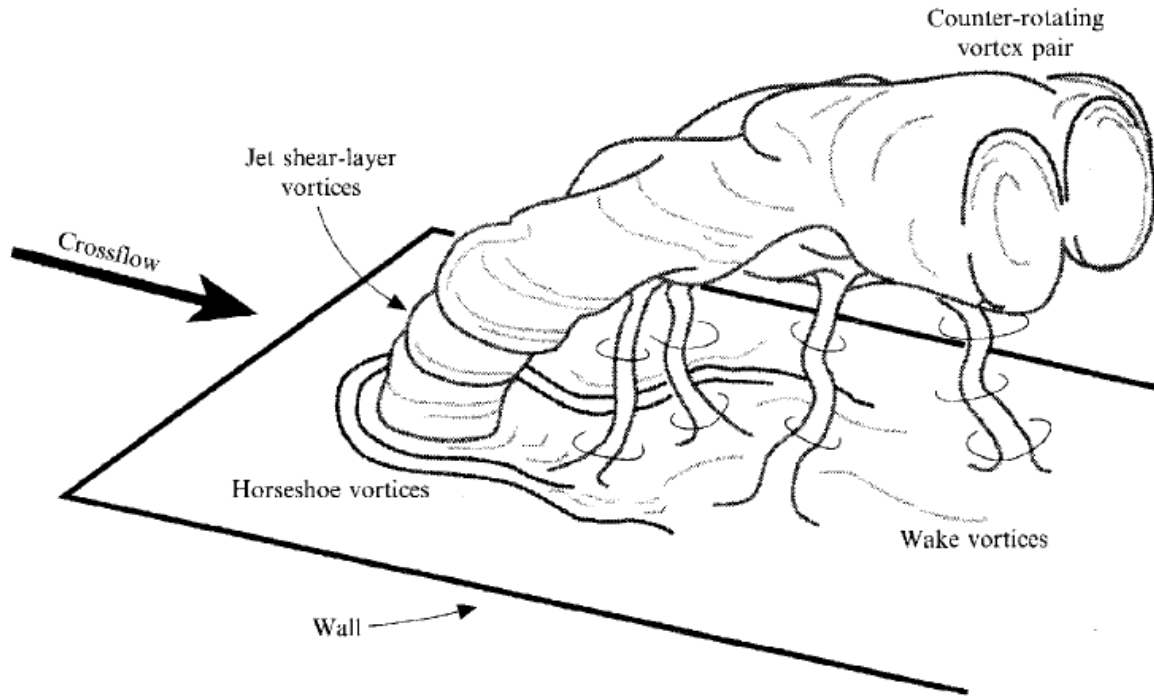


Figure 2-3: Dominant coolant jet fluid structures (image from Fric and Roshko [4])

Eberly and Thole [11] suggest that the CRVP receives the most attention of the dominant fluid structures surrounding the coolant jet because of its impact on film cooling performance. Their PIV data indicated that the presence of the CRVP affects the fluid momentum near the downstream hole edge, drawing hot freestream air down towards the cooled surface in place of the coolant. This hot flow entrainment negatively affects film cooling performance. Thus, the CRVP contributes to the mixing of freestream and coolant fluids mentioned by Lawson et al. [9]. Lawson et al. further highlight that this mixing could potentially increase the heat transfer to the cooled surface because of increased h_f values. To help evaluate the potential benefit of applying film cooling to a surface, another performance metric is required.

2.1.4. Additional Performance Metrics

For a given flow condition and T_∞ value, a reduction in heat flux corresponds to a reduction in surface temperature. A film cooling scheme's ability to reduce the heat flux to a surface is often studied and reported as the net heat flux reduction, Δq_r [9]

$$\Delta q_r = 1 - \frac{q_f''}{q_0''} = 1 - \frac{h_f(T_{aw} - T_w)}{h_0(T_\infty - T_w)} = 1 - \frac{h_f}{h_0} \left(\frac{1 - \eta}{\phi} \right) \quad (2-14)$$

Where the subscript 0 represents conditions where no film cooling is present. Δq_r is most often used to assess the effect of applying film cooling to a surface as compared to a case without film cooling. A large number of studies exclusively measure η with no means to measure ϕ , so these studies often assume a constant ϕ value. Values of Δq_r are of order unity with positive values indicating decreased heat flux to the turbine blade and negative values indicating increased heat flux to the turbine blade. Therefore, large positive values of Δq_r are desired. Maximum values for Δq_r may be attained by minimizing q_f''/q_0'' , reducing h_f , or increasing η or ϕ . As found by Ammari et al. [5], coolant injection usually increases h_f , so the best way to increase Δq_r is by developing highly effective cooling schemes.

However, Rutledge et al. [14] note several weaknesses associated with the use of Δq_r . Because the goal of film cooling is to reduce the surface temperature of turbine components, film cooling schemes should be evaluated based on their ability to reduce airfoil surface temperature, not heat flux. Further, the development of Δq_r requires that ϕ , and therefore T_w , be the same between the film-cooled and non-film-cooled cases. Therefore, any reduction in heat flux to the airfoil corresponds to decreased heat load to the internal cooling scheme at the expense of the external cooling scheme. However, any change in heat flux to the turbine blade predicted by Δq_r requires a change in T_w , violating the assumption of constant ϕ . Additionally, careful inspection

of Eq. (2-14) reveals that Δq_r may be maximized if q_f'' is zero. This condition corresponds to that of an adiabatic wall, reachable by allowing the turbine blade to reach thermal equilibrium with the freestream flow. However, for a modern gas turbine engine, a turbine blade in thermal equilibrium with the freestream flow would almost certainly fail catastrophically.

With these two large shortcomings in mind, Rutledge et al. proposed use of a different metric to evaluate the performance of a turbine cooling scheme: the $\Delta\phi$ method. $\Delta\phi$ is simply the change in ϕ attained by adding film cooling to an internal cooling scheme, and it is defined as

$$\Delta\phi = \phi - \phi_0 = \frac{T_{w0} - T_w}{T_\infty - T_c} \quad (2-15)$$

Where ϕ_0 and T_{w0} refer to the overall effectiveness and surface temperature for a case with only internal cooling. However, the assumptions used to develop the $\Delta\phi$ method do require that the Biot number be matched between the uncooled and cooled test cases. Thus, valid application of the $\Delta\phi$ method provides a direct comparison of the surface temperatures attained with and without film cooling.

While Δq_r essentially only provides qualitative information on the effect of film cooling, $\Delta\phi$ provides a clearly defined quantitative performance metric. As mentioned in the initial discussion of overall effectiveness, ϕ provides direct knowledge of the turbine surface temperature, which is of paramount concern to engine designers. Rutledge et al. [14] note that the $\Delta\phi$ method may also be used to calculate the maximum allowable freestream temperature, $T_{\infty,max}$ for a given cooling scheme or make direct comparisons between two different cooling schemes. Separate cooling schemes may be evaluated to determine the difference in $\Delta\phi$, and thus T_w , or they may be compared to find the change in maximum allowable freestream temperature.

2.2. Cooling Geometry Effects

Many different factors affect film cooling performance. Each film cooling scheme is defined by several important dimensions as illustrated in Figure 2-4. The pitch spacing, P , represents the space between holes and is usually nondimensionalized by the coolant hole diameter, d . The hole length, L , is similarly presented in nondimensional form. Most cooling schemes utilize a fairly shallow coolant injection angle, α . Typical α values are generally around 30 degrees [4]. Shallower angles would help ensure that coolant remains near the surface instead of penetrating into the freestream, but machining constraints limit how shallow the angle may be.

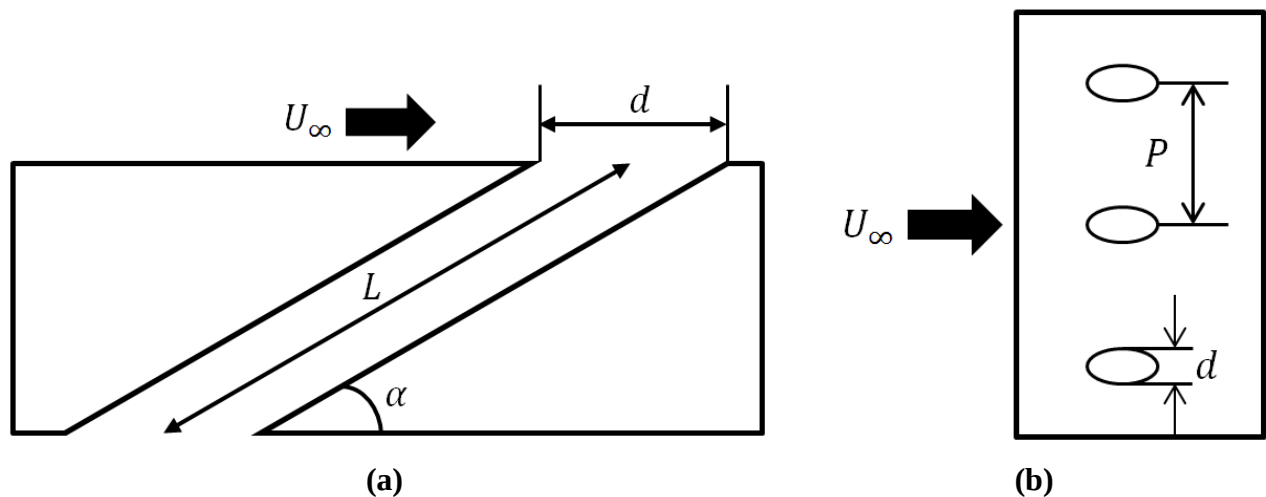


Figure 2-4: Schematic of cylindrical cooling hole geometry: (a) side view and (b) top view

Several geometric factors directly influence the performance of the film cooling scheme. These geometric considerations include items such as coolant injection angle, coolant hole spacing, and coolant hole shape. Each of these subjects has received treatment in the literature, and a review of those studies follows.

2.2.1. Coolant Hole Spacing Effects

Spacing of coolant holes within a row directly affects film cooling performance [4]. If coolant holes are spaced widely, coolant covers a small percentage of the surface area resulting in poor cooling effectiveness. Decreasing the spanwise spacing improves coolant coverage and thus enhances cooling effectiveness. Pitch spacing of three hole diameters is common, but spacing as great as eight hole diameters is not uncommon. Bogard and Thole note that lateral hole spacing is very important to determine whether or not each coolant jet functions independently. With sufficient spacing, the jets will not interact laterally, and the overall performance of a single row of holes may be predicted via the known performance of a single jet.

A study by Baldauf et al. [15] examined hole spacings of $P/d = 2, 3$, and 5 for a single row of streamwise oriented holes on a flat plate inclined at $\alpha = 30$ degrees with $DR = 1.8$. Measurements of η were made and results were presented as laterally averaged effectiveness, $\bar{\eta}$. The $P/d = 3$ and 5 results indicated that the jets acted independently, resulting in increasing peak $\bar{\eta}$ values up to a certain blowing ratio followed by decreasing peak $\bar{\eta}$ values as the coolant jets separated from the surface. Peak $\bar{\eta}$ values were obtained at $M = 0.5$ while maximum area-averaged η values, $\bar{\bar{\eta}}$, were found at $M = 0.85$. While these larger spacing cases demonstrated independent jet behavior, the $P/d = 2$ case showed that the jets did interact. The usual trend of increasing peak $\bar{\eta}$ values up to a certain blowing ratio followed by a decrease in peak $\bar{\eta}$ was observed, but the peak $\bar{\eta}$ values began to increase again at $M = 1.7$. From $M = 1.7$ to 2.5, $\bar{\eta}$ increased monotonically for $x/d > 20$ while the near-hole $\bar{\eta}$ values also increased, but not monotonically.

Additional rows of cooling holes may also be added to improve coolant coverage and increase cooling performance over the entire surface [4]. This approach may be taken to ensure

than the entire surface receives film cooling in a technique known as full coverage cooling. A film cooling scheme attaining full coverage receives no further benefit from additional downstream rows of cooling holes. Full coverage cooling is most commonly applied for combustor cooling. To maximize engine performance, it is critical that the minimum number of rows be utilized to ensure complete coolant coverage over the surface to avoid hot gases reaching the combustor wall while not excessively cooling the part.

Harrington et al. [16] examined a full coverage film cooling scheme consisting of ten rows of nine holes with each row staggered such that its holes were not directly downstream of the immediately preceding upstream holes. The holes were situated on a flat plate with normal coolant injection, and adiabatic effectiveness measurements were made at varied blowing ratios. They found that full coverage cooling could be attained with four to eight rows of coolant holes depending on the blowing ratio. The definition of full coverage used by Harrington et al. referred to a condition where adding rows of film cooling holes provides no added benefit. Increasing blowing ratio from $M = 0.25$ to 0.65 yielded an increase in laterally averaged η of $\Delta\bar{\eta} \approx 0.10$. Increasing blowing ratio from $M = 0.65$ to 1.0 provided negligible benefit. Measurements were also made with a single row to make estimates using an axial superposition method. The superposition method provided relatively good agreement with experimental measurement over the first three rows, but a discrepancy between the methods arose starting at the fourth row. At the point of full coverage, this difference in $\bar{\eta}$ was approximately 15%.

2.2.2. Coolant Injection Angle Effects

The coolant injection angle relative to the surface, illustrated as α in Figure 2-4, has a relatively significant impact on film cooling performance. Bogard and Thole [4] note typical α values are around 30 degrees and increasing α tends to reduce η , especially at low momentum

flux ratios. However, they further comment that coolant injection normal to the surface yields improved values of laterally averaged η , $\bar{\eta}$, at high momentum flux ratios where jets with shallow injection angles tend to separate from the surface. A more extensive examination of this effect was offered by Baldauf et al.

Baldauf et al. [15] also examined the effect of varied surface injection angle on their streamwise oriented holes. Injection angles of $\alpha = 30, 60$, and 90 degrees were studied for $DR = 1.8$ and $Tu = 1.5\%$ with $P/d = 3$. They found that increasing α from 30 to 90 degrees yielded an approximate 30% decrease in peak $\bar{\eta}$ at lower blowing ratios. For $M > 1.2$, peak $\bar{\eta}$ values increased by up to 60% with increasing α from 30 to 90 degrees, although the overall performance of the scheme was considered quite poor. The $\alpha = 60$ degrees case yielded results fitting well between the shallow and steep injection angle cases. In addition, the normal injection case also indicated that a closed cooling film was established at high blowing rates, providing a fairly stable $\bar{\eta}$ value with very gradual decay downstream of $x/d = 13$. This effect was observed for M as low as 1.4 and was attributed to lateral interaction of adjacent jets. With a flowfield dominated by adjacent jet interaction effects, the typical jet penetration into the freestream and hot gas entrainment near the surface were greatly subdued. Thus, for the given hole spacing, normal injection would be preferable to shallow angle injection at high blowing ratios.

The coolant jet may also be oriented at an oblique angle relative to the freestream flow. This angle is known as a compound angle and a demonstrative schematic is depicted in Figure 2-5. Bunker [17] explains that holes with compound angle are most commonly found on highly curved airfoil surfaces, generally near the airfoil leading edge. In these highly curved regions, machining requirements necessitate steep injection angles relative to the surface for holes

without compound angle. Because steep injection angles generally reduce cooling performance, compound angle holes are typically placed on these surfaces. Compound angle holes also provide increased downstream surface coverage and present a broader jet profile to the freestream [4]. Increased coolant coverage should result in enhanced cooling performance, but the accompanying increase in heat transfer coefficient resulting from increased mixing as the jet is turned downstream requires further analysis to determine the merit of applying a compound angle. Bogard and Thole note Δq_r is often used to study the benefit of including compound angle in a cooling scheme.

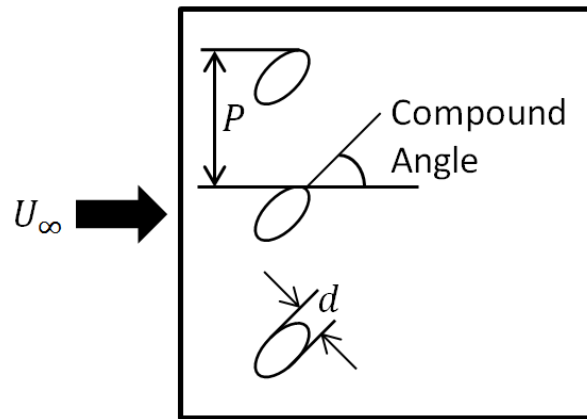


Figure 2-5: Schematic of cooling hole compound angle

Dittmar et al. [18] examined the effect of compound angle on a gently curved surface with a single row of fan-shaped holes with a 45 degree injection angle, spacing of $P/d = 4$, and flare angle of 14 degrees. One scheme utilized a compound angle of 35 degrees while the other used streamwise oriented holes. Experimental conditions included blowing ratios of $M = 0.5$, 1.0, 1.5, and 3.0 with $Tu = 9\%$. The measured contours of η for $M = 1.0$ demonstrated the compound angle ensured greater spreading of coolant, resulting in more uniform and more persistent downstream coolant coverage. This was also true at a high blowing ratio of $M = 3$. Purely streamwise injection provided more concentrated coolant coverage immediately

downstream of the coolant holes for both blowing ratios, resulting in greater peak η values compared to the compound angle scheme. Laterally averaged η values indicated that the compound angle and streamwise injection cases provided extremely similar results at all experimental conditions with only a very slight difference in $\bar{\eta}$ observed for $M = 1.5$.

Further measurements of h_f/h_0 and Δq_r were also made for the two cooling schemes described above. Dittmar et al. [18] found that laterally averaged Δq_r was greater for the compound angle injection for all blowing ratios examined. The data showed this difference most strongly over the range $10 < x/d < 20$ with the difference decreasing downstream of $x/d = 20$. However, $\overline{\Delta q_r}$ was calculated using an assumed value of 1.5 for $1/\phi$, making these conclusions highly suspect. Experimental measurements of ϕ would be preferable to ensure accurate results were obtained. Therefore, the difference in $\overline{\Delta q_r}$ shown between the two configurations can be traced to differences in $\bar{\eta}$ and $\overline{h_f/h_0}$ between the two configurations. The $\overline{h_f/h_0}$ data indicated that coolant injection increased heat transfer coefficient values for both schemes at all blowing ratios, but the streamwise injection scheme displayed a greater increase for all cases except $M = 1.5$ where the two schemes performed very similarly. Additionally, $\overline{h_f/h_0}$ increased with blowing ratio for the streamwise injection scheme while the compound angle scheme demonstrated the opposite trend.

2.2.3. Coolant Hole Shape Effects

To maximize cooling effectiveness and minimize temperature gradients, it is desired that the coolant be spread uniformly over the cooled surface. Bunker [17] notes that the ideal geometry to accomplish this is a two-dimensional (2D) slot with uniform coolant supply injecting coolant parallel to the freestream flow, as shown in Figure 2-6. The purely axial

injection provides maximum insurance that the flow remains attached to the surface, and the continuous nature of the slot ensures complete lateral coverage. However, application to gas turbines limits the use of the slot geometry due to competing design requirements such as aerodynamics, fabrication, and mechanical and thermal stress. To improve cooling performance to approach the ideal performance of the 2D slot, several different hole shapes have been developed.

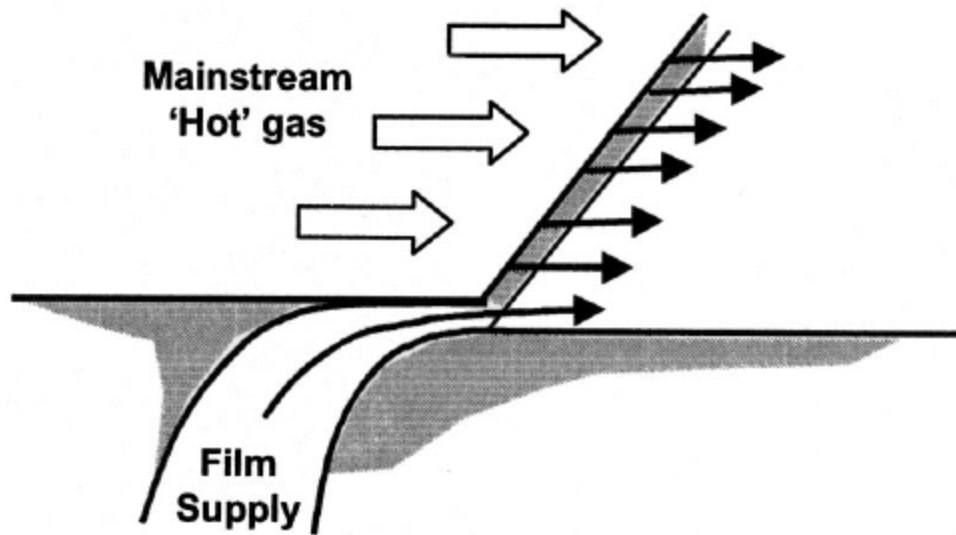


Figure 2-6: Continuous 2D slot cooling scheme (image from Bunker [17])

Because the cylindrical cooling hole is one of the simplest geometries to implement mechanically, it is the most common hole shape. These holes have a straight, circular cross-section with no contouring or cross-sectional area variation and are placed at discrete locations on the surface. This clearly does not replicate the slot's performance. Additionally, Eberly and Thole [11] note that jets produced by these holes have a strong tendency to separate from the blade surface when their momentum becomes too great relative to the freestream flow. The discrete nature of these holes may also result in poor coolant coverage. To improve film cooling coverage and performance, several new hole shapes have emerged. Among these, there are

shapes that rely on altering the shape of the hole exit, known as shaped holes, and others that place the hole into a recess within the blade surface, known as trenches.

Shaped holes rely on an expansion of the hole exit to function as a diffuser. Bunker [17] comments that this diffusion reduces jet momentum, providing reduced aerodynamic mixing losses, and enhances lateral coolant spreading, permitting lower blowing ratios and potentially increasing cooling effectiveness and turbine efficiency. Shaped holes commonly exhibit two primary alterations to the cylindrical hole. These changes include layback and fan-shaping. Because shaped holes are essentially a modification of the cylindrical hole, a section of the hole passage retains a circular cross-section. This section is known as the hole throat. These shaped hole geometries are illustrated in Figure 2-7 where D represents the cooling hole throat diameter (otherwise referred to as d in this document), L represents the hole length, L_T represents the length of the hole throat, α represents the hole angle relative to the surface, β represents the flare angle of the fan shape, and δ represents the layback angle of the hole. Bunker notes that typical values for β and δ lie in the range of 10-15 degrees to avoid flow separation within the hole.

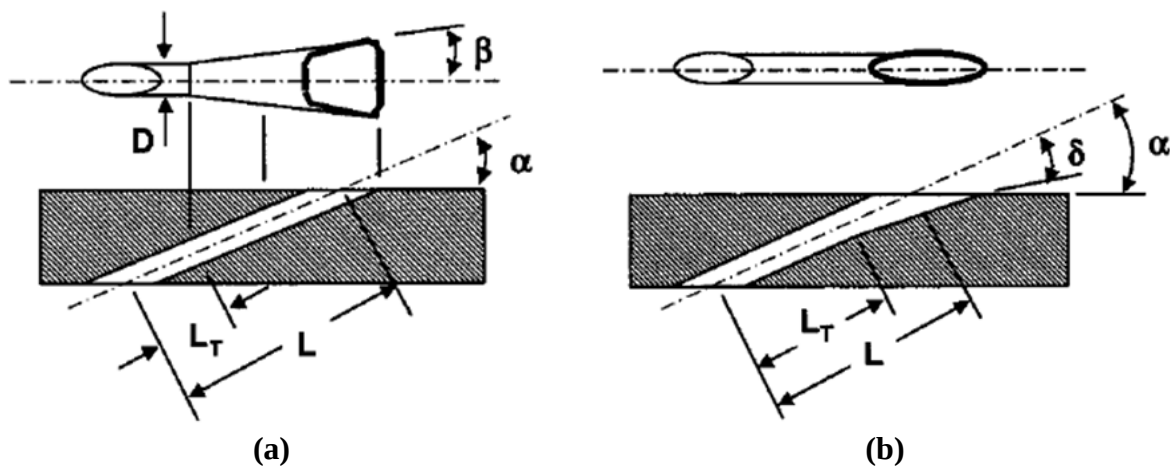


Figure 2-7: Common geometries for holes with shaped exits: (a) fan-shaped hole, (b) laidback hole (images from Bunker [17])

Saumweber et al. [19] investigated three cooling hole shapes with streamwise orientation on a flat plate, 30 degree injection angle relative to the surface, $P/d = 4$, and $DR = 1.7$ at blowing ratios ranging from $M = 0.5$ to 2.5. The shapes included cylindrical holes, a fan-shaped hole with $\beta = 14$ degrees, and a laidback fan-shaped hole with $\beta = 14$ degrees and $\delta = 15$ degrees. They found that the coolant jets from the shaped holes remained attached to the surface even for the highest blowing ratios tested. Further, the tight pitchwise spacing and enhanced lateral coolant spreading of the shaped holes allowed the individual coolant jets to merge and form a uniform cooling film similar to that afforded by a slot. This afforded a 40% improvement in laterally averaged adiabatic effectiveness at $M = 0.5$. However, performance for both types of shaped holes was very similar, indicating that very little benefit is gained from adding layback to a fan-shaped hole.

Dittmar et al. [18] also examined the effect of hole shape on heat transfer coefficient. They compared the fan-shaped hole described previously to a double row of staggered cylindrical holes with equivalent diameter and spacing. The data indicated that shaped holes increased the heat transfer coefficient more than the cylindrical holes at all blowing ratios. Because the shaped holes provided a lower momentum jet, it was speculated that this effect must be due to turbulent energy generated with the cooling hole and transported downstream into the boundary layer.

Another approach to improve the cylindrical hole is to place it in a shallow recess along the blade surface, thus creating a trench. For holes with typical streamwise injection angles, the coolant jet impinges on the recess lip causing the coolant to fill the recess before joining the freestream. This coolant flow blockage increases lateral spreading, helps prevent coolant jet separation, and results in enhanced film cooling performance. Bunker [20] presented an

excellent diagram of this flow phenomenon, and it can be seen reproduced in Figure 2-8. The recess depth is shown as S and the axial width is W .

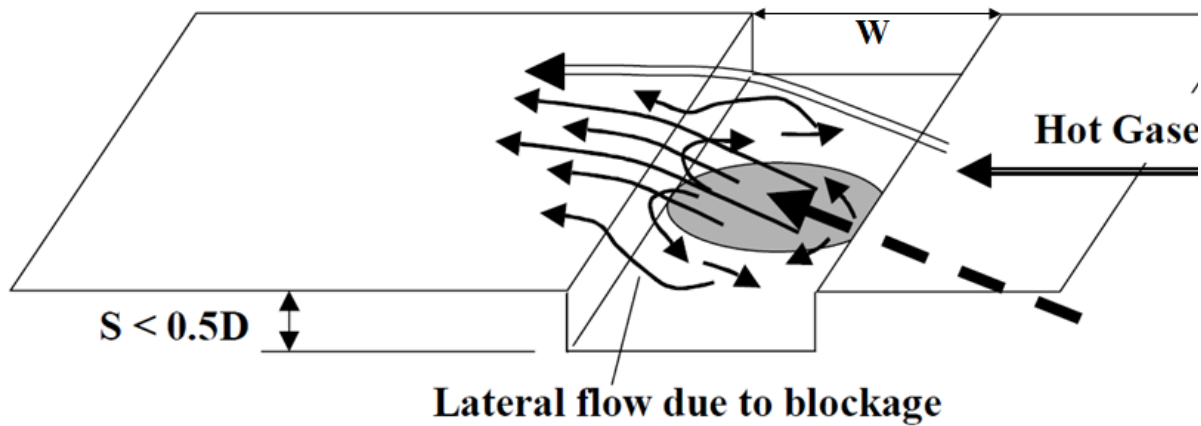


Figure 2-8: Shallow surface recess flow interaction diagram (image from Bunker [20])

Fric and Campbell [21] leveraged this effect when they developed the cratered hole, which is simply a cylindrical cooling hole placed within a circular recess on the blade surface as shown in Figure 2-9. This design is very similar to that shown in Figure 2-8 except the recess in the surface encompasses only one hole and is circular rather than rectangular. Comparing the cratered hole design to cylindrical cooling holes in a flat plate, Fric and Campbell found that the cratered holes exhibited a 50% improvement in η over the cylindrical holes at $M = 1$. This improvement in η was found to exceed 100% for $M = 5$. Further, Fric and Campbell highlighted that the cratered hole design may be applied in conjunction with other methods intended to preserve turbine components at high temperatures, such as Thermal Barrier Coatings (TBCs). Creating cratered holes on a surface with cylindrical holes could be accomplished by plugging the coolant hole exit and applying the TBC around the plugged hole.

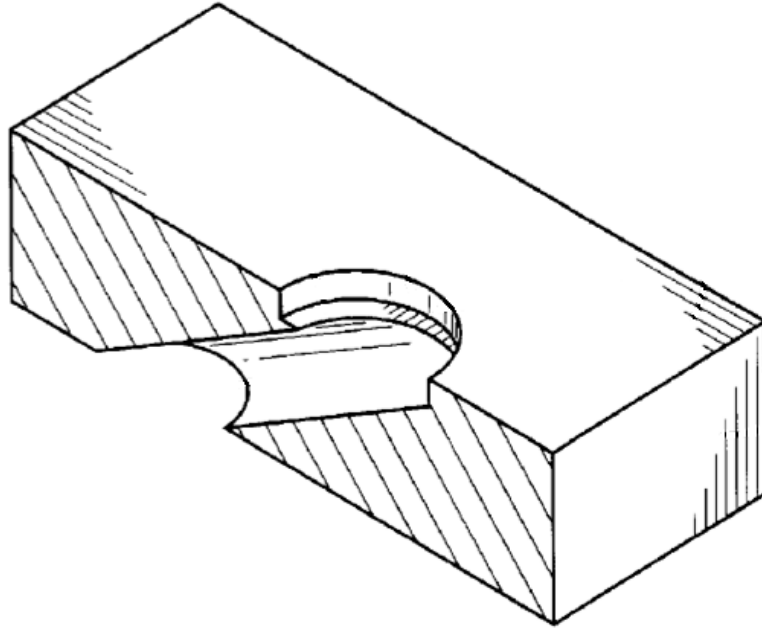


Figure 2-9: Cratered hole cross-section diagram (image from Fric [21])

Bunker [20] extended the crater concept to two dimensions with his development of the trenched cooling hole. Instead of placing each individual hole within its own recess, the recess was extended to form a long trench more similar to a slot. In his study, Bunker examined two different trenched cooling hole schemes. The first, referred to as “holes-in-slot,” utilized holes oriented in the spanwise direction while the other utilized holes oriented in the streamwise direction. Both schemes were compared to an equivalent number of cylindrical holes of identical arrangement with their exit flush to the surface. All cooling schemes utilized pitch spacing of $P/d = 4$ and an injection angle of 30 degrees relative to the surface. Measurements of η were taken for $DR = 1.8$ and $Tu = 4.5\%$ over the range $M = 0.75$ to 4.

For the “holes-in-slot” schemes, a slot depth of $S/d = 3$ and slot widths of $W/d = 1.13$ and 1.5 were used [20]. The narrower slot outperformed the wider slot for all experimental conditions. It was also found that coolant accumulated in the slot in the direction of coolant ejection, resulting in effectiveness values approaching those of the continuous 2D slot at one end and minimal effectiveness values at the other. A shallower trench with streamwise oriented

holes alleviated the asymmetric coolant accumulation of the “holes-in-slot” cooling schemes, providing superior cooling performance. The shallow trench was of depth $S/d = 0.43$ and required $W/d = 2$ to accommodate the elliptical footprint of the cylindrical hole at its exit. This scheme yielded a 50-75% improvement in η over cylindrical holes in the region $x/d < 40$. The shallow trenched cooling hole design thus provides cooling performance approaching that afforded by shaped cooling holes. Manufacture of this hole can be readily accomplished via the TBC additive manufacture approach.

Work performed by Albert and Bogard [7] also compared the performance of cylindrical holes and shallow trenched holes. Experimental measurements of η and ϕ were made using two geometrically identical models for which the arrangement and spacing of the cooling holes were identical between the two models—only the hole shape differed. The cooling scheme consisted of three rows of showerhead cooling holes with one row biased to the vane pressure side and the other two rows biased to the suction side. Measurements of η were made using a near adiabatic, low conductivity foam model while ϕ measurements were made with a Corian model intended to match the Bi distribution of a turbine vane at engine conditions. It was found that the trenched holes outperformed cylindrical holes flush to the airfoil surface. The trench helped to prevent coolant jet separation and distribute coolant laterally, resulting in higher effectiveness values near the coolant holes than with the standard circular hole design. However, both hole designs performed similarly far downstream of the cooling holes and at lower blowing ratios for which the coolant jets from the standard holes remained attached.

2.3. Flow Characteristic Effects

Flow characteristics may also have a powerful impact on film cooling performance. Some of the more important effects include freestream Mach number and freestream turbulence

intensity, Tu . The character of the coolant feed flow is also of consequence, as a plenum-fed cooling scheme behaves differently than a scheme fed by an internal crossflow. Further, the orientation and velocity of this internal crossflow can have a significant impact on the performance of a cooling scheme.

2.3.1. Mach Effects

Below the sonic regime, freestream Mach number is generally insignificant other than to describe flow velocity and potential compressibility effects. However, Gritsch et al. [22] demonstrated that the interaction between the jet and freestream becomes much more complex when the freestream flow becomes supersonic. They tested a single hole of three separate geometries on a flat plate at blowing ratios ranging from $M = 0.25$ to 2.0 and freestream Mach numbers of $Ma_\infty = 0.3, 0.6$, and 1.2 at a density ratio of $DR = 1.85$. The hole geometries consisted of a cylindrical hole, a fan-shaped hole with a flare angle of $\beta = 14$ degrees, and a laidback fan-shaped hole with a layback angle of $\delta = 15$ degrees and the same flare angle as the fan-shaped hole. Each geometry used the same injection angle of $\alpha = 30$ degrees. Laterally averaged measurements of η revealed that adiabatic effectiveness generally increased from subsonic to supersonic freestream Mach numbers. Additionally, the cylindrical holes experienced a much different effect from supersonic freestream flow than did the shaped hole geometries.

For a subsonic freestream ($Ma_\infty = 0.3$ and 0.6), the performance of the cylindrical hole was essentially insensitive to freestream Mach number [22]. However, because the jet acted as an obstacle to the supersonic freestream flow, the interaction of the jet with the supersonic freestream flow caused a bow shock upstream of the jet. The bow shock created an adverse pressure gradient which resulted in a boundary layer separation from the surface. After passing

the coolant hole, the combined jet and freestream flows underwent a recompression shock, forcing the flow back down to the surface. The pressure field resulting from this sequence of shocks thus prevented typical coolant jet penetration into the freestream flow. Gritsch et al. state that this could be of particular benefit at high blowing ratios where jets from cylindrical holes tend to separate.

The shaped holes studied by Gritsch et al. [22] revealed a trend similar to but less pronounced than that exhibited by the cylindrical holes when comparing subsonic and supersonic freestream flows. This was attributed to the reduced coolant momentum caused by the diffusion action of the shaped hole exits, resulting in decreased penetration into the freestream flow. While this result was similar between both cylindrical and shaped holes, the shaped holes also exhibited some dependence on Mach number for subsonic freestream flows. Gritsch et al. attributed the difference between the $Ma_\infty = 0.3$ and 0.6 cases to two separate effects: the increased pressure ratio required to attain an equivalent blowing ratio at higher freestream Mach number and the tendency of coolant to separate from one side of the diffuser section in the coolant hole. Using laser light sheet flow visualization, they found that the coolant distribution at the hole exit plane depended on the pressure ratio across the hole, resulting in increased hole centerline flow for higher pressure ratios. The separation from one side of the hole diffuser section created a skewed coolant distribution resulting in reduced $\bar{\eta}$ values.

2.3.2. Freestream Turbulence Effects

Freestream turbulence has the effect of increasing mixing between the coolant jet and hot freestream flow. This increased mixing generally reduces film cooling performance, as shown by Baldauf et al. [15] and Saumweber et al. [19]. For their streamwise oriented cylindrical holes inclined at $\alpha = 30$ degrees, Baldauf et al. [15] varied Tu from 1.5 to 4% at blowing ratios of $M =$

0.4, 0.83, and 1.7. A consistent decrease in laterally averaged η was observed downstream of the holes while the peak $\bar{\eta}$ values near the hole location were fairly consistent.

In addition to their study on coolant hole shapes, Saumweber et al. [19] also performed a study investigating the effect of freestream turbulence intensity on adiabatic cooling effectiveness where Tu was varied from 3.6 to 11%. This study used the cooling configuration and hole shapes described previously. For cases where the coolant jets remained attached to the surface, the enhanced mixing from increased Tu reduced η regardless of the hole shape or blowing ratio. This was the case for low to moderate blowing ratio with the cylindrical holes for which reductions in local η and laterally averaged η were seen to be as great as 40 and 25% respectively. However, while coolant jets from cylindrical holes separate at high blowing ratio, usually resulting in reduced effectiveness, Saumweber et al. found that increased Tu improves lateral coolant spreading because turbulent mixing brings more coolant back to the surface, thus improving cooling effectiveness. Because the jets from the shaped holes remained attached at all blowing ratios, increased Tu was found to decrease η in all cases. This reduction of η was as great as 30% and was most pronounced at low M values. These results are consistent with the findings of Baldauf et al. [15] despite the smaller range of Tu examined by Baldauf et al.

2.3.3. Internal Flow Effects

Many studies in film cooling literature have utilized a plenum for coolant feed flow. While this is convenient for experimentation, it is not representative of the crossflow conditions existing in real gas turbine engine hardware. Thole et al. [23] studied the effects of coolant feed flow Mach number on a single, large-scale, cylindrical cooling hole inclined at $\alpha = 30$ degrees on a flat plate at a freestream Mach number of $Ma_\infty = 0.25$. The coolant feed flow was oriented parallel to the freestream flow and the coolant Mach number, Ma_c , was assessed at the cooling

hole entrance and varied between $Ma_c = 0$ and 0.5. Measurements made using Laser Doppler Velocimetry (LDV) provided flowfield velocities as well as turbulence intensities.

The flowfield velocity data indicated that increasing coolant Mach number caused the coolant jet to shift towards the leeward edge of the coolant hole [23]. This result was consistent for locations within the hole, at the hole exit, and slightly above the hole. Thole et al. explained that this occurs because a separation region forms at the entrance to the coolant hole, and the coolant Mach number determines where this occurs. For the plenum-fed scheme, the separation region occurs on the leeward edge of the coolant hole entrance, while the higher coolant Mach number cases exhibit a separation region on the windward edge. The presence of this region disturbs the coolant flow, biasing it towards the opposite edge of the hole. Turbulence intensity data at all measured hole locations revealed that matching Mach numbers between the coolant feed and freestream flows produced the lowest turbulence levels, with the lowest turbulence levels occurring for the minimum difference between freestream and coolant Mach numbers of $Ma_\infty = 0.25$ and $Ma_c = 0.3$.

Gritsch et al. [22] also studied the effects of coolant Mach number but focused on the impact on laterally averaged adiabatic effectiveness using the three hole geometries described in Section 2.3.1. Their study also examined the effect of coolant crossflow orientation, utilizing coolant flows both parallel and perpendicular to the freestream at Mach numbers of $Ma_c = 0$ and 0.6. Cooling performance was found to be dependent on coolant Mach number as well as flow orientation. This effect was strongest for the cylindrical hole and weakest for the laidback fan-shaped hole with the fan-shaped hole exhibiting a level of effect between the other two geometries.

For the cylindrical hole, coolant Mach number and flow orientation had the greatest impact on cooling performance at higher blowing ratios ($M = 1.0$ and 1.5) [22]. Gritsch et al. stated that a leeward shift of the coolant jet with increased coolant Mach number caused the coolant to eject more towards the leeward edge of the hole, resulting in reduced mixing with the freestream flow. The windward shift characteristic of plenum-fed holes would thus result in increased jet mixing with the freestream, causing the observed reduction in $\bar{\eta}$ values. Gritsch et al. hypothesized that a perpendicular crossflow helps keep the coolant jet near the wall, thus resulting in improved lateral coolant spreading and peak $\bar{\eta}$ values. The fan-shaped hole exhibited a similar dependence on coolant Mach number but cooling performance was adversely affected by the perpendicular crossflow. The laidback fan-shaped hole only demonstrated dependence on coolant Mach number at the highest blowing ratio ($M = 1.5$), exhibiting the same trends as the fan-shaped hole.

2.4. Conjugate Heat Transfer Effects

Film cooling schemes consist of both an internal and external component. The interaction of these two parts of the overall cooling scheme is known as conjugate heat transfer. The coupled nature of the two cooling scheme components directly lends itself to effectiveness measurements using ϕ . Measurements of ϕ better represent real turbine airfoils. Because highly conductive materials are used for actual gas turbine engine parts, internal cooling may play a very significant role in reducing turbine airfoil temperatures. Some studies have found that the contribution of the internal cooling scheme may even outweigh the external cooling [8,24]. Further, conditions suited to produce optimal η values may not be the best conditions to maximize ϕ [8].

Sweeney and Rhodes [25] tested several specimens of their effusion cooled combustor liner constructed of titanium. The double-walled cooling scheme consisted of an internal wall with spars attached to the external wall to provide conductive heat transfer paths and normal holes to provide impingement cooling on the external wall. The external wall featured film cooling holes, and each specimen varied either film injection angle or hole spacing. For all the specimens tested, it was found that increasing the blowing ratio increased overall effectiveness. This is consistent with results found measuring η , especially for the relatively low blowing ratio values tested ($M < 1$). They further found that reducing film injection angle from $\alpha = 90$ degrees to $\alpha = 30$ degrees increased overall effectiveness. Reducing hole spacing from $P/d = 14.8$ to $P/d = 10.5$ also improved overall effectiveness. As addressed in Sections 2.2.1 and 2.2.2, both of these modifications increase film cooling coverage and should increase the effectiveness of the cooling scheme.

Experiments performed by Lawson et al. [9] also showed a trend of increasing ϕ with increasing M in spite of a drastically different cooling scheme and range of M values tested. Their Haynes 230 conductive test specimen incorporated five plenum-fed cooling holes with a layback angle of $\delta = 15$ degrees and spacing of $P/d = 3$. This layback likely helped to keep the coolant jets attached to the surface over the range of blowing ratios tested spanning from $M = 1$ to 3. Despite a less effective internal cooling scheme than that used by Sweeney and Rhodes [25], Lawson et al. [9] still observed increasing $\bar{\phi}$ values over the entire range of blowing ratios tested. The same was true when observing ϕ contours. While the reported data cannot be used to definitively determine if the coolant jets separated from the surface, the laidback holes likely kept the flow attached, as indicated by Saumweber et al. [19]. Because the experiments of Sweeney and Rhodes [25] and Lawson et al. [9] failed to guarantee coolant jet separation and no

effectiveness measurements without film cooling were made, no conclusions regarding the relative contribution of the internal cooling scheme can be made. Williams et al. [8] performed ϕ measurements with coolant jet separation and without film cooling altogether, and their results are discussed next.

Williams et al. [8] assessed the film cooling performance of a single row of circular cooling holes with an internal impingement cooling scheme on the suction side of a turbine vane. A low conductivity foam model was used to measure η and a model constructed of Corian to provide Bi matched to engine conditions was used to measure ϕ . Flow rate sweeps were conducted to assess the effects of internal and external cooling on the airfoil. They found good agreement with the work of Baldauf et al. [15] after correcting for different pitch. Span-averaged values of η ($\bar{\eta}$) were highest for the lowest flow rate and predictably declined with increasing momentum flux ratio as the coolant jets lifted off of the airfoil surface. However, while $\bar{\eta}$ decreased with increasing I , $\bar{\phi}$ values remained constant and then increased up to $I = 1.69$. Further flow rate increase beyond $I = 1.69$ yielded no appreciable increase in $\bar{\phi}$. Contour plots of the data revealed a region of diminished η immediately downstream of the coolant holes with increasing I , indicating coolant jet lift-off. While ϕ contours revealed a similar trend, increasing I also improved cooling performance between the cooling holes. This resulted from improved internal cooling due to increased flow through the center of the airfoil feeding the coolant holes.

Williams et al. [8] also noted that the greatest spanwise variation in ϕ occurred at the lowest flow rate. However, the lowest flow rate also yielded the largest values of η in the study. If a cooling scheme were evaluated based solely on the external cooling, the lowest flow rate would be selected and large spanwise variations in ϕ would be present. Spanwise variation in ϕ

corresponds to thermal gradients which typically result in thermal stress, reducing part life in gas turbine engines. Additionally, while the largest spanwise variation in ϕ was only $\Delta\phi = 0.05$, a change in ϕ of as little as $\Delta\phi = 0.02$ could result in a change in airfoil temperature of as much as 30 °C. Han et al. [2] note that a temperature increase of this magnitude could halve part life for temperature driven failure mechanisms. Also, the values of ϕ were found to be markedly larger than for η at high flow rates [8].

The effects of internal cooling were isolated by measuring ϕ with two cooling holes blocked to provide an estimate of ϕ_0 [8]. Little spanwise variation was observed, so $\overline{\phi_0}$ was used in further calculations. While $\overline{\phi_0}$ varied little from $I = 0.38$ to $I = 1.11$, it increased steadily from $\overline{\phi_0} = 0.22$ to 0.28 for further flow rate increases. Comparing this peak value of $\overline{\phi_0}$ to the peak value of $\overline{\phi}$ with film cooling ($\overline{\phi} = 0.32$), it becomes readily apparent that the impingement jets of the internal scheme provide a bulk of the cooling capacity. This is consistent with the work of Esgar [24], who analytically found that external cooling had little impact on airfoil temperatures compared with the contributions of internal cooling. However, this may not be true depending on the internal cooling arrangement.

Albert and Bogard [7] made ϕ measurements on a model identical to that used by Williams et al. [8] with a different internal cooling arrangement. Because the model of Albert and Bogard [7] did not incorporate the internal impingement cooling used by Williams et al. [8], the trends observed in their measured ϕ values differed slightly. Albert and Bogard [7] observed decreasing $\overline{\phi}$ values with increasing blowing ratio on the airfoil pressure side while the showerhead region showed increasing $\overline{\phi}$ with increasing blowing ratio. Further, their ϕ distributions varied considerably based on location relative to the cooling holes, especially when compared to the ϕ distributions measured by Williams et al. [8]. This is consistent with the

difference in internal cooling scheme, as the more effective internal impingement scheme of Williams et al. [8] would ensure better surface temperature uniformity than the simple channel flow scheme of Albert and Bogard [7].

2.5. Experimental Techniques

As technologies have matured, methods for measuring film cooling performance have also evolved. Measurement techniques can be grouped into two major categories: thermal measurement methods and mass transfer measurement methods. The selection of a measurement technique depends largely on the parameter to be measured, the flow conditions the measurement must be taken in, and the desired spatial resolution. There is also some debate in the literature regarding the accuracy of mass transfer measurements and their ability to accurately model all of the relevant phenomena in a film cooling flow field.

2.5.1. Thermal Measurement Methods

Thermal measurement methods have been used to measure h , η , and ϕ , and have been used exclusively to experimentally measure parameters such as Δq_r and $\Delta\phi$. A large number of thermal techniques have been used, including thermocouples [26], thermochromic liquid crystals (TLCs) [27], and infrared thermography [28]. In addition, both steady [7] and transient [28] infrared thermography techniques have been used. Past experiments utilizing thermal measurement methods generally utilized two model types: a low-conductivity model or a conductive model. The performance metric to be measured ultimately determines the type of model used.

While no material can truly be adiabatic, many materials have sufficiently low thermal conductivity as to approximate an adiabatic wall. This readily allows measurement of T_{aw} , and

thus η , with only a few simple assumptions. The most important of these assumptions is that of one-dimensional conduction through the model [4]. This is commonly accepted as true only outside the near-hole region where multi-dimensional conduction effects may occur under the influence of coolant passing through the hole. A simple conduction correction is performed based on the assumption of 1-D conduction through the model to account for the limited heat transfer through the model. Because many low conductivity models are made of plastic [22] or foam [26] with relatively low melting temperatures, these models are mostly tested at near ambient conditions.

Conductive models are sometimes used to represent the heat and mass transfer phenomena present at engine conditions. This conductivity means the model readily transfers heat, so T_{aw} is difficult to determine. Thus, ϕ measurements representative of engine conditions are best made with conductive models. Several studies have experimentally measured ϕ using conductive models, including those by Albert and Bogard [7], Lawson et al. [9], and Sweeney and Rhodes [25].

However, DeLallo [29] described a method to obtain both T_{aw} and h_f for high temperature film cooling tests with a metallic test article. By matching external flowfield conditions for two tests, the adiabatic wall temperature could theoretically be matched between the two tests. Varying the surface temperature by changing farfield boundary conditions caused a reduction in heat flux without affecting the flowfield and thus T_{aw} , enabling heat flux to be expressed as a function of surface temperature. Assuming constant properties, applying a linear curve fit, and extrapolating to where $q'' = 0$, as in Figure 2-10, allows estimation of T_{aw} while measuring the slope of the fit curve provides an estimate for h_f . This method is consistent with redefining Eq. (2-2) as shown in Eq. (2-16)

$$q'' = h_{app}(T_{aw,app} - T_w) \quad (2-16)$$

Where $h_{app} = -h_f$ and the subscript *app* refers to values apparent from the extrapolation method.

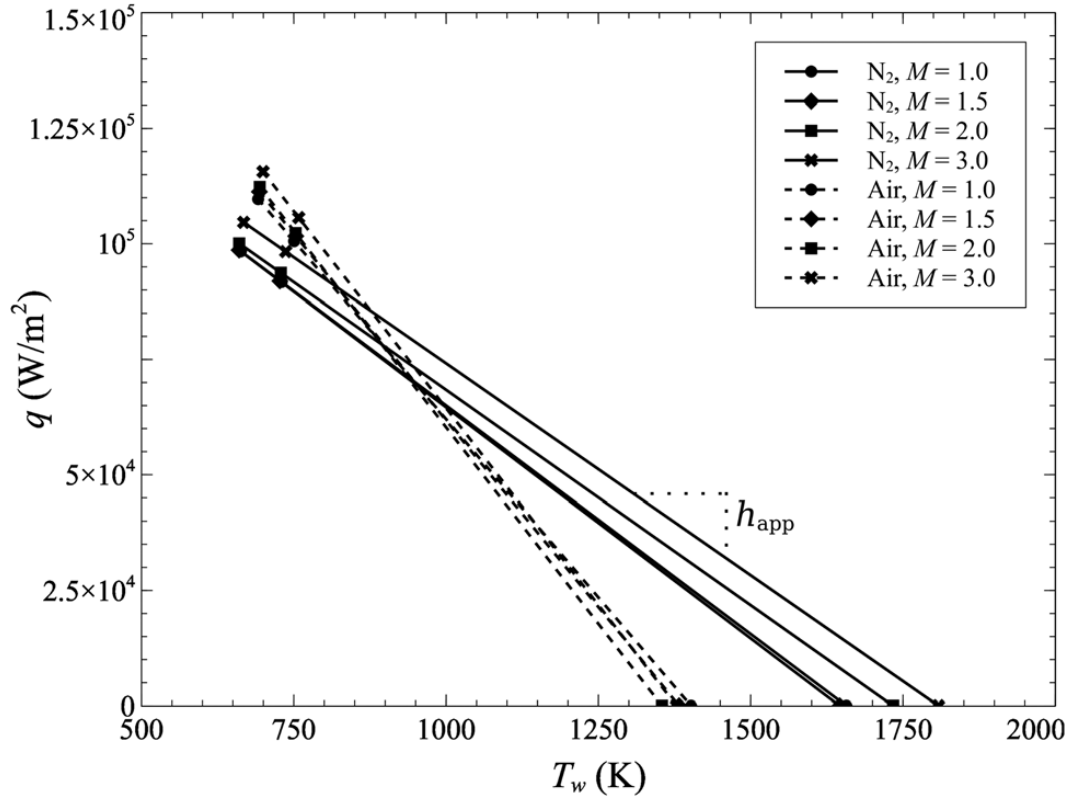


Figure 2-10: Adiabatic wall temperature extrapolation method (from DeLallo [29])

However, DeLallo noted that non-reacting film coolant provided a greater adiabatic wall temperature than reacting film coolant. Such a result was entirely contrary to expectation, as the secondary flame of reacting film coolant should theoretically increase heat transfer to the test article. This unexpected and non-intuitive result required further investigation.

Greiner et al. [30] noted that the large freestream-to-wall temperature difference in DeLallo's work created significant potential for variable property effects within the boundary layer. A computational study was performed to validate methods used to account for these

variable property effects. Greiner et al. found that q'' is not a linear function of T_w , but actually creates a curve as shown in Figure 2-11.

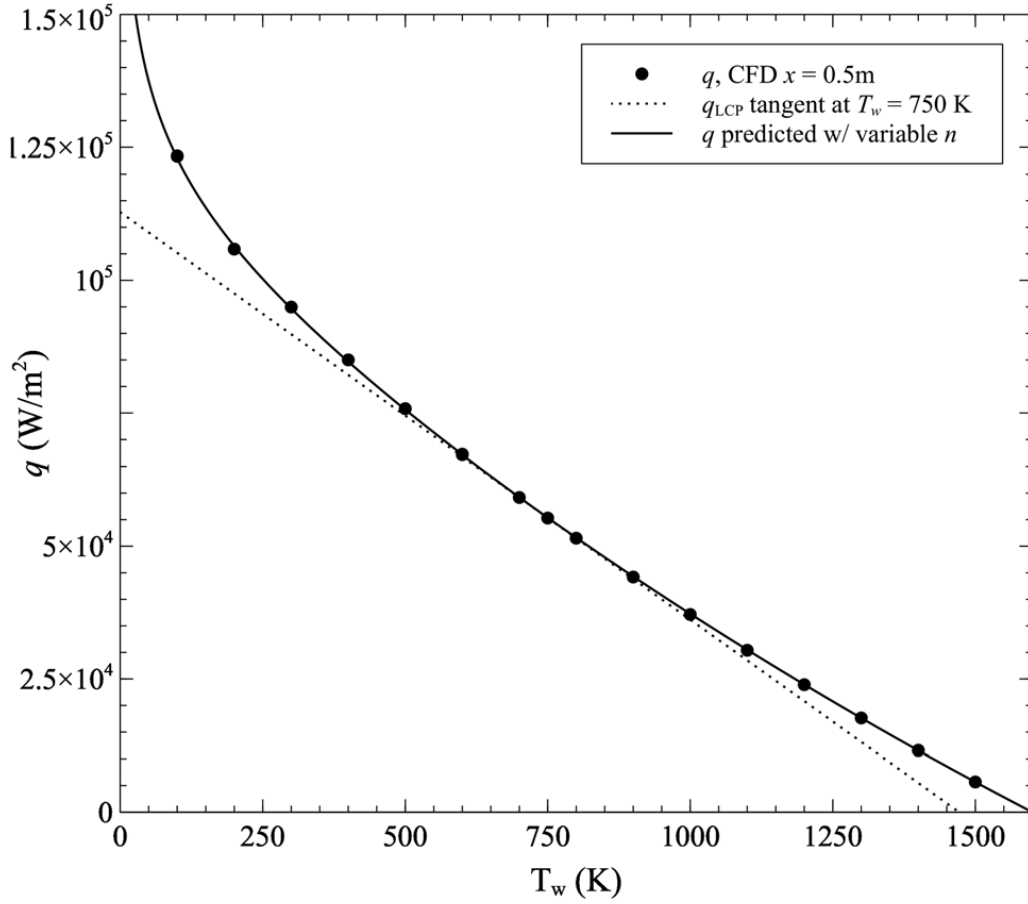


Figure 2-11: Effect of variable properties on heat flux over a large range of wall temperatures (from Greiner et al. [30])

An expression to correct $T_{aw,app}$ was generated by comparing computational T_{aw} results to $T_{aw,app}$ found using locally constant properties, leading to an enhancement of DeLallo's extrapolation method. The necessary relations are shown in Eqs. (2-17) and (2-18).

$$T_{aw} = \frac{T_w [nT_w - (1+n)T_{aw,app}]}{(n-1)T_w - nT_{aw,app}} \quad (2-17)$$

$$h_{CP} = \frac{q''|_{T_w}}{T_{aw} - T_w} \left(\frac{T_w}{T_{aw}} \right)^{-n} \quad (2-18)$$

where the subscript CP denotes the constant property solution where all properties are evaluated at the freestream temperature and n is a coefficient consistent with the property ratio method of Kays et al. [6], shown later in Eq. (2-43). The value of n may be taken as a constant of -0.2 or a temperature dependent value may be assessed using Eq. (2-19) as suggested by Greiner et al.

$$n = -(0.129 + 0.171T_w/T_\infty) \quad (2-19)$$

Comparing the results of this method to computational results, Greiner et al. found respective errors of 5.1% and 4.8% for estimated values of T_{aw} and h_{CP} .

Thermal measurement methods may further be described based on the technique used to acquire thermal data. Earlier studies like that performed by Sinha et al. [26] utilized thermocouples to measure temperatures, but these studies were limited by the discrete measurement nature of thermocouples. Sinha et al. utilized thin ribbon thermocouples with several junctions at various locations on the surface to help alleviate this shortcoming. In spite of using 60 point measurements, their data still lacked detailed information on the surface temperature between the thermocouple junctions.

To improve temperature measurement spatial resolution, Ekkad et al. [27] used TLCs painted onto a test surface. The color appearance of these crystals is sensitive to temperature, so measurements of the TLC color using a CCD-RGB camera can be calibrated to yield a temperature. Because the TLCs cover the entire surface, detailed temperature measurements can be made unlike with thermocouples. However, Ekkad et al. [28] and Wright et al. [31] note several difficulties with this method, chief of which is the limited operating temperature for the TLCs (cannot exceed 100 °C). Further challenges arise from the need to paint the model surface black before applying the TLCs and the sensitivity of the TLCs to lighting and camera angle [31]. While an additional coat of paint and in-situ calibration of the TLCs to account for lighting

and camera angle are moderate inconveniences, the limited temperature range of the TLCs is a serious hindrance. This primary shortcoming spurred the need for yet another measurement technique.

Film cooling researchers next developed surface temperature measurement techniques relying on the radiative heat transfer that occurs when an object is not in thermal equilibrium with its surroundings. Because much of this radiation lies within the infrared range of the electromagnetic spectrum, infrared (IR) thermography cameras were a logical choice for a new temperature measurement method. Ekkad et al. [28] note that IR measurement methods provide significant advantages over TLCs. The first is the capacity to make direct measurements of the surface temperature as opposed to the indirect method employed with TLCs. Further, IR cameras allow a much greater operating temperature range, making them much more attractive for film cooling experiments. Additionally, Ekkad et al. [28] note that surface preparation for IR thermography may only require a coat of inexpensive black paint as compared to the expensive liquid crystals needed for TLC measurements.

While both the TLC and IR thermography methods are extremely attractive for their spatial resolution capabilities, they do require direct line-of-sight optical access to the test surface. Although intrusive, thermocouples do not require this design consideration for an experimental rig. Further, the sort of radiation to be measured must be considered when creating a viewport for these two types of experiments. Because the TLCs emit radiation in the visible light spectrum, any material sufficiently transparent to the naked eye is acceptable for use as a viewing window. The IR camera instead measures infrared radiation, which the naked eye does not detect. Thus, materials for an IR window must be carefully chosen to ensure adequate transmission of IR radiation. Examples of IR window materials include quartz [9], sapphire

[18], sodium chloride [16], and zinc selenide [25]. Lawson et al. [9] note that quartz windows are primarily used for near-IR measurements, providing 90% transmission of IR radiation for wavelengths less than $2.3 \mu\text{m}$.

Further difficulties with IR cameras may arise due to the highly complex nature of radiative heat transfer. Incropera et al. [3] state that the net radiative heat flux emitted from a body depends on the emissive power of the body, reflected irradiation, absorbed irradiation, and, depending on the transparency of the body, transmitted radiation. Greiner et al. [32] comment that net radiative heat flux may be portrayed more simply as the difference in the radiation emitted from the surface, $q''_{rad,o}$, and incident on the surface, $q''_{rad,i}$. Most surfaces are optically gray, meaning that their emissivity, ε , lies between 0 and 1. Because most surfaces of interest in the film cooling world are opaque, they also reflect a fraction of the incident radiation according to $1 - \alpha_w$. Surfaces are further assumed to be gray, so α_w is assumed equal to ε_w .

$$q''_{rad} = q''_{rad,o} - q''_{rad,i} = \varepsilon_w(\sigma T_w^4 - q''_{rad,i}) \quad (2-20)$$

Greiner et al. [32] further note that radiative heat transfer processes apply not only to the test article and other solids, but also to any gases that may function as participating media. This is of particular concern in a reactive film cooling environment where other media such as flames and combustion products may be present.

The fundamental methodology employed for IR thermography methods depends on the desired measurement. Tests seeking to characterize η or ϕ alone typically rely on steady methods where the test surface is allowed to reach a stable temperature that is then recorded. The surface temperature is also recorded in conjunction with the freestream temperature and a coolant temperature measured at the appropriate location for η or ϕ . For tests observing η , the initial calculation of η is then corrected to account for assumed 1-D conduction through the

model [4]. No correction is necessary for calculation of ϕ as the conduction is incorporated into the measurement of ϕ based on having a properly scaled Biot number. A great number of experimental works have used IR thermography to measure η , including the studies performed by Albert and Bogard [7], Baldauf et al. [15], and Dittmar et al. [18]. Studies utilizing IR thermography to measure ϕ include those performed by Williams et al. [8], Lawson et al. [9], and Sweeney and Rhodes [25].

IR thermography has also been used to measure heat transfer coefficient distributions. A transient test method developed by Vedula and Metzger [33] for use with TLCs and applied by Ekkad et al. [28] for IR thermography permits simultaneous measurement of adiabatic effectiveness and the heat transfer coefficient. This transient method requires that the model begin at a uniform surface temperature before the freestream temperature is rapidly changed. By modeling the test surface as a semi-infinite solid with transient conduction, the governing equation is reduced to Equation (2-21) with boundary conditions specified by Equations (2-22)-(2-43) [28]

$$k \frac{\partial^2 T}{\partial y^2} = \rho c_p \frac{\partial T}{\partial t} \quad (2-21)$$

$$-k \left. \frac{\partial T}{\partial y} \right|_{y=0} = h(T_{ref} - T_w) \quad (2-22)$$

$$T|_{y=\infty} = T_i \quad (2-23)$$

$$T|_{t=0} = T_i \quad (2-24)$$

where t represents time and the subscript i indicates the initial condition. Solving the governing equation using the specified boundary conditions yields a solution of the form [28]

$$\frac{T_w(t) - T_i}{T_f(t) - T_i} = 1 - \exp\left(\frac{h_f^2 \alpha t}{k^2}\right) \operatorname{erfc}\left(\frac{h_f \sqrt{\alpha t}}{k}\right) \quad (2-25)$$

The surface temperature of the model is recorded in time and two points in time may be used to create a system of equations with two unknowns for a given location. Solution of this system at every point of the test surface provides values for both h and η everywhere on the test surface. While this is a very powerful testing method, it does require the ability to rapidly change the freestream temperature from one steady condition to another.

Lawson et al. [9] developed an infrared thermography method to account for radiation reflected off the test article as well as any radiation measured from the test article surroundings. Their method incorporated a traditional in-situ calibration of the camera using thermocouples, but also included a post-test calibration with a blackbody radiation source. A modified form of the Stefan-Boltzmann Law, shown in Eq. (2-26) was used to determine temperatures based on measured radiation intensity

$$T = mJ^e + b \quad (2-26)$$

where J is measured radiation intensity from the test article and m , e , and b are constants calculated from the blackbody calibration. Reflectivity was measured using a specialized chamber, and the test specimen emissivity was calculated by subtracting the reflectivity from unity.

The radiosity observed by the camera, expressed in Eq. (2-27) [9], includes radiation emitted from the test plate as well as incident radiation from the surroundings reflected by the test plate.

$$J_{cam} = \varepsilon J_w + \rho J_{sur} \quad (2-27)$$

where ρ is the reflectivity of the test specimen and the subscripts *cam* and *sur* refer to the radiation observed by the camera and incident from the surroundings. Special reflective markers were used to determine the incident radiation. With knowledge of the test plate emissivity and

reflectivity, incident radiation from the surroundings, and radiosity measured by the camera, Eq. (2-27) could be solved for the radiation emitted by the test plate, J_w . This value was then used with Eq. (2-26) and the measured constants from the blackbody calibration to calculate surface temperatures. While more accurate than traditional IR calibration procedures, specialized equipment and a great deal of time and effort are required to successfully employ this IR method.

2.5.2. Heat-Mass Transfer Analogy

As previously stated, many film cooling studies have utilized mass transfer methods to measure h and η . The heat-mass transfer analogy is the foundation for mass transfer test methods in film cooling research. This analogy is rooted in the identical form of the governing equations for heat and mass transfer and depends upon the assumption that heat and mass both diffuse in the same manner. The governing equation for heat transfer and appropriate boundary conditions are shown in Equations (2-28) and (2-29) while the governing equation for mass transfer and appropriate boundary conditions are shown in Equations (2-30) and (2-31) [3]

$$u^* \frac{\partial T^*}{\partial x^*} + v^* \frac{\partial T^*}{\partial y^*} = \frac{1}{Re_L Pr} \frac{\partial^2 T^*}{\partial y^{*2}} \quad (2-28)$$

$$T^*(x^*, 0) = 0, T^*(x^*, \infty) = 1 \quad (2-29)$$

$$u^* \frac{\partial C_A^*}{\partial x^*} + v^* \frac{\partial C_A^*}{\partial y^*} = \frac{1}{Re_L Sc} \frac{\partial^2 C_A^*}{\partial y^{*2}} \quad (2-30)$$

$$C_A^*(x^*, 0) = 0, C_A^*(x^*, \infty) = 1 \quad (2-31)$$

where Sc is the Schmidt number ($\mu/\rho D_A$), D_A is a mass diffusion coefficient for species A , C represents species concentration, the subscript A represents a given species, and the superscript asterisks (*) indicate that the variables are nondimensional. Careful inspection reveals that the two governing equations and their boundary conditions are of identical form. Therefore, if the nondimensional parameters and boundary conditions are matched, the results of two separate

tests should be identical. While this is commonly applied in fluid dynamics to account for differences in physical geometry or flow conditions, in this case it permits different testing methods.

The heat-mass transfer analogy is particularly attractive because of the different ways that matter and energy behave physically. Many film cooling studies seek to quantify the adiabatic effectiveness obtained with a given film cooling scheme and operating condition, and this objective is frequently accomplished with thermal measurement techniques and low-conductivity models to approximate an adiabatic wall [22,26]. However, because no substance is truly adiabatic, heat transfer still occurs and this energy transfer must be accounted for. Alternatively, it is generally extremely simple to prevent mass from crossing a boundary. Because the governing equations for heat and mass transfer are both of the same form, mass transfer measurements of an impervious wall effectiveness should directly correspond to the adiabatic effectiveness that would be measured with a thermal test method and a truly adiabatic surface. Thus, mass transfer methods eliminate the conduction related error arising from the non-adiabatic nature of all materials.

However, mass transfer methods remove the contributions of fundamental thermophysical properties such as c_p and k . Thermophysical properties are instead replaced with mass transfer properties, such as the mass diffusion coefficient, D_A . This means that while mass transfer methods may alleviate some of the issues encountered when using thermal test methods, other difficulties may be introduced. Nicoll and Whitelaw [34] note that the governing equations for heat and mass transfer may be regarded as equivalent as long as the total Lewis number, which is the sum of the laminar and turbulent Lewis numbers, is unity. The laminar Lewis number, Le , relates energy and mass diffusion, as is readily apparent in Equation (2-32).

$$Le = \frac{Pr}{Sc} \quad (2-32)$$

This definition makes it readily apparent that if the turbulent Lewis number is unity, both energy and mass diffuse identically. A great number of studies have relied on this assumption and used mass transfer methods, but none of the studies reviewed attempted to validate this assumption. However, Nicoll and Whitelaw [34] explicitly state that existing work supports the assumption of unity Lewis number. A brief discussion further exploring the agreement of thermal and mass transfer methods is presented in Section 2.5.4.

2.5.3. Mass Transfer Measurement Methods

Mass transfer measurements have rarely, if ever, been used to measure ϕ , Δq_r , or $\Delta\phi$, but they have been used fairly extensively to measure h and η . Examples of mass transfer methods include gas chromatography [34], naphthalene sublimation [35], the swollen polymer method [5], and Pressure Sensitive Paint (PSP) [36]. Each of these methods presents unique capabilities and challenges, and each will be discussed in the following text.

Early film cooling research conducted by Nicoll and Whitelaw [34] used a gas chromatography method to measure impervious wall effectiveness, which is equivalent to adiabatic effectiveness as long as the underlying assumptions for the heat-mass transfer analogy, detailed in Section 2.5.2, still hold true. A single slot was used to inject flow parallel to the test surface. Helium gas functioned as a tracer and constituted approximately one percent of the injected gas volume. Samples of the flow immediately adjacent to the test surface were taken downstream of the injection slot and stored for analysis by gas chromatography. By comparing the concentration of helium at the point of injection with some downstream location, the impervious wall effectiveness for that location was calculated. The sampling method for this technique requires a port through which to draw gas samples, allowing only measurements at

discrete points in space. As with thermocouples, the gas chromatography method is very limited in spatial resolution, and more complete distributions of effectiveness were desired.

An early method used extensively to provide more detailed spatial measurements was the naphthalene sublimation method. Coating a test plate in naphthalene and exposing it to freestream flow lacking naphthalene vapor causes the layer of naphthalene on the test plate to sublime away [35]. By measuring the extent of naphthalene sublimation, mass transfer coefficients equivalent to heat transfer coefficients may be determined. Local mass transfer coefficient values may be obtained by precisely measuring the change in naphthalene thickness at a given point, potentially accomplished using a precision depth gauge or holographic interferometry. Goldstein and Jin [37] also presented a method to calculate η using two separate mass transfer coefficient values obtained with pure air and naphthalene-saturated air injection.

Goldstein and Cho [35] stress that the primary advantages of naphthalene sublimation stem from the ability to accurately measure mass transfer coefficient distributions, precisely control boundary conditions, and avoid many of the difficulties inherent in thermal testing. However, this technique is plagued by a plethora of limitations. Naphthalene surface coatings must be reapplied before each test run and must be of uniform thickness, typically accomplished by casting the surface in a mold. Long run times (up to two hours) are required to develop sublimation depths sufficient for accurate measurements, and these thickness changes fundamentally change the shape of the test article. High velocity and high-shear flows (such as impingement jet flowfields) must be avoided to prevent aerodynamic/viscous heating effects and mechanical erosion of the naphthalene coating. Finally, all results obtained are time-averaged values, as the process of naphthalene sublimation occurs so slowly as to prohibit responsive transient measurements.

To mitigate some of the difficulties associated with the naphthalene sublimation method, Ammari et al. [5] utilized a swollen polymer method to make measurements of the heat transfer coefficient over a flat plate with holes inclined at $\alpha = 90$ deg and 35 deg. Elevated density ratios were obtained through anaerobic foreign gas injection using argon and carbon dioxide. The test surface was coated with a silicone polymer impregnated with an organic agent and the thickness of this surface coating was measured at all points using laser holographic interferometry. As the freestream flow passes over the test plate, the organic agent within the polymer transfers into the freestream and the coating thickness changes. The time rate of this change at a given point in space is proportional to the local mass transfer coefficient which is equivalent to the local heat transfer coefficient. While the swollen polymer method provides excellent spatial resolution, it does require a specialized laser setup and a silicone polymer surface coating. This method still suffers from several of the limitations experienced with naphthalene sublimation, such as surface shape change, long run times, and an inability to collect transient data. A new mass transfer measurement technique was required to overcome these prohibitive limitations.

The PSP method succeeded in providing the high spatial resolution of the naphthalene sublimation and swollen polymer methods while eliminating several of the greatest limitations, such as long run times, slow measurement response, and surface shape change. PSP may provide measurement response as rapid as 100 kHz, eliminating the need for long run times and providing the capability to measure transient flow phenomena [38]. The paint is also applied directly to the surface and no mass is lost over time, leaving the surface unchanged. However, the PSP method is not perfect and suffers from issues related to illumination uniformity, temperature dependence, paint photo-degradation, stray light, camera shot noise, poor signal-to-noise ratios in low velocity flows, and model displacement and deformation. Fortunately, the

extensive body of research conducted with PSP has resulted in a number of strategies to minimize these errors.

PSP is fundamentally an oxygen sensor, and careful calibration and measurement permits determination of the partial pressure of oxygen present in the flow. Luminescent molecules (luminophores) are suspended within an oxygen permeable polymer binder and excited by a specific wavelength light (405 nm is commonly used) [38]. As the luminophores relax to their unexcited state, they typically emit a photon of a longer wavelength than the excitation light. However, in the presence of oxygen, this relaxation may occur without radiative emission through a process known as oxygen quenching. More oxygen quenching occurs in the presence of higher partial pressures of oxygen, resulting in reduced emitted light intensity. Other items affecting the luminescent signal from the paint include illumination intensity, luminophore concentration within the paint, paint layer thickness, and detector sensitivity. These spatial variations are accounted for through a ratio of luminescent intensity at an unknown condition to intensity at a known reference condition. This approach, known as the radiometric PSP method, relies on a modified form of the Stern-Volmer equation [38]

$$\frac{J_0}{J} = A(T) + B(T) \frac{P}{P_0} \quad (2-33)$$

where J is emitted light intensity, A and B are constants, and the subscript 0 refers to a reference condition. This equation provides a means to calibrate PSP and also shows the paint's dependence on both temperature and pressure.

The issue of temperature dependence has not been entirely resolved, but the sensitivity of modern paints to temperature fluctuations has been greatly reduced. Binary PSP uses two separate luminophores excited by the same wavelength light: one that is temperature and pressure sensitive, and another that is temperature sensitive and pressure insensitive [38]. The

pressure and temperature sensitive luminophore is known as the signal probe, and the temperature sensitive, pressure insensitive luminophore is known as the reference probe. These two luminophores emit at spectrally distinct wavelengths, permitting collection of two separate images for any given test condition. Figure 2-12 from Wiese et al. [39] schematically shows the difference between a single component and binary PSP. In Figure 2-12, the signal probe emission is referred to as sensitive while the reference probe emission is labeled as reference.

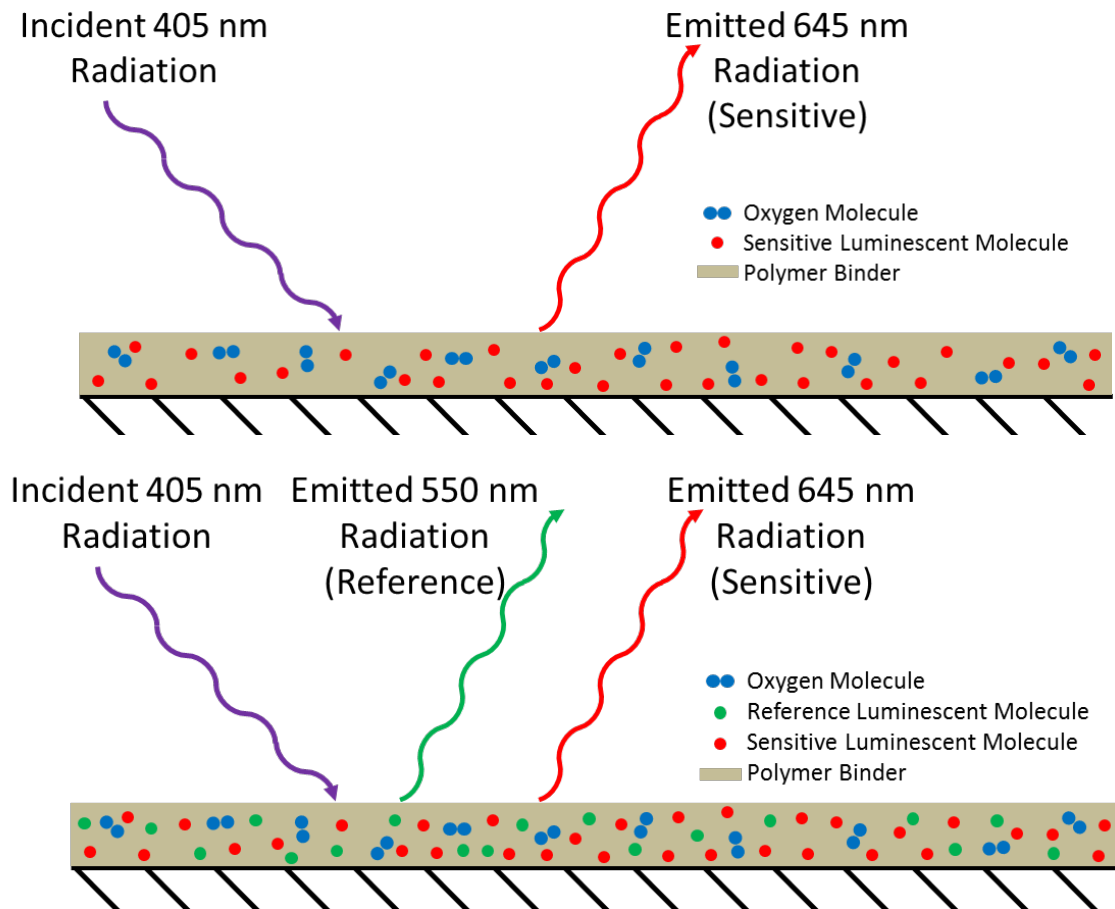


Figure 2-12: Comparison of single component and binary PSP (image from Wiese [39])

A ratio of the responses of the reference and signal probes enables elimination of illumination intensity effects [38]

$$R(P, T, n_S, n_R) = \frac{F_S(P, T)n_S \mathcal{I}}{F_R(P, T)n_R \mathcal{I}} = \frac{F_S(P, T)n_S}{F_R(P, T)n_R} \quad (2-34)$$

where R is the signal to reference ratio, F is the individual luminophore response, n is the number density for the luminophores, $\mathcal{I}$ is local illumination intensity, and the subscripts S and R refer to the signal and reference probes. This ratio fails to remove temperature and luminophore uniformity effects. A ratio of signal to reference ratios at a reference and unknown condition similar to that used in the Stern-Volmer equation of Eq. (2-33) removes probe number density effects [38]

$$S(P, T) = \frac{R_0(P_0, T_0) \frac{n_S}{n_R}}{R(P, T) \frac{n_S}{n_R}} = \frac{R_0(P_0, T_0)}{R(P, T)} \quad (2-35)$$

where S is the system response. While two types of luminophore are required to remove effects from non-uniform illumination, the reference probe also serves to minimize temperature dependence. A single image capture may be used to generate reference and signal probe data by forming images using sensors of only one color with the camera sensor array. With this process, the red sensors produce the signal image while the green sensors produce the reference image.

Using the spectrally distinct luminescence from the two probes, the effect of temperature on luminescence at a given pressure can be mitigated within the calibration. Pressure calibrations at various temperatures enable development of a master calibration that accounts for temperature dependent variations in luminophore response at a given pressure. The binary PSP produced by Innovative Scientific Solutions, Inc. reduces temperature sensitivity to less than 0.5% per K (~ 50 Pa/K) over temperature and pressure ranges spanning from 283K to 313K and from 10kPa to 160kPa respectively. Additional methods to reduce temperature variation include use of a controlled temperature wind tunnel and highly conductive materials for model construction.

While the binary PSP method eliminates non-uniform illumination effects and greatly reduces temperature sensitivity, other issues still persist. The ratio of ratios in Eq. (2-35) needed to generate system response doubles camera shot noise because four images are required as opposed to just two images for a single component PSP [38]. Averaging a large number of images may reduce the camera shot noise to an acceptable level. PSP also degrades with exposure to the excitation light wavelength. This requires dark model storage and minimized illumination duration while testing to ensure maximum paint life. Model displacement and deformation effects may be minimized with static placement for test equipment and a robust model and model anchoring system. Finally, stray light is accounted for by subtracting the influence of ambient light from all test images. Dark frame subtraction will be discussed as part of the PSP test methodology in the following paragraphs.

Using the heat-mass transfer analogy, η may be represented as concentration, specifically oxygen concentration as applied for PSP [36]

$$\eta = \frac{T_{\infty} - T_{aw}}{T_{\infty} - T_c} \approx \frac{C_{\infty} - C_w}{C_{\infty} - C_c} = 1 - \frac{C_{O_2,mix}}{C_{O_2,air}} \quad (2-36)$$

where the O_2 subscript indicates oxygen and the subscripts *mix* and *air* represent cases with foreign gas and air injection, respectively. The concentration of oxygen is related to the partial pressure of oxygen through mole fractions and molecular weights, allowing Eq. (2-36) to be rewritten as [36]

$$\eta = 1 - \frac{1}{\left(\frac{P_{O_2,air}}{P_{O_2,mix}} - 1\right) \frac{MW_{fg}}{MW_{air}} + 1} \quad (2-37)$$

where P_{O_2} represents the partial pressure of oxygen, MW represents molecular weight, and the subscripts *fg* and *air* represent a foreign gas and air, respectively. The foreign gas used must be anaerobic with minimal oxygen content to provide a usable signal. This anaerobic gas displaces

oxygen near the test surface and prevents the oxygen quenching reaction, thus increasing the observed light intensity from the PSP.

A series of images is collected for both the foreign gas and air injection cases to generate a pressure field [36]. A reference condition, known as the “wind off” case, is defined and an image collected with excitation illumination, no freestream flow, and no coolant flow. An experimental condition is also required, known as the “wind on” case, and the appropriate image is collected with excitation illumination depending on what sort of pressure field is to be determined. Eq. (2-37) requires pressure fields for both air and foreign gas injection, so the wind on case requires freestream flow with either air or foreign gas injection. To account for stray light, an image is captured with no freestream or coolant flow and no excitation illumination. This “background” image is then subtracted from both the wind off and wind on images to isolate the light emitted from the PSP from any ambient light

$$\frac{I_0 - I_b}{I - I_b} = f\left(\frac{P_{O_2}}{P_{O_2,0}}\right) \quad (2-38)$$

where the subscript b refers to the background image. Eq. (2-38) is thus characterized for a given PSP over a range of pressures and temperatures. The resulting calibration is then used to calculate the necessary pressure fields for calculation of η as found in Eq. (2-37).

2.5.4. Comparison of IR and PSP Measurement Methods

A large body of research has been conducted using the PSP method and many of the available studies have produced trends consistent with results obtained via thermal test methods. Han and Rallabandi [40] conducted a review of the available gas turbine film cooling literature utilizing the PSP method. The “classical” trends observed with thermal methods were confirmed with PSP methods. These trends include phenomena such as jet separation with increased

coolant mass flux, resistance to coolant jet separation with shaped hole exits, decreased effectiveness with increased freestream turbulence, and increased effectiveness with compound angle and hole exit shaping. However, while overall trends were confirmed, direct comparison of η values obtained using IR thermography and PSP methods has demonstrated some disagreement between the two methods.

Wright et al. [31] performed a study on a single cooling scheme using both IR thermography and PSP measurement methods. The cooling scheme consisted of seven cylindrical cooling holes of diameter $d = 4$ mm oriented at $\alpha = 30$ degrees with a 45 degree compound angle and spacing of $P = 3d$. These cooling holes were placed in a flat plexi-glass test specimen. PSP measurements were made using nitrogen gas at room temperature for coolant, while the IR measurements used air heated to 43.3 °C. These conditions correspond to density ratios of $DR = 0.97$ and 0.93 for PSP and IR thermography conditions, respectively. Experimental conditions targeted $M = 0.6$ and 1.2 , and a comparison of $\bar{\eta}$ values obtained at these conditions was made. For both cases, PSP showed a region near the hole where the coolant jet detached and then reattached to the surface. While the IR results failed to show this phenomena, they also consistently measured greater $\bar{\eta}$ values than PSP for $M = 0.6$. On the contrary, good agreement between PSP and IR measurements was seen at $M = 1.2$ everywhere except the near hole region and far downstream of the hole ($x/d > 15$).

While the data of Wright et al. [31] suggest that PSP is the superior method for observing jet liftoff and reattachment, coolant jet liftoff is not usually observed until $I > 1$. The $M = 0.6$ PSP condition for which coolant jet liftoff was observed resulted in $I = 0.58$, suggesting the coolant jet should remain attached. Wright et al. also correctly state that PSP is preferable to IR thermography because it eliminates any potential error from heat transfer through the test

specimen. However, data accuracy using the PSP method relies on the validity of the heat-mass transfer analogy, underlying assumptions for the PSP method, and rigorous application of the PSP method.

However, Wiese et al. [39] found a very different trend when comparing their η values obtained with PSP and IR thermography. A representative turbine airfoil leading edge model consisting of a semi-cylinder followed by a flat afterbody was used. A single cooling hole of $d = 5.08$ mm oriented with a 90 degree compound angle and surface injection angle of $\alpha = 20$ degrees was placed at $\gamma = 21.5$ degrees from the stagnation line on the leading edge of the semi-cylinder. A variety of different gases (nitrogen, argon, and carbon dioxide) was used for both IR and PSP tests with identical temperature conditions ($T_c = 295$ K, $T_\infty = 315$ K) imposed for all tests. Air was also used as coolant for IR tests. Momentum flux ratio was selected as the primary parameter of interest, with tests targeting $I = 0.5$ and 1.0 . Comparison of η values across the model span at $x/d = 5$ was performed. PSP results still revealed phenomena not readily apparent in IR thermography results, such as bifurcated coolant jets, but PSP results were consistently greater than those obtained using IR thermography. Some speculation on the root cause of this difference was offered, suggesting that low turbulence levels resulted in a non-unity total Lewis number, allowing energy to diffuse comparatively better than the mass diffusion of the anaerobic gases. The nature of PSP as an oxygen sensor was also cited as a potential cause for these elevated effectiveness values, as energy may have diffused through the flow via molecular collisions to warm the surface while the physical mass (oxygen) may not have diffused to the surface. This would cause elevated intensity for the PSP tests and produce inflated η values.

It is worthwhile to note that the experimental setup and conditions of the studies conducted by Wright et al. [31] and Wiese et al. [39] differed greatly. Aside from the obvious differences in model shape and cooling arrangement, each study used a different material for its low-conductivity thermal model. Wright et al. used plexi-glass while Wiese et al. used a low conductivity foam, providing thermal conductivity values of $k \approx 0.17$ [41] and $0.03 \frac{W}{m \cdot K}$ [42] respectively. The elevated thermal conductivity of the model used by Wright et al. certainly calls into question any conclusions drawn regarding the accuracy of IR measurements as compared to PSP measurements. It is also unclear whether or not Wright et al. performed a correction to account for the conduction through their test plate while Wiese et al. explicitly state the uncertainty in their conduction correction. If no conduction correction was performed, the IR measurements of Wright et al. cannot be relied upon for meaningful comparison with PSP data. Further, different data presentation methods make it difficult to effectively compare results between the two studies. The results presented by Wiese et al. fail to account for the entire flowfield by focusing at a single downstream location. The culmination of all these issues indicates that neither study conclusively proves that one experimental method provides more accurate results than the other. However, as discussed in Sections 2.5.1 and 2.5.3, each method provides unique benefits that suit different experimental objectives. The intended objective should be critically analyzed before deciding on a measurement technique.

2.6. Scaling

Avoiding the high temperatures and pressures associated with actual gas turbine engines offers great flexibility in measurement techniques and experimental setup. As such, a great abundance of film cooling research has been performed at near ambient conditions. However, very little work has been conducted to assess how accurately measurements made at near

ambient conditions predict film cooling performance at engine conditions. The following discussion focuses on the effect of various nondimensional parameters on scaling film cooling performance and an existing method used to scale ϕ from ambient to engine conditions known as the matched Biot method.

2.6.1. Nondimensional Parameter Effects on Scaling

Nondimensional parameters are used to ensure that results obtained at a specific flow condition may be universally applied to other conditions. Several relevant nondimensional parameters have been used to match film cooling studies to engine conditions, including DR , VR , M , I , Re_∞ , and Re_c . Of these, DR , M , I , and Re_∞ have received a majority of the attention. A brief review of studies examining these parameters is offered next, followed by discussion of less conventional scaling parameters.

DR has been studied as an independent parameter, primarily investigating the effects of DR on η . Eberly and Thole [11] also investigated DR effects using a steady-state infrared thermography technique. The cooling geometry was a flat plate with a single row of five cylindrical holes inclined at 30 degrees to the surface and spaced at $P/d = 6.7$. The values of DR studied included 1.2 and 1.6. For all blowing ratios studied ($M = 0.6, 1.0$), it was found that a higher value of DR provides improved centerline effectiveness. The higher DR cases also provided enhanced coolant spreading. These improvements were attributed to the reduced I values attained with a higher DR value, reducing coolant jet penetration into the freestream and allowing the coolant to better spread and cover the film-cooled surface.

Narzary et al. [36] performed a similar study on a representative turbine vane model using a Pressure Sensitive Paint (PSP) technique. The model of Narzary et al. utilized a total of nine rows of cylindrical cooling holes with four rows on the vane suction side, 2 rows on the

vane pressure side, and 3 rows in the leading edge region, known as the showerhead. The showerhead rows were situated such that one row aligned with the vane stagnation line and the other two rows were located 15 degrees to either side of the stagnation line. Pressure side rows were located at $C_x = 0.17, 0.47, 0.66$, and 0.85 while the suction side rows were located at $C_x = 0.18$ and 0.59 . All subsequent rows in a given region were staggered such that the holes in a following row aligned with the middle of the space between holes from the preceding row. Holes in the showerhead region were oriented 30 degrees to the surface and normal to the streamwise direction while both pressure and suction side holes were oriented 45 degrees to the surface and at 45 degrees to the streamwise direction. All showerhead rows consisted of 22 holes while the pressure and suction side rows consisted of alternating rows of 23 and 22 holes. Hole spacing was uniform for all rows at $P/d = 8.2$. While Narzary et al. utilized a different experimental method and model than Eberly and Thole, their results indicated the same general trends. This suggests that coolant density has a very profound impact on film cooling performance and stresses the importance of matching DR in film cooling experiments.

This result was reaffirmed in a computational study conducted by Greiner et al. [43] modeling a single fan-shaped, laidback cooling hole placed in an adiabatic surface. To investigate the importance of various nondimensional parameters in scaling film cooling performance, a baseline case of η was calculated at engine temperature ($T_\infty = 1829$ K, $T_c = 625$ K, $M = 0.946$). Subsequent cases utilizing near ambient conditions and various combinations of matched nondimensional parameters then calculated η distributions and were compared with the results from the baseline case. These near ambient cases were divided into four categories: matching M and Re ; matching M and DR ; matching flow and property ratios using fictitious gas properties; and matching M , DR , and Re with real gas properties. Each of the cases matching Re

with real gas properties investigated the effect of matching the coolant and freestream Reynolds numbers separately, as fixed length scales and thermophysical property variations prohibit simultaneous matching of both Reynolds numbers.

Greiner et al. [43] found the best match between η from ambient conditions using real gas properties and η from engine conditions when M , DR , and Re were matched. When only M and Re were matched, a near unity value of DR prevented other parameters such as I from being matched and resulted in an underprediction of η . Matching M and DR alone caused elevated freestream and coolant Re values resulting in an overprediction of η consistent with the increased momentum present in the freestream and coolant flows. For the cases with fictitious gas properties, the final case indicated that simultaneous matching of M , DR , and Pr and Re for both coolant and freestream flows provides an excellent match to engine conditions, but could not be replicated in a laboratory and was thus set aside. An earlier fictitious gas case where flow and property ratios alone were matched produced inaccurate centerline values of η . An attempt to correct this deficiency by matching all flow and property ratios as well as Re for both coolant and freestream flows resulted in unrealistic Pr values and completely failed to match the baseline η values. This case serves to stress the pivotal role that Pr plays in scaling film cooling performance.

For real gas properties, it was found that matching M , DR , and Re_c produced the best overall agreement with engine conditions in terms of local η values [43]. Alternatively, matching Re_∞ instead of Re_c provided superior agreement in terms of $\bar{\eta}$ values. Averaging these two cases provided the best prediction of η at engine conditions of any of the ambient cases considered. The benefit of this averaging process was slight for the $M = 1$ cases, but an additional case at $M = 3.0$ showed that this averaging provides much greater benefit when

dealing with severely separated coolant jets. Additional cases at $M = 0.5$ and 3.0 also served to reinforce the finding that matching M , DR , and Re best scales η from ambient to engine conditions.

An additional investigation by Greiner et al. [44] studied the scaling of η from ambient to engine conditions in the presence of a thermal boundary layer as opposed to the adiabatic surface examined in the previous study. The same computational model was used, but a different methodology was developed to calculate η with a conductive boundary condition. Profiles of q_f'' were calculated for ten different cases utilizing various isothermal wall temperatures. By interpolating the q_f'' results to determine the point at which $q_f'' = 0$, T_{aw} and thus η were determined. Cases at near ambient conditions matched to the baseline case at engine conditions specifically focused on the parameters that were found to best scale η with an adiabatic boundary condition (M , DR , and Re). It was confirmed that matching Re_c provides the best match between ambient and engine conditions for both an adiabatic and isothermal boundary. Because all of the work conducted by Greiner et al. was computational, experimental work is desperately needed to validate or refute these findings.

A computational study by Rutledge and Polanka [45] using the same model geometry as Wiese et al. [39] studied the influence of less commonly considered scaling parameters such as the advective capacity ratio, dynamic viscosity, specific heat, and thermal conductivity. The advective capacity ratio, ACR , is defined as [45]

$$ACR = \frac{\rho_c U_c c_{p,c}}{\rho_\infty U_\infty c_{p,\infty}} = M \frac{c_{p,c}}{c_{p,\infty}} \quad (2-39)$$

A baseline case of η values was established using air as a coolant and properties for other gases were used to generate η values for comparison to the baseline case. Each of the previously mentioned scaling parameters was matched separately to provide η values for comparison. ACR

was found to best scale the magnitude of η while I was found to best scale the coolant jet trajectory. Matched M provided the best agreement in peak η values but failed to correctly place the coolant jet trajectory. Thus, neither M nor I perfectly scaled the hydrodynamics of the flowfield, but I provided the best match for thermal boundary conditions. Individual property effects were isolated by matching all gas properties to the air baseline with the exception of a single property. Density was found to cause the greatest disturbance in peak η values of any property considered. Varying thermal conductivity and specific heat caused a lesser effect and altering the dynamic viscosity had minimal impact. Experimental measurements supporting these claims were obtained by Wiese et al. [39] and are described next.

Experimental measurements supporting these claims were obtained by Wiese et al. [39] Their study comparing η values obtained using PSP and IR thermography methods also investigated the effect of varying flow and property ratios in scaling η . Several gases were used to vary flow and property ratios at near ambient conditions and measured distributions of η were compared for these varied flow and property ratios. It was found that the location of coolant jet effectiveness was best scaled by I while the magnitude of η was best scaled by ACR . ACR also scaled the location and shape of the effectiveness region well but was inferior to I in this respect. Wiese et al. [39] comment that ACR represents the amount of heat that may be absorbed by the coolant gas, potentially preventing this energy from reaching the cooled surface. While the traditionally held claim that M scales the thermal transport capacity of the coolant (Bogard and Thole [4]) still holds, the results of Wiese et al. suggest that ACR may be a better parameter to scale coolant thermal transport capacity. This is especially true because ACR accounts for variation in specific heat between the coolant and freestream gases that may be caused by a difference in gas species or temperature variation.

2.6.2. Matched Biot Method

Many studies alleviate issues associated with matching internal heat transfer phenomena to engine conditions by measuring η instead of ϕ , but this provides no direct knowledge of turbine airfoil surface temperatures. As mentioned in Section 2.1.1, engine designers may numerically estimate T_w using η and other relevant flow parameters [7], but measuring ϕ and correctly matching to engine conditions provides direct knowledge of T_w . However, accurately scaling the heat transfer processes present at engine conditions to a near ambient test presents an immense challenge.

In order to correctly match heat transfer phenomena in a turbine blade to actual engine conditions, several parameters must be matched. The first of these parameters to be considered was the Biot number, defined using the external heat transfer coefficient and thickness of the turbine airfoil, $Bi = h_f k / t$. Relying on the analytical work of Martiny et al. [10], Sweeney and Rhodes [25] determined that correctly matching Bi for a near ambient test is critical to obtain results directly scalable to engine conditions. To correctly match Bi to engine conditions, Sweeney and Rhodes considered the decrease in convective heat flux into their effusion cooled combustor liner when shifting from engine conditions to their planned experimental conditions. They offset this threefold decrease in heat flux by constructing their test specimen out of titanium, providing thermal conductivity approximately 2.5-3 times less than typical turbine materials. By appropriately decreasing thermal conductivity based on changing heat flux conditions and maintaining the same thickness, the Biot number remained matched to engine conditions.

The work of Albert and Bogard [7] further stressed the importance of matching Bi , but also introduced other parameters. By assuming 1-D heat transfer through the wall, Albert and Bogard arrived at a relationship between η and ϕ , shown below in Eq. (2-40) [7]

$$\phi = \frac{1 - \eta}{1 + Bi + h_f/h_c} + \eta \quad (2-40)$$

The addition of the h_f/h_c term stresses the importance of correctly matching the relative contributions of internal and external cooling. This coincides directly with the importance of Bi in correctly scaling the relationship between convective heat transfer into the turbine airfoil and conductive heat transfer through it. While Eq. (2-40) has primarily been used to stress the most important parameters to match when scaling ϕ from near ambient to engine conditions, it could conceivably provide a means to estimate ϕ from a known η or η from a known ϕ .

Albert and Bogard [7] acknowledge that the assumption of one-dimensional heat transfer is not valid in regions with three-dimensional heat transfer effects such as near film cooling holes or internal airfoil walls. However, they assert that correctly matching the experimental condition to the engine condition will yield the same ϕ distribution. This requires matching the distributions of Bi and η values across the airfoil, the ratio of heat transfer coefficients, and the relevant internal and external geometries of the airfoil. Furthermore, engine condition flows entail much higher velocities and gas temperatures than are commonly experienced in most film cooling experiments. High velocities may be corrected for using the relevant recovery temperatures in the definitions for ϕ and η ($T_{c,r}$, $T_{\infty,r}$, $T_{aw,r}$). In most cases, this does not affect the experimental condition, but does alter the interpreted temperature values at the engine condition. Albert and Bogard note the temperature differences between experimental and engine conditions create two separate difficulties: thermal property temperature dependence and large thermal property variations in the flowfield between the surface and freestream fluid.

Thermal property variations ultimately affect the ratio of heat transfer coefficients. Using Nusselt number correlations, Albert and Bogard [7] showed that the ratio of external-to-internal heat transfer coefficients can be represented as

$$\frac{h_f}{h_c} = \frac{\left(\frac{Nu_x k}{x}\right)_f}{\left(\frac{Nu_{D_h} k}{D_h}\right)_c} = \frac{\left(\frac{C_1 Re_x^a Pr^b k}{x}\right)_f}{\left(\frac{C_2 Re_{D_h}^m Pr^n k}{D_h}\right)_c} \quad (2-41)$$

Where C_1 , C_2 , a , b , m , and n represent the appropriate constants for Nusselt number correlations. A ratio may be formed by evaluating Eq. (2-41) at both experimental and engine conditions. If the experimental facility was carefully designed and temperature dependence for Pr is assumed to be negligible, the constants in the ratio of engine to experimental conditions divide out and the ratio reduces to

$$\frac{\left(\frac{h_f}{h_c}\right)_{engine}}{\left(\frac{h_f}{h_c}\right)_{model}} = \frac{\left(\frac{k_f}{k_c}\right)_{engine}}{\left(\frac{k_f}{k_c}\right)_{model}} \quad (2-42)$$

Because Eq. (2-42) is primarily a function of temperature, scaling from ambient to engine conditions introduces an appreciable error. Citing data for the F100-PW-100 turbofan engine ($T_{aw} = 1681\text{ K}$, $T_c = 819\text{ K}$) and their experimental rig ($T_{aw} = T_\infty = 305\text{ K}$, $T_c = 220\text{ K}$), Albert and Bogard [7] calculated a maximum error in ϕ of 0.04.

Additionally, the large temperature difference between the freestream fluid and turbine blade surface alters the relationship between the Nusselt number and Reynolds number for the flow. This variation may be accounted for by calculating a new value for Nu using the property ratio method of Kays et al. [6]

$$\frac{Nu}{Nu_{CP}} = \left(\frac{T_w}{T_\infty}\right)^n \quad (2-43)$$

where Nu_{CP} is the Nusselt number evaluated for the constant property solution. In this case, Nu_{CP} should be evaluated at T_∞ . Albert and Bogard [7] note that this variation in Nu can alter the ratio of heat transfer coefficients between engine and experimental conditions at a specific location, leading to additional error in ϕ . However, for typical engine conditions, this error is about an order of magnitude less than the error incurred by mismatching k_f/k_c between engine and experimental conditions.

While Albert and Bogard [7] presented a very powerful relation in Eq. (2-40), their work focused primarily on experimentally comparing the performance of two cooling hole schemes on a turbine vane pressure side. Virtually no work was performed to test the validity of Eq. (2-40) in a more realistic operating regime. Williams et al. [8] took the next step to assess the validity of Eq. (2-40) and the matched Bi technique. They also accounted for the heat gained by the coolant as it passes through the cooling holes, presenting a modified form of Eq. (2-40).

$$\phi = \frac{1 - \chi\eta}{1 + Bi + \frac{h_f}{h_i}} + \chi\eta \quad (2-44)$$

Where χ is the coolant warming factor, defined as:

$$\chi = \frac{T_\infty - T_{c,e}}{T_\infty - T_{c,i}} \quad (2-45)$$

As previously mentioned, this relation between ϕ and η may be rearranged to yield η as a function of ϕ , shown in Eq. (2-46). If not accounting for coolant warming, such as in Eq. (2-40), the coolant warming factor, χ , is simply omitted from the denominator.

$$\eta = \frac{\phi \left(1 + Bi + \frac{h_f}{h_i} \right) - 1}{\chi \left(Bi + \frac{h_f}{h_i} \right)} \quad (2-46)$$

Finally, Williams et al. [8] utilized Eq. (2-44) to make predictions of $\bar{\phi}$ using measured values of $\bar{\eta}$. Rearranging Eq. (2-44) allows prediction of $\bar{\phi}$ based on overall effectiveness without film cooling, ϕ_0 , and η .

$$\phi_0 = \frac{1}{1 + Bi + \frac{h_f}{h_i}} \quad (2-47)$$

By defining ϕ based on internal cooling alone, Eq. (2-44) may be redefined as:

$$\phi_p = \phi_0(1 - \chi\eta) + \chi\eta \quad (2-48)$$

where the subscript p refers to predicted values. Williams et al. found that $\bar{\phi}$ predictions were greater than $\bar{\phi}$ measurements for a well attached cooling jet ($I = 0.38$). As the momentum flux ratio was increased, good agreement was found between prediction and measurement ($I = 1.0$). As momentum flux ratio was increased to the point of coolant jet lift off ($I = 2.46$), the predicted $\bar{\phi}$ values were found to be less than the measured $\bar{\phi}$ values.

These results are in contrast to the work of Dees et al. [46] who found relatively good agreement between prediction and measurement. However, several differences exist between the studies performed by Dees et al. and Williams et al. The model used by Dees et al. incorporated a U-bend cooling channel with internal rib turbulators while Williams et al. used a radial channel with impingement cooling. Dees et al. utilized CFD to perform their predictions and only reported data for their model with internal rib turbulators and not their model with smooth internal walls. Also, Dees et al. performed a single test point for which they reported only coolant and freestream Reynolds numbers. All of these differences make comparing the works of Dees et al. and Williams et al. very difficult. Regardless, further work examining the validity of the relationship in Eq. (2-44) is certainly well warranted.

3. Experimental Methods

This investigation sought to address the issue of scaling film cooling performance from near ambient to engine representative temperatures. In support of the overall objectives investigating physical model size, scaling film cooling performance, and making effectiveness predictions, the objectives of Section 1.3 were achieved. New experimental capabilities were developed in the form of near ambient thermal and PSP measurement methods. IR calibration procedures for both engine representative and near ambient conditions were developed and analyzed prior to collecting film cooling performance metrics at near ambient and engine representative conditions.

Experiments for this investigation were conducted to examine the importance of matching various parameters in scaling film cooling performance. Based on the computational results of Greiner et al. [43, 44], the freestream Reynolds number and coolant Reynolds number received a great deal of attention in addition to the density ratio and blowing ratio. Two separate experimental facilities at Wright-Patterson Air Force Base (WPAFB), OH were to be used to investigate physical model size. The first is the Film Cooling Rig (FCR) in the Combustion Optimization and Analysis Laser Laboratory (COAL Lab) located at the Air Force Institute of Technology (AFIT). The second is the Leading Edge Film Cooling Tunnel (LEFCT) located in the Heat-Transfer and Aerothermal Lab (HAL) operated by the Air Force Research Laboratory (AFRL). Both facilities utilized the PSP method and geometrically similar models to measure adiabatic effectiveness.

Detailed discussion of both experimental facilities is provided in the ensuing sections. Emphasis is placed on the specific experimental setups used for this investigation. Challenges encountered during experimentation are presented with recommendations to improve rig

performance or avoid the specific difficulties encountered. Data reduction methods and experimental uncertainty are also discussed.

3.1. Film Cooling Rig

The AFIT FCR, shown in Figure 3-1, was designed by Shewhart [47] and Lynch [48], but Lynch alone performed the construction and initial testing of the rig. A brief outline of its various parts and features will be presented here, but further details may be found in Lynch [48]. Additionally, a detailed flow diagram of all supporting equipment and instrumentation is presented in Appendix A. This rig was made up of several key components including the Well-Stirred Reactor (WSR), the transition section, the main test section, and the airfoil test plate, all of which are shown in Figure 3-1.

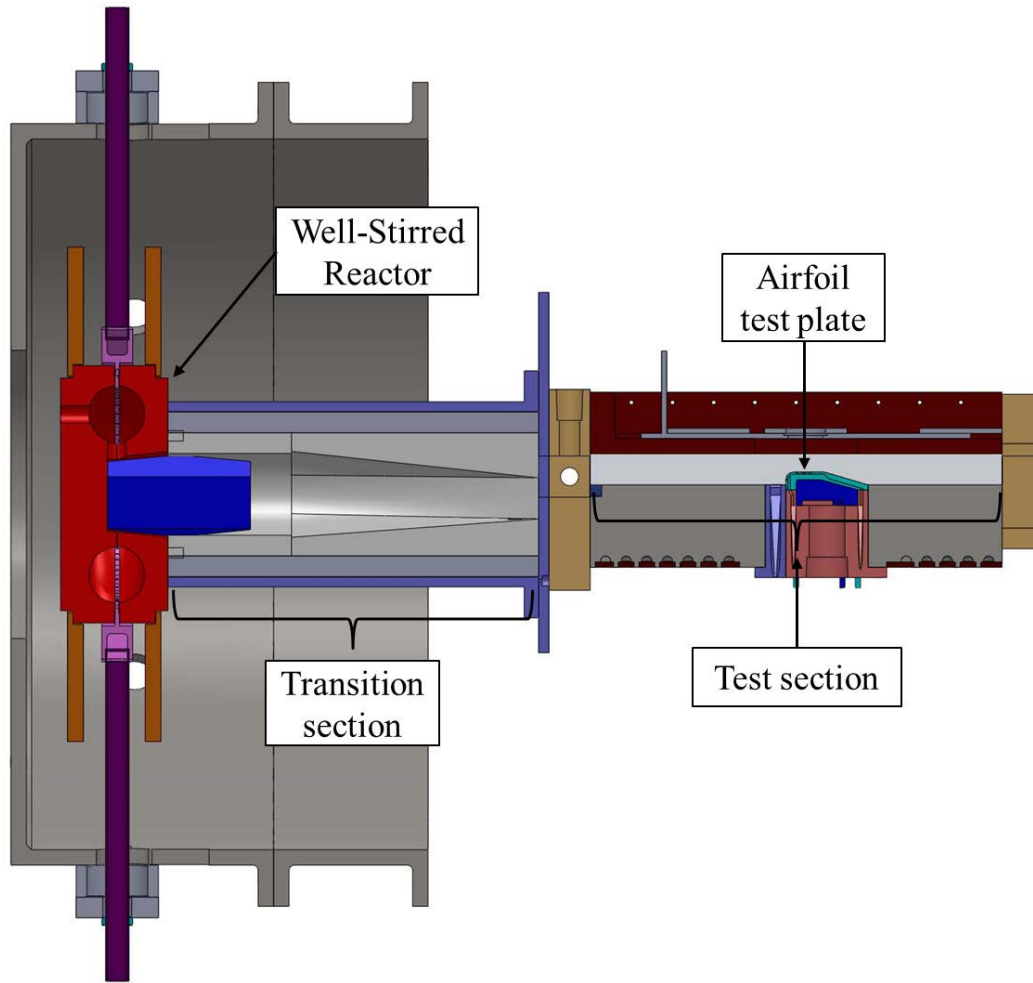


Figure 3-1: AFIT Film Cooling Rig and Well-Stirred Reactor cross-section

The FCR was designed specifically to allow testing of various film cooling schemes at a wide variety of conditions. Several of the FCR features in Figure 3-2 are highlighted to demonstrate the flexibility of the FCR. The front block opposite the test plate adjusts to allow variation of the Mach number without changing the mass flow through the test section. Modular design of the cooling block assembly permits various cooling geometries, such as a flat plate or representative turbine airfoil. Although primarily intended for high temperature testing fed by WSR combustion products, the FCR could also perform low to moderate temperature tests by

relying on existing COAL Lab equipment. For these tests, no fuel was introduced to the flow, so the WSR functioned only as a manifold.

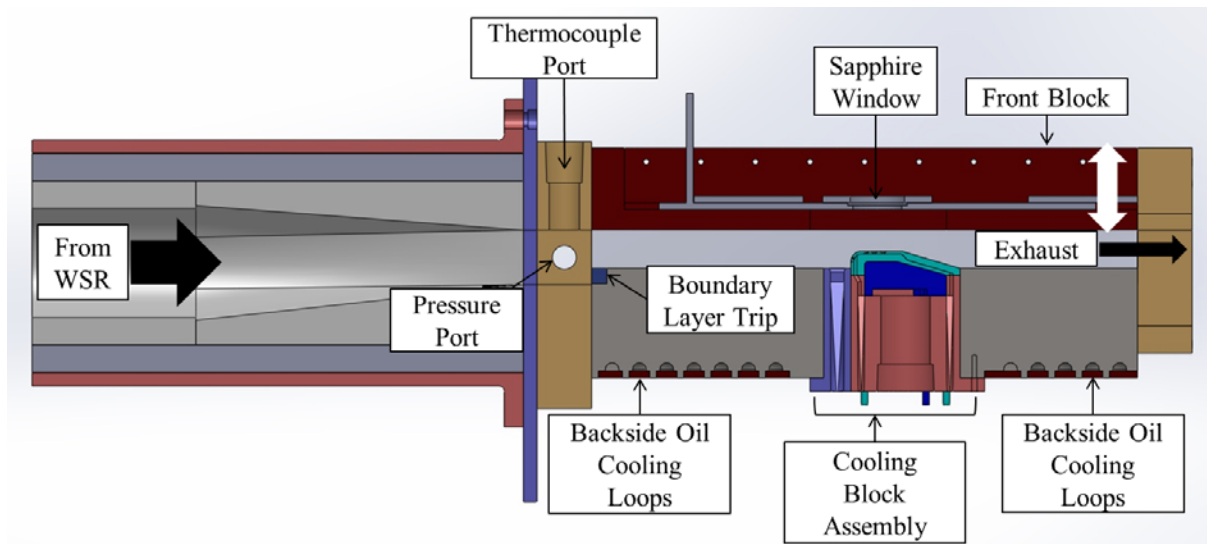


Figure 3-2: AFIT Film Cooling Rig cross-section

The FCR was mounted atop the transition section exiting the WSR. Its weight was primarily supported by the steel chimney housing the transition section ceramics. Four threaded rods, shown in Figure 3-3, provided additional support and served to stabilize the FCR. A total of four bolts affixed the FCR blocks to the baseplate with two bolts per block. The through-holes in the front block were cut as slots, allowing flexibility in the distance between the two blocks. Countersunk through-holes in the baseplate allowed mounting bolts to fix the main block to the baseplate while still maintaining a flow path seal between the baseplate and transition plate. The upper bracket helped maintain the spacing between the main and front blocks while also serving to keep the side plates in place. Difficulties were encountered with the existing system used to secure the upper bracket and main block and will be discussed in Section 3.4.3. The quartz window shown in Figure 3-3 occupied the side plate slot and provided optical access to the test section. For tests not requiring optical access, a metal side plate was used.

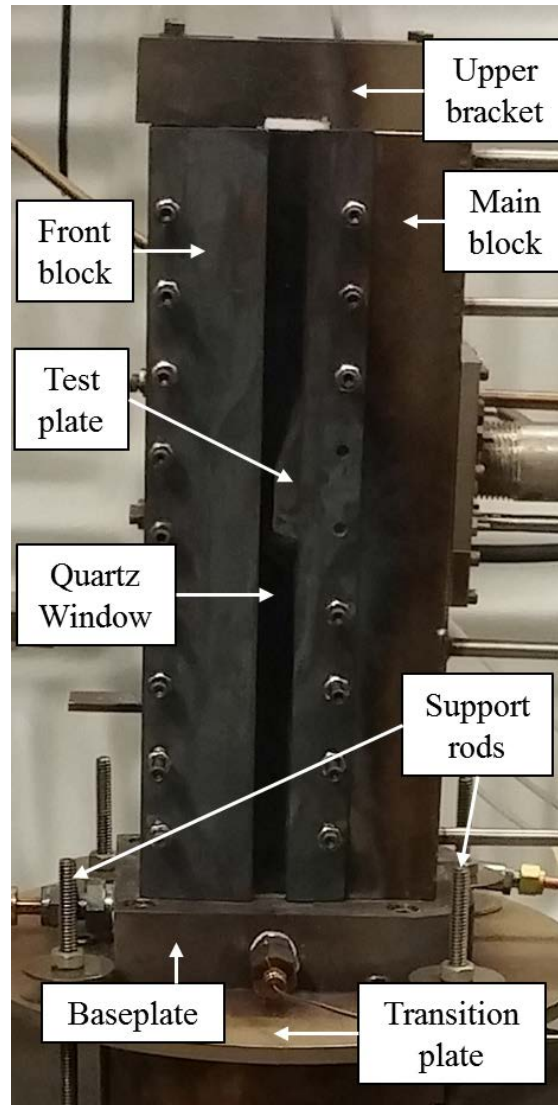


Figure 3-3: FCR side view

The oil cooling loops highlighted in Figure 3-2 were fed with heated oil from a 3 kW Mokon HTF 350 series heated oil system. This system afforded control over the main block surface temperature that would otherwise be exposed to ambient conditions and free convection. For this work, the intent was to control the far-field boundary condition to eliminate variability arising from ambient conditions. As in the case of other rigs designed for high temperature film cooling, for instance DeLallo [29], this could provide control over the wall temperature in the

test section. For all high temperature thermal tests the system was set to run at 422 K (300 °F setting on the machine).

The primary instrumentation in the FCR is shown in Figure 3-4. Both the inlet and test section thermocouples were used extensively in calculating performance metrics. The exit thermocouple was primarily used as a diagnostic measurement. However, it was also used to estimate the freestream temperature through the FCR as a function of downstream position.

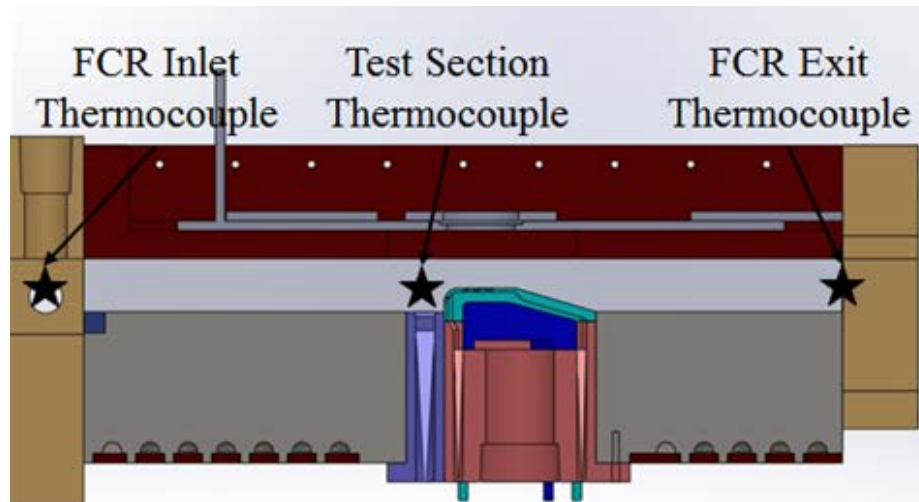


Figure 3-4: Key FCR instrumentation

3.1.1. Well-Stirred Reactor

For high temperature testing, a Well-Stirred Reactor (WSR) was used to supply the FCR with combustion products to more accurately simulate the operating environment of a high pressure turbine. The WSR, depicted in Figure 3-5, was based on the design of Nenniger et al. [49] with modifications implemented by Stouffer et al. [50] to generate conditions better representing the inlet to a high pressure turbine. The exploded view of Figure 3-5b shows the parts comprising the WSR and their assembly. The two toroid halves resided above and below a jet ring manifold sandwiched between the toroid halves. Two large steel plates situated above and below the toroid halves served to clamp the whole reactor assembly together. Springs placed

on the bolts clamping the two plates together allowed the WSR to expand under heating without creating excessive thermal stresses.

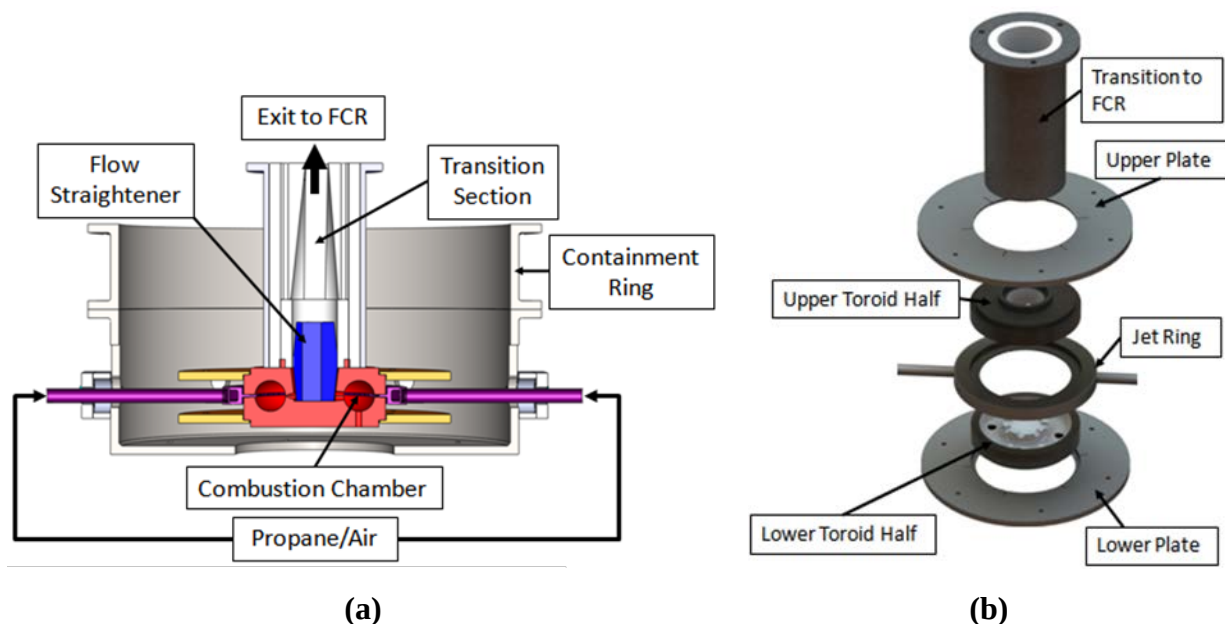


Figure 3-5: AFIT Well-Stirred Reactor diagrams: (a) side view cross-section and (b) exploded view (images from Lynch [48])

The 250 mL toroidal combustion chamber, shown in Figure 3-5a, was supplied with premixed air and propane at a controlled equivalence ratio. Reactants were fed into the combustion chamber through the jet ring labeled in Figure 3-5b. These reactants then combusted and combustion products exited the combustion chamber through eight holes evenly spaced around the inner circumference of the toroid. The flow then passed through a flow straightener at the exit of the combustion chamber and up through the ceramic transition stack, shown in Figure 3-5a. To accommodate the circular exit of the WSR and the rectangular entrance to the FCR, the transition section gradually shifted from the 2,027 mm² circular WSR exit area to the 1,290 mm² rectangular FCR entrance area.

In addition to the flow straightener, the transition section from the WSR to the FCR was comprised of three separate elements, all shown in Figure 3-6. These included two types of

ceramics and a steel chimney to house the ceramics. The flow transition was accomplished by 6 2.54 cm (1.0 inch) thick ceramic discs machined to gradually transition from the circular WSR exit to the rectangular FCR inlet. A 7.62 cm (3.0 inch) long circular ceramic cylinder was placed below these discs to accommodate the overall length of the transition section. The steel chimney housed the ceramics and also supported the weight of the FCR. The transition plate mounted atop the chimney provided mounting points for the body of the FCR itself.

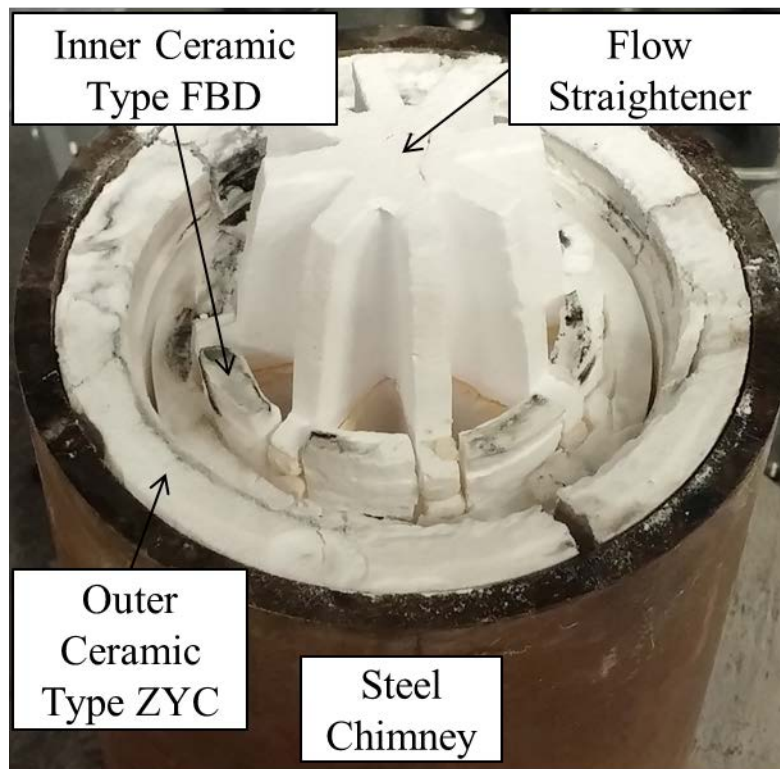


Figure 3-6: Inverted view of WSR transition section

Two types of zirconia ceramics, both supplied by Zircar Zirconia, Inc., were used for the transition section based on their material properties. A harder zirconia ceramic (Type FBD) was used for the inner portion of the transition section based on its reduced propensity to erode in high flow conditions as well as its machinability. However, the brittle nature of the Type FBD made it very prone to cracking under the thermal stress induced when transitioning from ambient to combustion temperatures. Alternatively, a much softer zirconia ceramic (Type ZYC) was

placed around the Type FBD pieces. The Type ZYC ceramics, while much softer and more prone to erosion under flow, did not exhibit the same thermal cracking behavior. This allowed a layering system to protect the steel chimney from the intense heat of the combustion products.

For low temperature testing, the transition stack did not experience temperatures high enough to warrant the use of ceramics. To prevent flow-induced wear and erosion of the costly ceramic transition stack, a new transition stack was designed and machined out of aluminum. As shown in Figure 3-7, the aluminum transition stack was comprised of 3 separate parts. The upper accomplished the transition from a circular to rectangular cross-section, the lower helped provide the requisite height for the stack, and the insert could either accommodate a ceramic flow straightener (top of Figure 3-7) or provide a different, more effective flow straightener design (bottom of Figure 3-7).

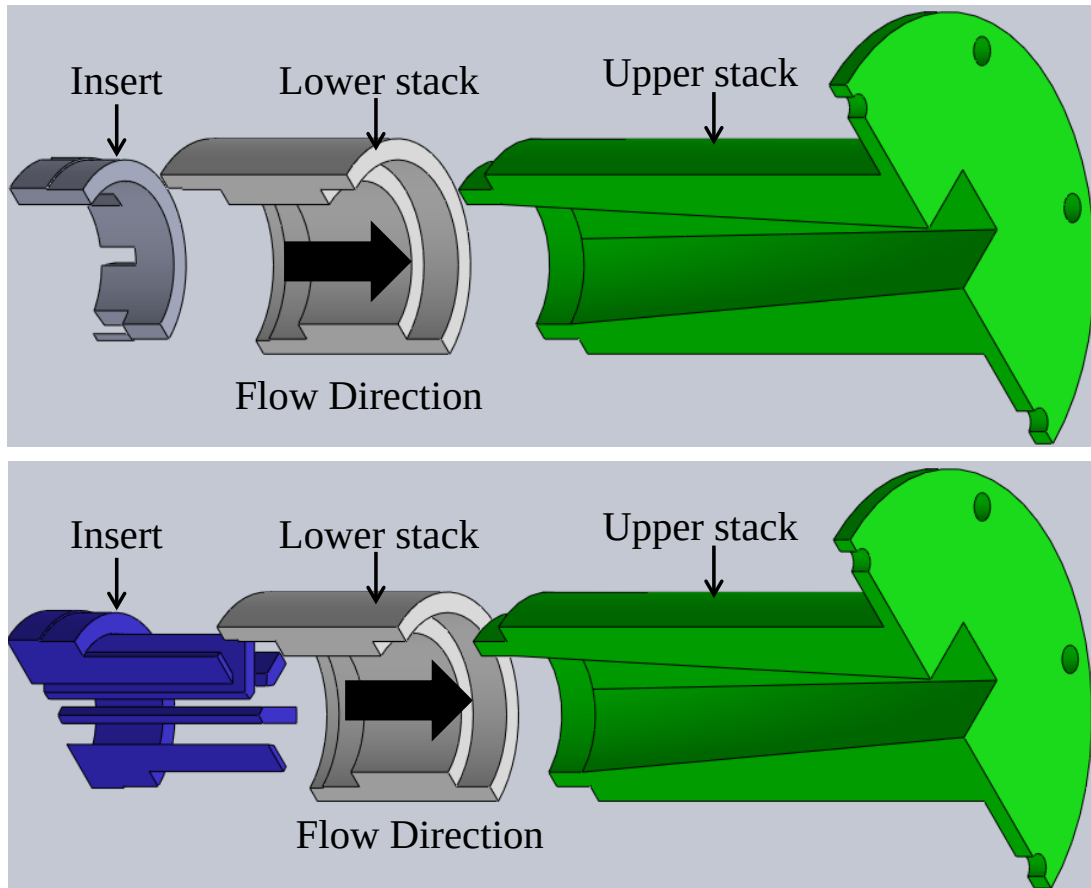


Figure 3-7: Exploded view of aluminum transition stack: ring to accommodate ceramic flow straightener (top) and new flow straightener (bottom)

The new flow straightener was designed based on one of several flow straightening designs tested by Bohan [51]. Figure 3-8 shows both Bohan’s “Design 5” flow straightener machined out of ceramic discs while Figure 3-9 shows a modified form of the design used for the new low temperature flow straightener machined out of aluminum. Machining limits forced Bohan to accept a large radius of curvature where the “arms” of the flow straightener meet the walls. Inserting the new flow straightener into the lower section of the aluminum stack required tight tolerances between the straightener arms and the walls of the lower stack section, resulting in undesirable sharp edges with the same radius of curvature. Aluminum offered much greater flexibility of fabrication and permitted reduction of this radius from 0.125” to 0.005”. Finally,

the flow straightener was designed to occupy the entire 3.0" length of the lower transition stack to minimize swirl velocity in the freestream flow.

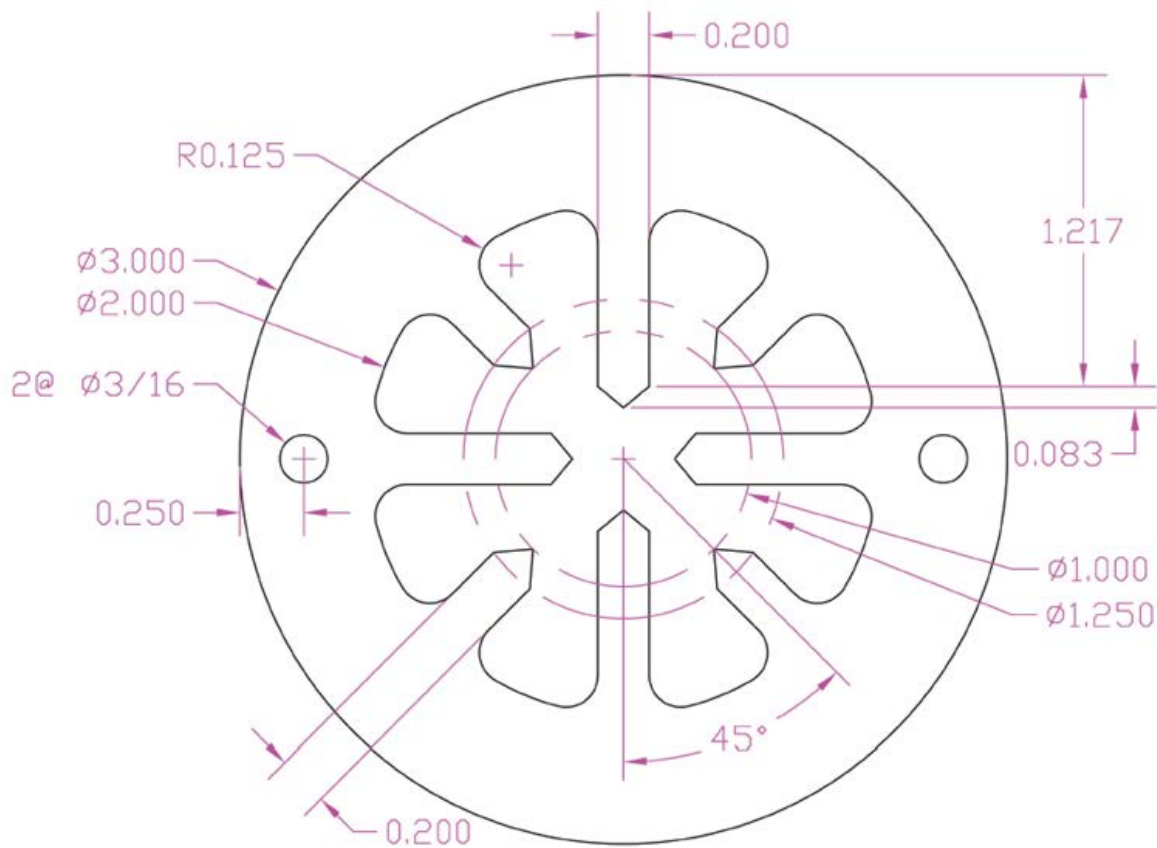


Figure 3-8: Design 5 flow straightener (image from Bohan [51])



Figure 3-9: Low temperature flow straightener

Further features of the WSR are primarily related to thermal management and instrumentation. Figure 3-10a shows the interior of the lower toroid and the jet ring. Four ports were located in the lower toroid, two of which were used. The first allowed injection of an air-ethylene ignition flame and the second allowed access for core reactor temperature measurements with a B-type thermocouple. Mica gaskets, like the one shown in Figure 3-10a, were placed between the jet ring and both toroid halves to ensure a good seal between the separate WSR parts. To protect the Inconel 625 body of the WSR from the extreme combustion temperatures, a Thermal Barrier Coating (TBC) of yttria-stabilized zirconia ceramic was applied to all surfaces of the toroid halves exposed to combustion products. Coolant water loops, shown in Figure 3-10b, were machined into the toroid halves to provide cooling for the WSR. The supporting coolant supply and return lines are also highlighted.

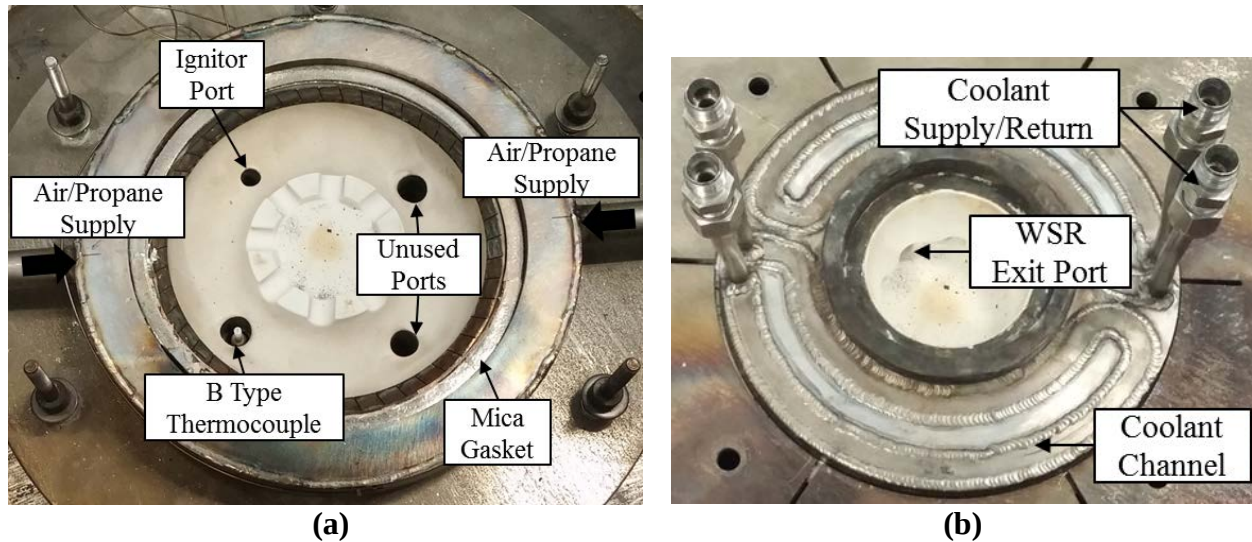


Figure 3-10: AFIT Well-Stirred Reactor: (a) lower toroid and jet ring and (b) upper toroid

3.1.2. Cooling Block Assembly

The cooling block assembly, shown in Figure 3-11 was designed to accommodate a variety of test plate shapes and film cooling schemes. It consisted of two separate pieces that

were inserted into the FCR body: the boundary layer bleed slot and the test block assembly. Both parts were made of a nickel-based superalloy under the proprietary name Hastelloy X. A large rectangular slot in the middle of the FCR main block accommodated the cooling block assembly with very tight tolerance to prevent flow leakage around the parts. Thermocouples were also placed in the cooling block assembly to allow temperature measurements of the coolant gas and the test plate surface and backside wall. The locations and use of these thermocouples will be discussed along with other relevant test plate details in Section 3.1.3.

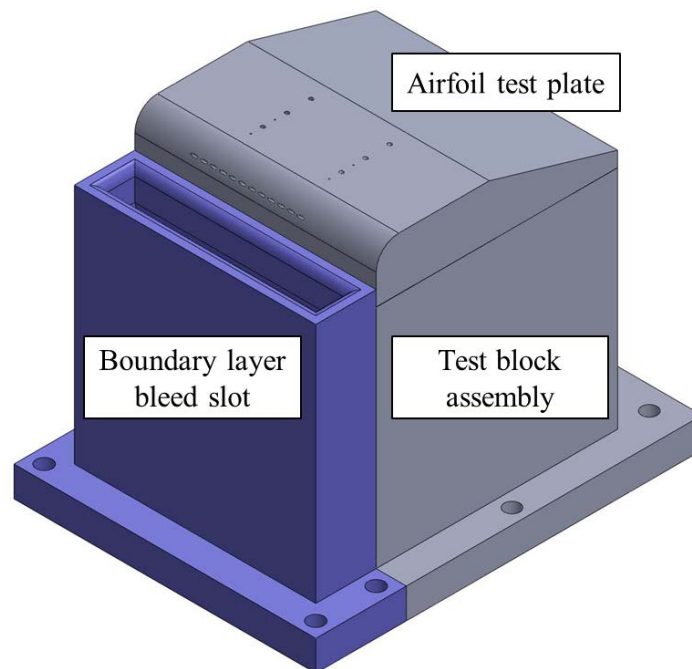


Figure 3-11: Cooling block assembly

The boundary layer bleed slot served to remove the relatively slow-moving boundary layer flow approaching the airfoil test plate prior to striking the test plate. A vacuum pump in series with a 200 SLPM MKS mass flow controller provided the ability to remove flow and control the amount of flow removed. The boundary layer flow removal should allow control over the stagnation point location on the leading edge of the airfoil test plate. The primary goal of removing boundary layer flow was to ensure that the stagnation point aligned with the base of

the quarter cylinder on the airfoil test plate. However, a combination of limited pressure differential and head losses within tubing limited the bleed flow rate to approximately 25 SLPM and raised doubts regarding the magnitude of location shift for the stagnation point. No successful verification of this stagnation point location has yet been performed, so an investigation of the impact of boundary layer flow removal on stagnation point location is well warranted.

As previously stated, the FCR was designed to accommodate a variety of film cooling schemes with holes fed by channel flow. To allow channel-fed film cooling schemes, the test block assembly featured coolant inlet and exhaust ports, a cooling insert to create the internal coolant channel, a passage for thermocouple leads, the airfoil test plate, and a main cooling block to house all the necessary parts. The overall test block assembly is shown in Figure 3-12 without thermocouples present.

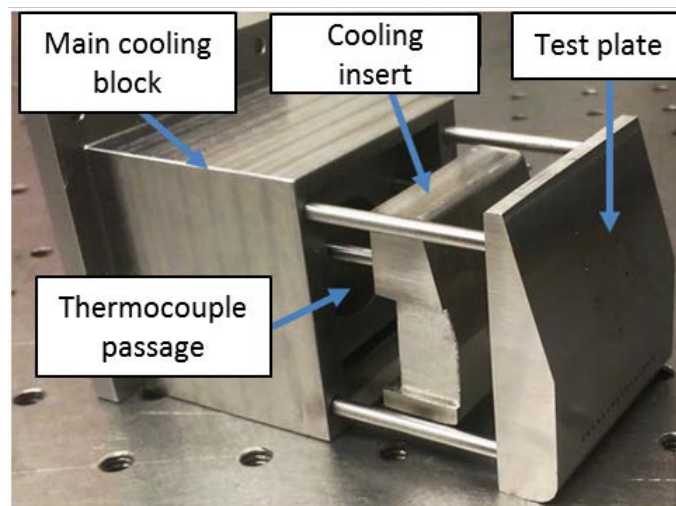


Figure 3-12: Test block assembly (image from Lynch [48])

The cooling insert shown in Figure 3-12 was placed between the airfoil test plate and the cooling block to form the internal cooling channel. This scheme was selected to better represent the channel-fed cooling present in real gas turbine airfoils. Coolant was fed into the test block

assembly through a 63.5 mm ($\frac{1}{4}$ inch) diameter tube and extracted through another tube of identical diameter as shown in Figure 3-13 along with the boundary layer bleed flow path. A vacuum pump provided suction to ensure the required coolant exhaust flow rate was achieved. Two MKS Alta 1480A mass flow controllers were used to control the coolant inlet and exhaust flow rates, thus allowing the film coolant flow rate to be controlled as the difference between the inlet and exhaust flow rates. Recesses cut along the sides and bottom of the cooling insert allowed thermocouples to pass around the insert and into the passage in the main cooling block.

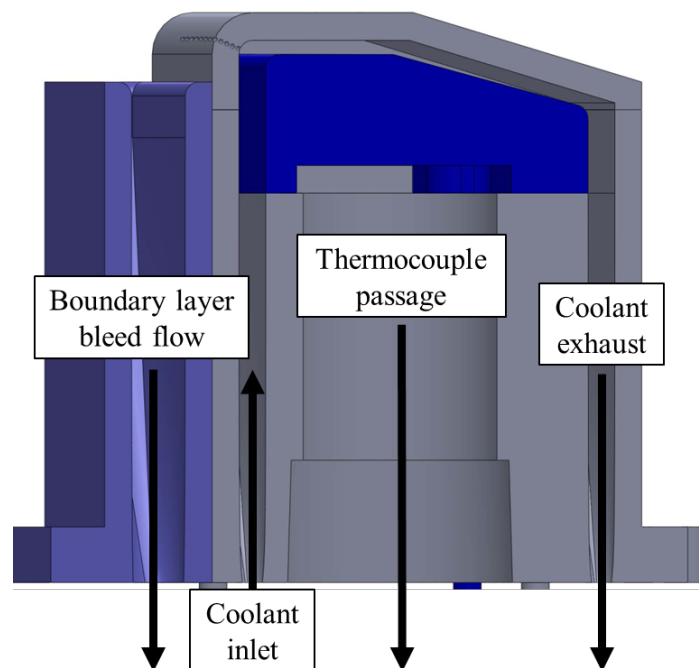


Figure 3-13: Cooling block assembly cross-section with flow paths

Coolant flow rates were calculated based on the parameters to be matched for a given test. For instance, if a freestream Reynolds number and blowing ratio were specified, the freestream velocity and ratio of coolant to freestream mass flux would be known. Using the fixed areas of the test section and coolant holes, the freestream mass flow and coolant mass flow could then be calculated. Excess coolant was required to create a true channel-fed cooling scheme. Coolant equivalent to the flow needed to supply two rows of cooling holes was

supplied in addition to the flow required for the existing holes to ensure an adequate crossflow existed in the internal cooling channel. For example, the one-row showerhead test plate had eight holes, so coolant was supplied to meet the film flow rate demands as well as flow sufficient to feed an additional two rows of holes (16 holes) that was extracted through the coolant exhaust.

For high temperature tests, both the boundary layer bleed and coolant exhaust flows reached temperatures exceeding the safe operating limits of the mass flow controllers used. The boundary layer bleed directly removed freestream flow at temperatures up to 1500K while the coolant flow was observed to reach approximately 900K. To reduce the gas temperatures prior to the mass flow controllers, Lynch [48] designed and implemented a heat exchanger for each flow line. Coolant exhaust flow was directed through a copper tube wound in several loops with approximately 1 m of length immersed in a bucket of water. The boundary layer bleed traveled through a 1.2 m concentric tube, counter-flow heat exchanger with cooling fluid supplied from the cooling water lines for the WSR toroid halves. The heat exchanger was originally fed by the upper toroid cooling line, but it was determined that the lower toroid cooling line absorbed less heat from the WSR and was better suited to cool the boundary layer bleed gas. This decision was made as part of a troubleshooting process related to coolant water overheating that will be discussed in greater detail in Section 3.4.2.

3.1.3. Test Plates

Two separate test plates were used in this investigation. Both used identical test plate body geometry with different cooling hole and instrumentation schemes. The profile of this body shape is shown in Figure 3-14 with all dimensions in millimeters. The shape incorporates a quarter cylinder at the leading edge followed by a flat section parallel to the FCR front block and terminates in a sloped afterbody returning the FCR channel to the same width as that upstream of

the test plate. This shape was selected to represent the leading edge of a turbine airfoil and closely mimics the model shape used by Ekkad et al. [28] without placing the model in the center of the test section. Accordingly, the leading edge diameter of $D = 9.50\text{mm}$ (0.374 inches) is one-tenth the size used by Ekkad et al. [28]. The thickness of the test plate was varied with axial position. The upstream portion containing the film cooling holes was made to provide film cooling holes with lengths representative of real turbine airfoils, while the downstream portion was made thicker to help reduce uncertainty in conductive heat flux calculations.

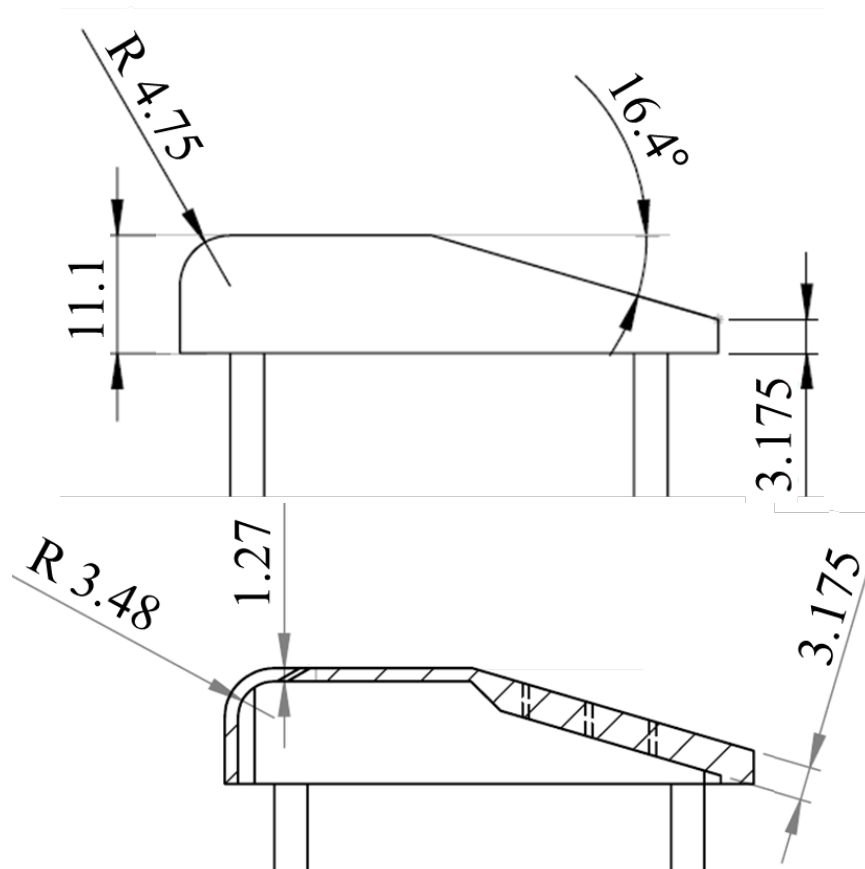


Figure 3-14: Side views of representative turbine airfoil leading edge test plate

The models used for this investigation differed based on the supported objective. Tests examining the effect of physical model size relied on a test plate with a single row of showerhead cooling holes located along the leading edge quarter cylinder at an angle of $\gamma = 21.5$

degrees with respect to the beginning of the quarter cylinder. Holes of $d = 0.51\text{mm}$ (0.02 inches) oriented at $\alpha = 20$ degrees with respect to the surface and a 90 degree compound angle. The ratio of hole length to diameter was $L/d = 7.31$. This geometry is shown in Figure 3-15 and matched the single hole used by Wiese et al. [39] on a model shaped identical to that used by Ekkad et al. [28].

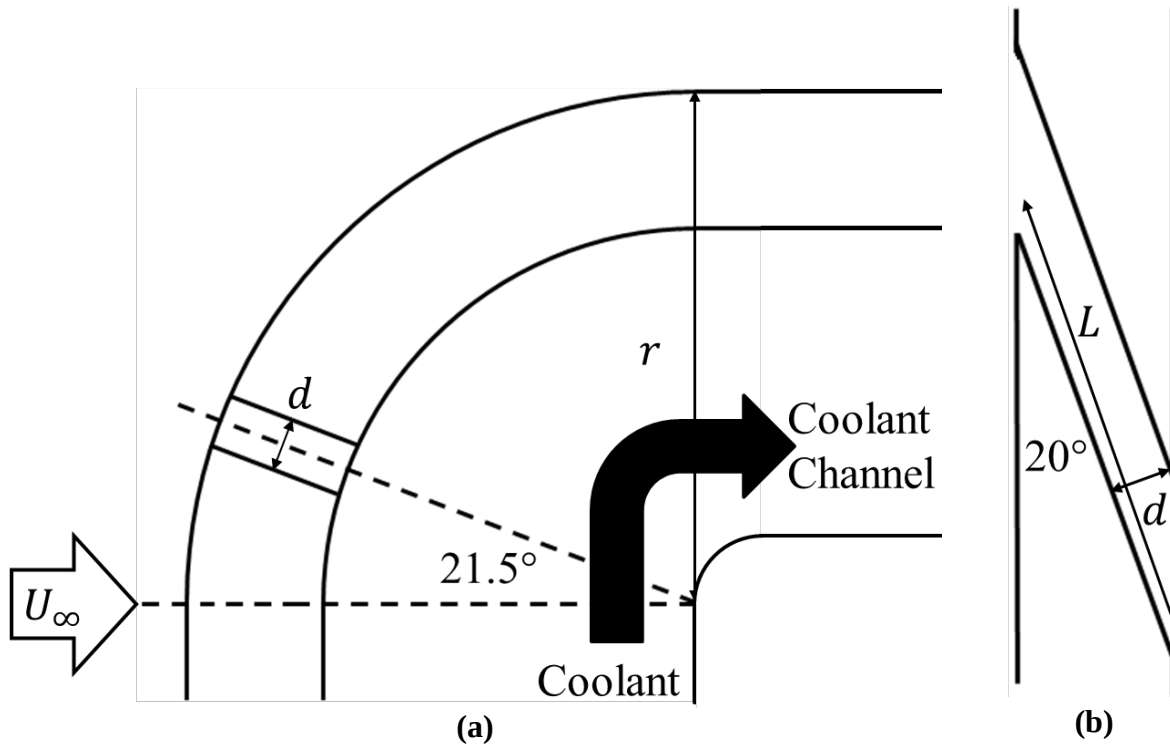


Figure 3-15: FCR one-row showerhead model schematics: (a) side view and (b) coolant hole top view

For the thermal scaling investigations, a cooling hole identical to that examined by Greiner [43,44] was used. This shaped hole was angled at $\alpha = 30$ degrees to the surface and incorporated both layback and fan-shaping diffusion with equal layback and flare angles of $\delta = \beta = 10$ degrees. Sixteen streamwise oriented holes of diameter $d = 0.508 \text{ mm}$ (0.020 inches), spacing $P/d = 4$, and length $L/d = 5$ were placed along the flat section of the test plate and distributed evenly about the test plate centerline. Detailed schematics of the individual hole and

overall cooling scheme geometries are shown in Figure 3-16. Figure 3-16a shows both a top and side view of the cooling hole.

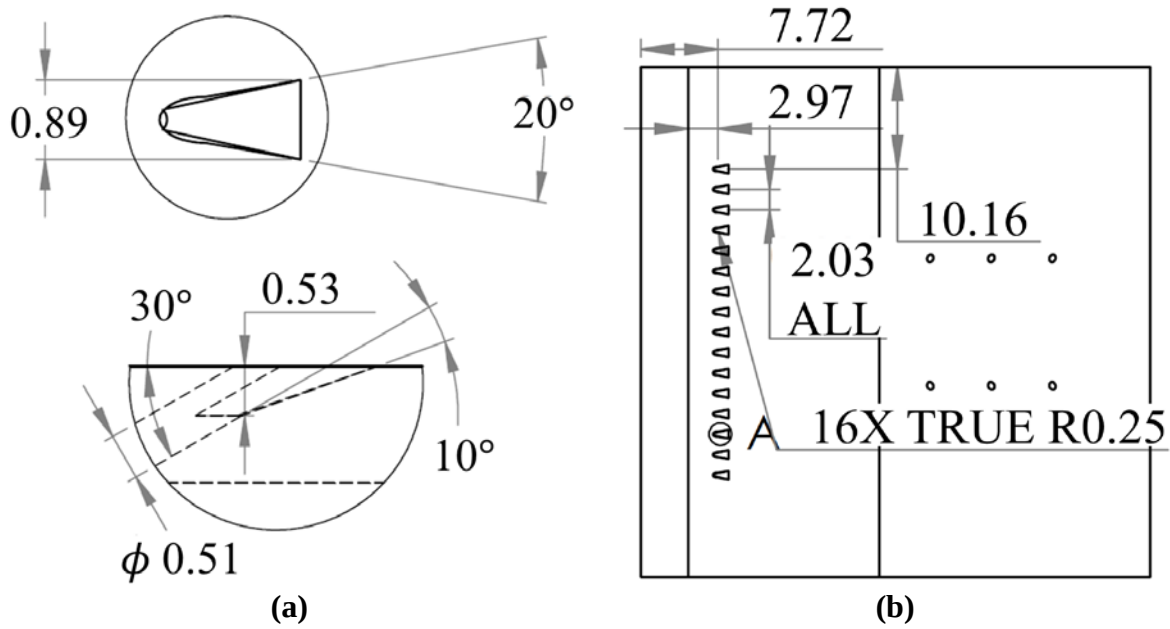


Figure 3-16: Schematics of (a) shaped hole and (b) shaped hole cooling scheme

The test plates were outfitted with thermocouples in several locations. Thermocouples were laser welded to the test plate surface and placed along the backside of the test plate and in the internal coolant channel. Surface thermocouples enabled calibration of the IR camera while backside thermocouples placed directly below their surface counterparts facilitated calculation of conductive heat flux through the test plate, q''_{cond} . The internal channel thermocouples provided coolant temperature measurements necessary to calculate ϕ and h_i .

Two different thermocouple placement schemes were used. The first scheme, shown in Figure 3-17, was developed by Lynch [48] and was intended for use with as many as 5 rows of cooling holes. To accommodate all these rows, surface thermocouples were placed well downstream of the first row of cooling holes. However, thermal tests for the current investigation only used a single row of cooling holes. The size of the IR window prevented

simultaneous imaging of thermocouples on the sloped afterbody and the single row of cooling holes, leaving only one thermocouple in the image. Further difficulties arising from spatially variable surface emissivity and window transmissivity demanded a greater number of temperature measurements in the near-hole region.

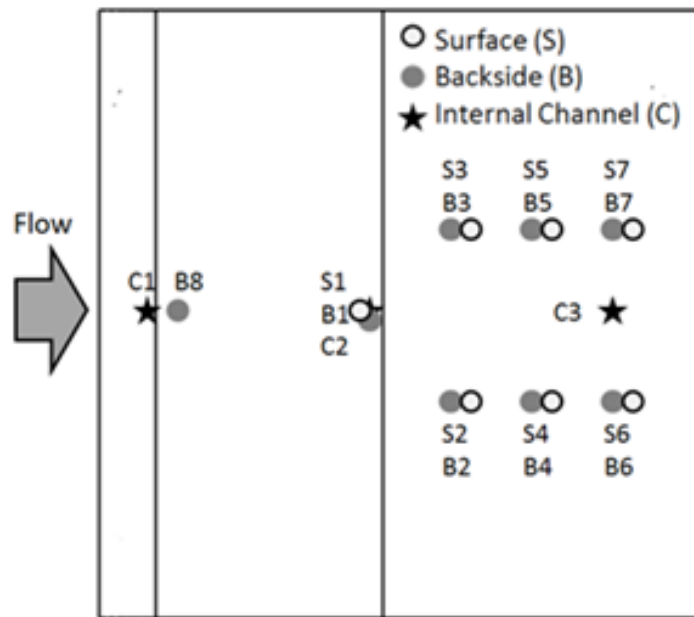


Figure 3-17: Lynch's downstream instrumentation scheme (image from Lynch [48])

The new instrumentation scheme, shown in Figure 3-18, was implemented to provide an adequate number of temperature measurements for IR camera calibration in the near-hole region. Pairs of thermocouples were placed at $x/d = 5$, 10, and 20 with each pair located along the centerline of a cooling hole. The pair at $x/d = 5$ aligned with the fourth hole outboard from the plate centerline, and the pairs at $x/d = 10$ and 20 were aligned with the third hole outboard from the plate centerline. Issues with cooling flow uniformity, addressed further in Section 3.4.1, demanded sealing film cooling holes located well away from the plate centerline. Only the first three holes outboard of the plate centerline were left open, so the thermocouples at $x/d = 5$ did not disrupt the coolant flow.

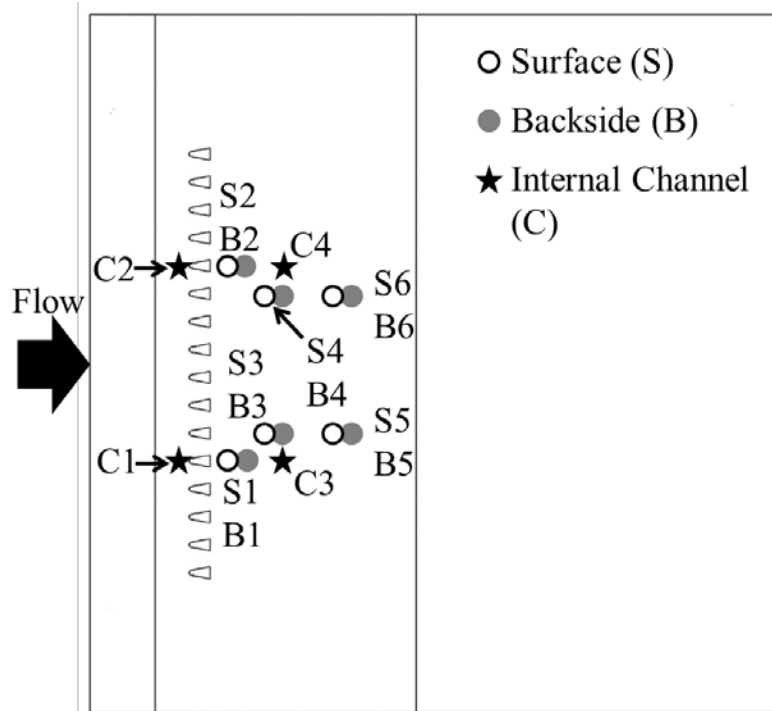


Figure 3-18: Near-hole instrumentation scheme

These coolant uniformity issues also gave rise to a new arrangement for the internal thermocouples. Instead of a single measurement at a given location, two thermocouples were placed at the same axial position and spread evenly across the span of the test plate ($\pm 18d = \pm 9.1$ mm, 0.36 in outboard of the test plate centerline) to check for temperature uniformity in the coolant feed flow. The internal channel thermocouples were bent 90 degrees to place the tip approximately halfway into the space between the cooling insert and the test plate (~ 2.54 mm, 0.1 in). The placement of all these thermocouples is shown in the cross-section of Figure 3-19.

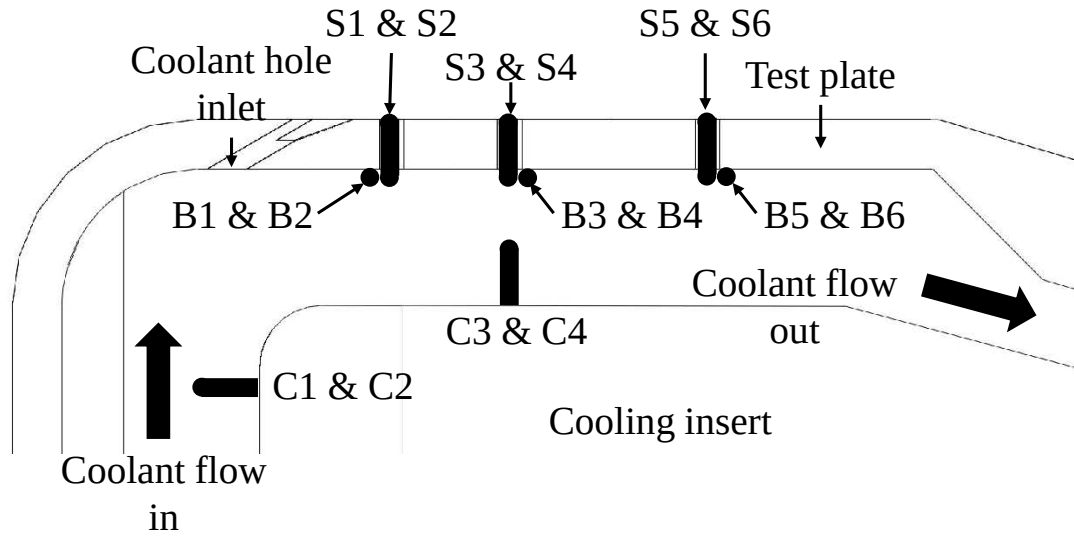


Figure 3-19: Cross-section of near-hole instrumentation

3.2. Additional Equipment

Several additional pieces of equipment were required for the testing process. While equipment already present in the COAL Lab was used to the maximum extent possible, several new pieces of equipment were either acquired or manufactured to support the experiments conducted. Additionally, several cameras were used over the course of testing.

3.2.1. COAL Lab Equipment

Existing resources within the COAL Lab provided several key capabilities for testing. Freestream flow could be supplied by either the two Kaeser air compressors common to AFIT labs or the Ingersoll-Rand compressor dedicated for COAL Lab use. Because relatively low flow rates were required, the AFIT Kaeser compressors were almost exclusively used. The same air source was used to supply air for film cooling while compressed gas cylinders located in the tank farm provided anaerobic gases for film cooling. The coolant exhaust and boundary layer bleed flows were removed from the FCR using a Welch WOB-L 2585 vacuum pump capable of a maximum free air displacement of 201 Liters per minute [52]. After exiting the FCR, these gas

flows were removed through a fume hood and exhausted outside of the building. Because the fume exhaust system served other test rigs, a system of dampers and other flow restrictors was used to maximize exhaust suction through the fume hood above the FCR.

The tank farm housing the anaerobic gas cylinders also contained four 120 gallon compressed liquid propane tanks. These tanks provided the propane flow necessary to fuel the WSR for high temperature tests. Two Zimmer propane vaporizers heated the liquid propane to ensure gaseous propane delivery to the WSR. Cylinders of anaerobic gases used for film coolant included carbon dioxide, argon, nitrogen, and helium. A cylinder of ethylene in the tank farm provided fuel for an ethylene-air spark ignitor used to initiate propane combustion within the WSR.

A Gaumer Process 12.5 kW electric heater provided the capacity to heat the freestream flow to temperatures in excess of 580 K (600 °F). This heater was used to preheat combustion reactants for high temperature testing and heat the freestream flow for near ambient tests. Coolant flow was heated using a 1200 W OMEGALUX AHPF-121 electric heater, providing gas temperatures as high as 600 K. A 400 W heater of the same type was available to provide a maximum of 1600 W of heating power, but it was not used for any of the experiments in this study.

Mass flow controllers provided control over all relevant gas flows. A Fox Thermal Instruments, Inc. FT2 flow meter provided flow rate information for freestream flow control. The freestream flow rate was physically controlled by a Eurotherm 2404 process controller in conjunction with a Badger Meter, Inc. Type 807 valve and Type 755 pneumatic actuator. Propane flow was controlled by a Brooks 5853i mass flow controller capable of propane flow rates as high as 200 SLPM. Various MKS mass flow controllers were used to control gas flows

for film coolant, the boundary layer bleed, and the spark ignitor. The MKS controllers, including MKS MC20 and Alta 1480 models, were managed using an MKS Model 647C multi gas controller. Section 3.3 provides details on the application of these flow controllers and their maximum flow rates.

The MKS flow controllers were calibrated using a Mesa Labs Bios Definer 220 capable of measuring flow rates between 0.3 and 30 SLPM. Measurements were taken for each controller starting at 5% of the controller's flow capacity and increasing by 10% up to 95%. A sweep of decreasing flow rate was then taken starting at 100% of flow capacity decreasing by 10% down to 10%. A trend line was fit to the data with the measured flow rate as a function of the flow rate setpoint. Adjustments were made to correct the slope of the trend line to unity and eliminate any y-intercept and the process was repeated to validate the accuracy of the calibration. Because maximum flow rates for these controllers did not exceed 30 SLPM, controllers with full scale flow rates in excess of 30 SLPM were calibrated using the points for a 30 SLPM controller. A more detailed description of this calibration process is available from Lynch [48].

Temperature measurements were made using thermocouples and an infrared camera. Two B-type thermocouples were used to measure combustion gas temperatures while K-type thermocouples were used for all other applications. Pressure measurements were obtained using a 32 channel pressure scanner (ESP-32HD) manufactured by Pressure Systems, Inc. The pressure scanner consisted of 32 separate transducers all located within a single housing, enabling simultaneous measurement of all 32 channels [53]. An accompanying DTC Initium controller also manufactured by Pressure Systems, Inc. provided data acquisition capability for the pressure scanner and served to correct thermal variations in the pressure scanner without an extensive calibration process [54].

Data collection was accomplished using software developed by Lynch [48]. Two separate LabView programs were used. The first program controlled the freestream flow rate and an array of solenoids used to close flow lines. This program was also used to collect temperature and flow rate data. The second LabView program interfaced with the Initium controller and was used to re-zero the pressure scanner, control pressure data acquisition parameters, and acquire pressure data.

3.2.2. New Equipment

While the existing COAL Lab equipment was sufficient for high temperature testing, several new pieces of equipment were acquired or manufactured to facilitate near ambient tests. A Cole-Parmer 1C6 cooling/heating circulating bath was acquired to expand the attainable range of coolant temperature. The bath was capable of cooling fluid to 253 K or heating to 373 K. A concentric tube counter-flow heat exchanger approximately 2 m long was constructed to cool or heat coolant gas before entering the FCR. The necessary lengths of tubing were routed from the mass flow controllers to the heat exchanger and from the heat exchanger to the FCR coolant inlet.

Several pieces of equipment were needed strictly for PSP testing. The new aluminum flow straightener and transition section described in Section 3.1.1 and a new optical window for the test section were designed and manufactured. The new optical window was composed of fused silica (quartz) for visible light transparency and designed to fit into the mount for the existing sapphire window plate. Other pieces of equipment, such as an excitation illumination source, PSP image acquisition and processing software, a CCD camera, and supporting optics and filters for the camera were all supplied by Innovative Scientific Solutions, Inc. (ISSI). Further description of these items can be found in the PSP setup of Section 3.3.2.

3.3. Test Setups

The objectives of this investigation demanded two different types of testing. Experiments exploring the importance of physical model size were solely conducted using the PSP measurement technique, and those investigating temperature dependent scaling effects used the IR technique. While the test setups for these two different techniques shared many common features, several items were unique to each setup. The primary differences between the two setups will be addressed in the following discussion.

3.3.1. IR Thermography Setup

The thermal test setup will be described in terms of instrumentation, any deviations from the FCR description of Section 3.1, and a discussion of the calibration used for the IR camera. Thermal tests consisted of both high temperature combustion and comparatively low temperature non-combustion conditions. The instrumentation for these two testing conditions varied slightly, and those differences are highlighted in Figure 3-20. The inlet thermocouples were varied between the two temperature regimes based on material and instrumentation limits. The B-Type thermocouple for high temperature testing provided a smaller uncertainty, but B-Type thermocouples fail to accurately measure temperatures below 400 K. Additionally, while the K-Type thermocouple provides accurate measurements for near ambient conditions ($T_{\infty} < 400$ K), it cannot withstand intense combustion temperatures for an extended period of time. Thus, the B-Type thermocouple was used for high temperature conditions and the K-Type thermocouple was used for near ambient conditions.

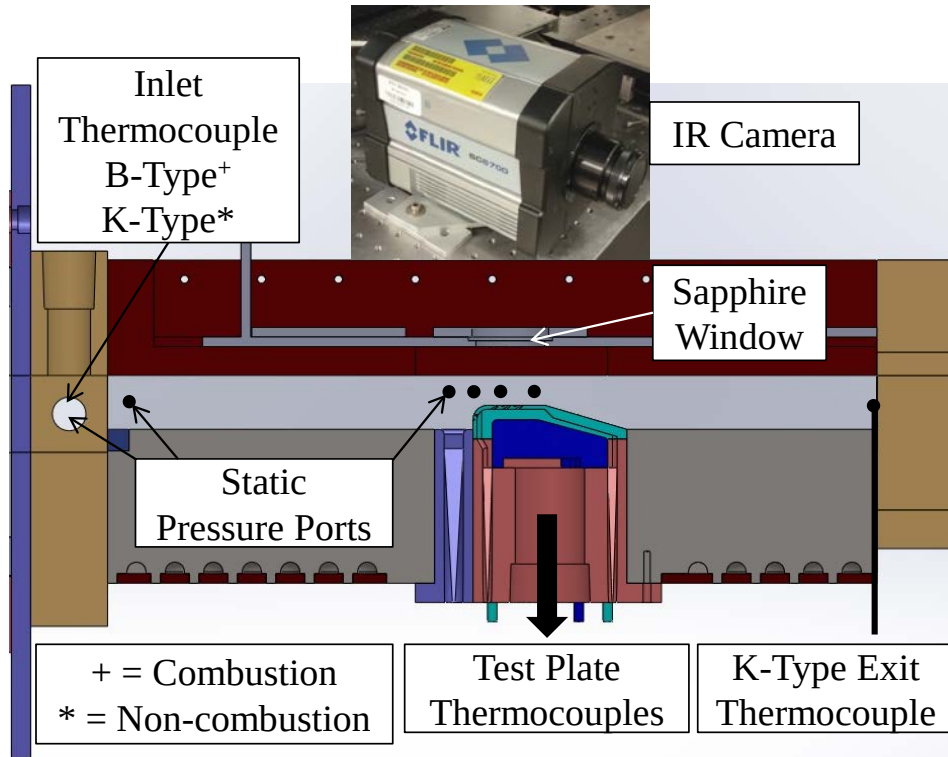


Figure 3-20: IR thermography test setup

Other differences between the two experimental regimes not shown in Figure 3-20 dealt with mass flow and temperature control. Similar model mass flow controllers were used to control the various coolant mass flows, but the ranges of the controllers were adjusted to minimize uncertainty in the coolant flow rates. All mass flow controllers were calibrated prior to use with a Mesa Labs Definer 220 (up to 30 SLPM) or DryCal 1020 (up to 500 SLPM) and air as the calibration gas. The Cole-Parmer circulating bath and concentric tube heat exchanger discussed in Section 3.2.2 were used to reduce coolant gas temperature for the near ambient conditions while the Omegalux heaters described in Section 3.2.1 raised coolant gas temperature for combusting conditions. The 48 kW Gaumer Process heater was used to elevate freestream temperatures for near ambient conditions and preheat freestream flow for combustion in the WSR for combusting conditions.

3.3.1.1. IR Camera Calibration

All thermal tests were accomplished using a FLIR SC-6700 infrared camera. The IR camera was used to determine test plate surface temperatures at a high spatial resolution. This camera required calibration both spatially and to relate measured photons to test plate surface temperature. A description of both calibrations follows.

The spatial calibration assessed a value of pixels per inch by measuring the number of pixels between two points and dividing by the known distance between points. Positions were assigned relative to the film cooling holes and then normalized by cooling hole diameter. While this method cannot resolve surface curvature or lens distortion, IR images were exclusively taken on the flat portion of the test plate normal to the IR camera and lens distortion was assumed to be negligible.

A relationship between test plate surface temperature and emitted radiation intensity was required for IR testing. To accomplish this, surface temperatures measured via thermocouples and corresponding IR images were captured over a range of temperatures spanning the expected experimental conditions. Because radiative heat transfer exhibits a fourth-order behavior, as shown in Eq. (2-26), the relationship between temperature and intensity was initially characterized using the following form

$$T = aJ_{rad}^{1/4} + b \quad (3-1)$$

where J_{rad} is a count of photons striking the IR camera sensor and a and b are constants assigned by a curve fit. This process was repeated for both the near ambient non-combustion and high temperature combustion conditions used in this study. No film cooling gas was used during IR calibration in either condition. A more detailed description of the calibration testing procedures and IR camera settings follows.

Radiative emission from the test plate was recorded by the IR camera as a count of photons. This count of photons was related to surface temperature as shown in Eq. (3-1). An average over several pixels close to the surface thermocouples was desired to minimize uncertainty in the IR calibration. However, the surface thermocouple weld beads were observed to emit differently from the rest of the test plate surface for both combustion and non-combustion conditions (see Figure 3-21). Because the weld beads protruded into the flow slightly (approximately 2.5 mm, 0.1 inch), it is possible that the observed difference in emission was caused by a real temperature difference. Alternatively, the material used to laser weld the thermocouples to the test plate surface was unknown, introducing the possibility of different emissivity on the weld beads. Due to the high thermal conductivity of the test plate, it was assumed that any temperature difference between the surface temperature and the point of measurement at the thermocouple weld bead would be negligible.

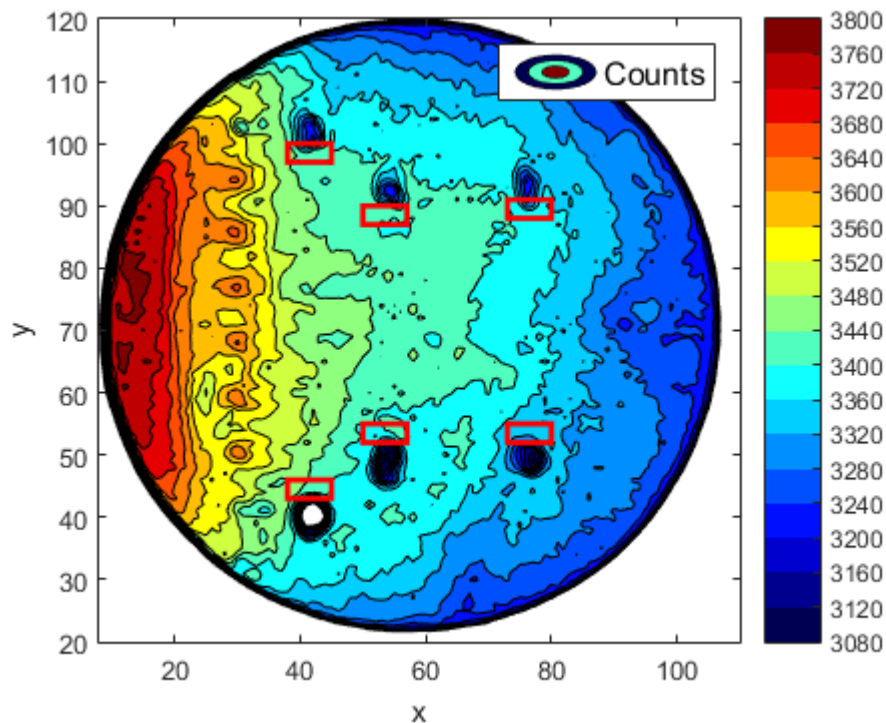


Figure 3-21: IR calibration intensity measurement region

Accordingly, a small array of pixels were selected in the immediate vicinity of each surface thermocouple to provide an average J_{rad} value. This region measured three pixels across the test plate span and seven pixels along the streamwise direction. Figure 3-21 shows a representative high temperature calibration image with red rectangles placed around the regions averaged to generate the J_{rad} values used for calibration. The size of the regions was chosen to best represent the conditions adjacent to each thermocouple contributing to the registered temperature measurement.

Tests conducted at lower temperatures without combustion in the WSR presented minimal potential for participating media in the FCR test section, so the IR camera was configured to collect photons of all measurable wavelengths. The camera integration time was set to 0.25 s to allow capture of enough photons for an acceptable signal to noise ratio without oversaturating the camera. A range of surface temperatures spanning slightly beyond expected experimental conditions was achieved by heating the freestream air using the 12 kW Gaumer Process heater. Test points were collected from highest to lowest temperature with no points repeated. Uniform test plate emissivity was achieved by painting the test plate with a flat black paint.

An initial calibration for the non-combustion condition was completed using Eq. (3-1). However, because of the apparent curvature of the calibration data, another calibration was accomplished using a modified form of Eq. (3-1) incorporating a quadratic term. The modified equation is given as Eq. (3-2)

$$T = aJ_{rad}^{1/4} + bJ_{rad}^{1/2} + c \quad (3-2)$$

Where a , b , and c are constants generated by a curve fit. The results of the subsequent calibration are shown in Figure 3-22. Each surface thermocouple is presented individually to

demonstrate that the relationship between surface temperature and emitted radiation is consistent across the test plate surface. Surface thermocouple 1 (TC1 in Figure 3-23) was omitted from this calibration because it provided faulty data for the temperature range covered.

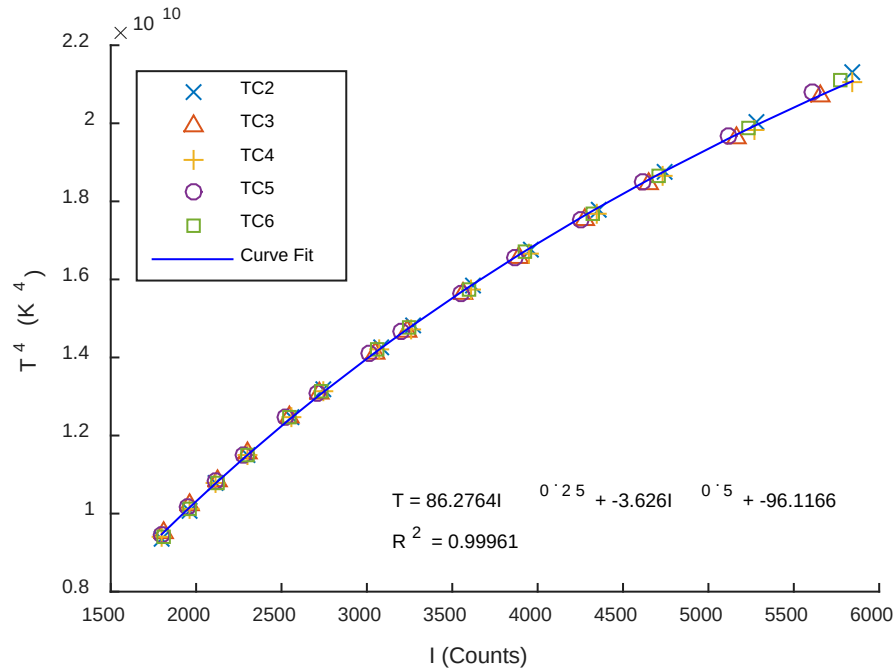


Figure 3-22: Non-combustion condition IR calibration with quadratic term

For combustion conditions, the IR camera was configured to eliminate influence from participating media in the freestream flow, such as radiation from the combustion products or flames. This was accomplished using a 3900 nm bandpass filter. The camera integration time was set to 0.01 s to prevent oversaturation. Freestream temperature was varied by altering both freestream flow rate and equivalence ratio, generating surface temperatures spanning slightly beyond the range of expected experimental conditions. These points were collected ranging from lowest to highest temperature with no repeat of any points. Figure 3-23 shows the initial calibration data and curve, accomplished using the three-term form of Eq. (3-2).

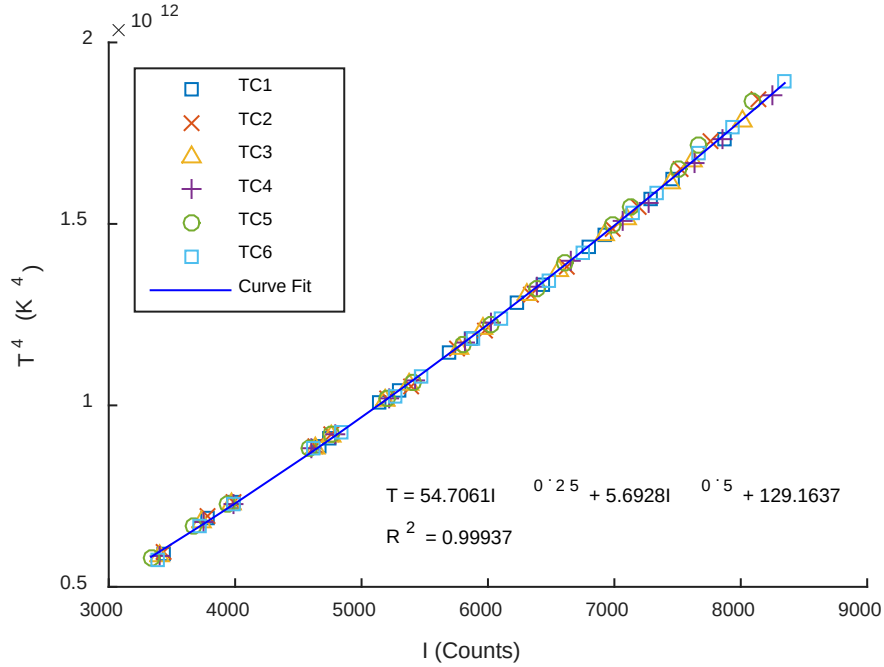


Figure 3-23: Combustion condition IR calibration with quadratic term

Both the calibrations of Figure 3-22 and Figure 3-23 were assessed for accuracy after testing completion. Values of temperature and emitted radiation intensity measured during experiments were plotted with their respective calibration curves, the results of which are presented in Figure 3-24 and Figure 3-25. A new calibration was performed solely using the experimental data, as indicated in the legend as “Experimental Curve Fit.” Curve fit accuracy was assessed by calculating the difference between experimentally measured temperatures and temperatures predicted using the appropriate curve fit and intensity value, as shown in Eq. (3-3)

$$\Delta T = T_{exp} - T_{calc} = T_{exp} - \left(aJ_{rad,exp}^{\frac{1}{4}} + bJ_{rad,exp}^{\frac{1}{2}} + c \right) \quad (3-3)$$

where the subscripts *exp* and *calc* represent an experimentally measured temperature and calculated temperature based on a measured radiation intensity. The root mean square (ΔT_{rms}) and maximum value of this temperature difference (ΔT_{max}) were used to determine the suitability of the curve fit for the experimental data collected.

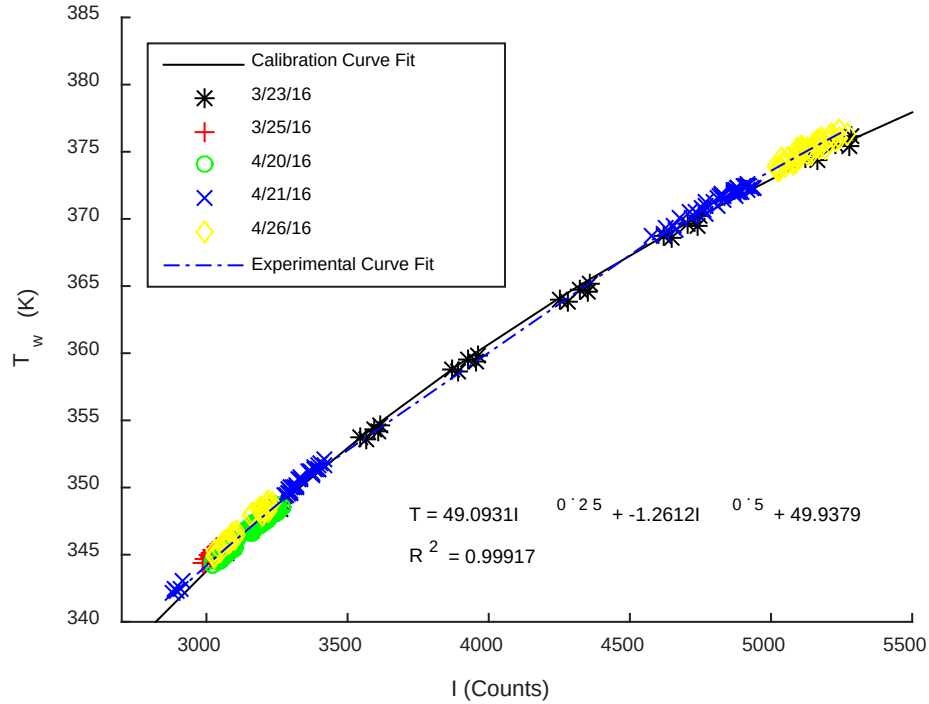


Figure 3-24: Non-combustion condition IR calibration comparison to experimental data

The non-combustion condition experimental curve fit agreed quite well with the original calibration curve fit, as shown in Figure 3-24. Performing a new curve fit with the experimental data reduced ΔT_{rms} from 0.50 K to 0.35 K and ΔT_{max} from 1.26 K to 0.99 K. Uncertainty in the measured temperature values was 0.75% of the measured temperature, yielding an uncertainty of approximately 2.74 K for an assumed average value of 365 K. This improvement provided by the new experimental curve fit lies within instrument uncertainty, so the original calibration curve was used to calculate surface temperatures for all non-combustion test cases.

However, for the lower experimental test plate temperatures, Figure 3-25 displays a systematic deviation from the original calibration curve. The experimental curve fit reduced ΔT_{rms} from 6.70 K to 3.56 K while ΔT_{max} decreased from 23.26 K to 10.26 K. Assuming an average temperature of 1125 K, these ΔT_{max} values represent 2.06% and 0.91% of the measured temperature. Uncertainty in measured temperature for combustion conditions was assumed to be approximately 8.50 K (0.75% of average value of 1125 K), meaning the experimental curve fit

actually provides meaningful improvement over the calibration curve fit. Thus, surface temperatures were calculated using the experimental curve fit for all combustion test cases.

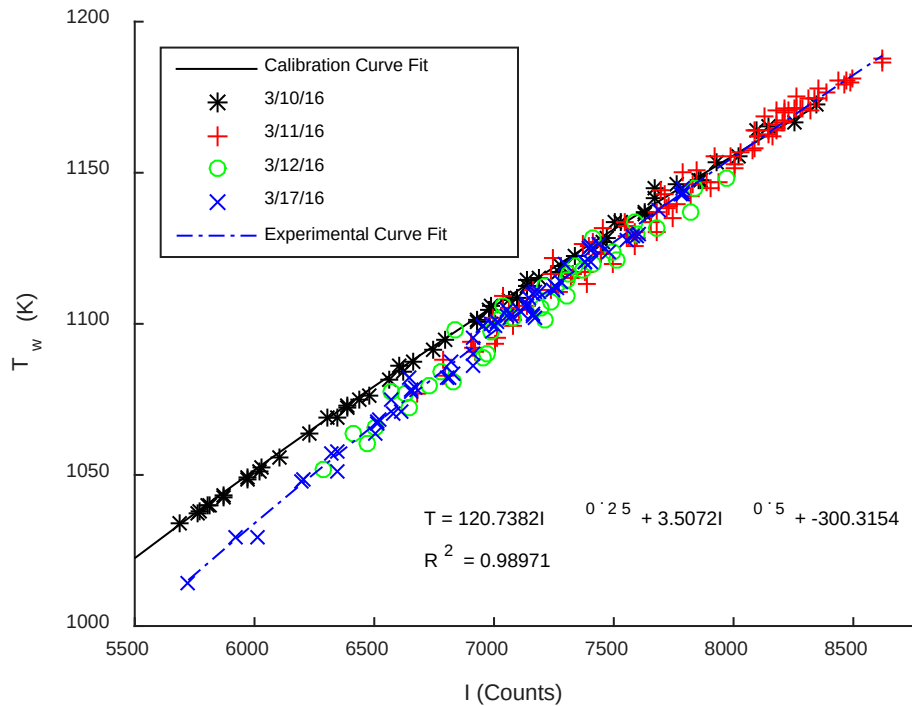


Figure 3-25: Combustion condition IR calibration comparison to experimental data

An increased photon count for the same surface temperature could be caused by several phenomena. First, increasing test plate emissivity would be consistent with the chronological order of the calibration test points, collecting points from low to high temperature in time. The calibration test duration could allow the requisite “burn-in” time to achieve stable surface emissivity by the end of the nearly four hour long test. Further, the initial “burn-in” was attempted using a Hencken burner, providing lower surface temperatures than would typically be encountered in the FCR test section. For this reason, it is recommended that future “burn-in” be accomplished with the test plate in the FCR test section. Additionally, future IR calibrations should repeat some points to ensure temporal stability in test plate surface emissivity. Analysis of previous IR calibrations indicates that stable surface emissivity may be attained with as little

as 4 hours of burn time. Combustion condition calibration data from three separate days shown in Figure 3-26 show excellent agreement between data points collected with cumulative burn time ranging from 4 to 17 hours. Some surface emissivity increase is evident, but this effect is minimal.

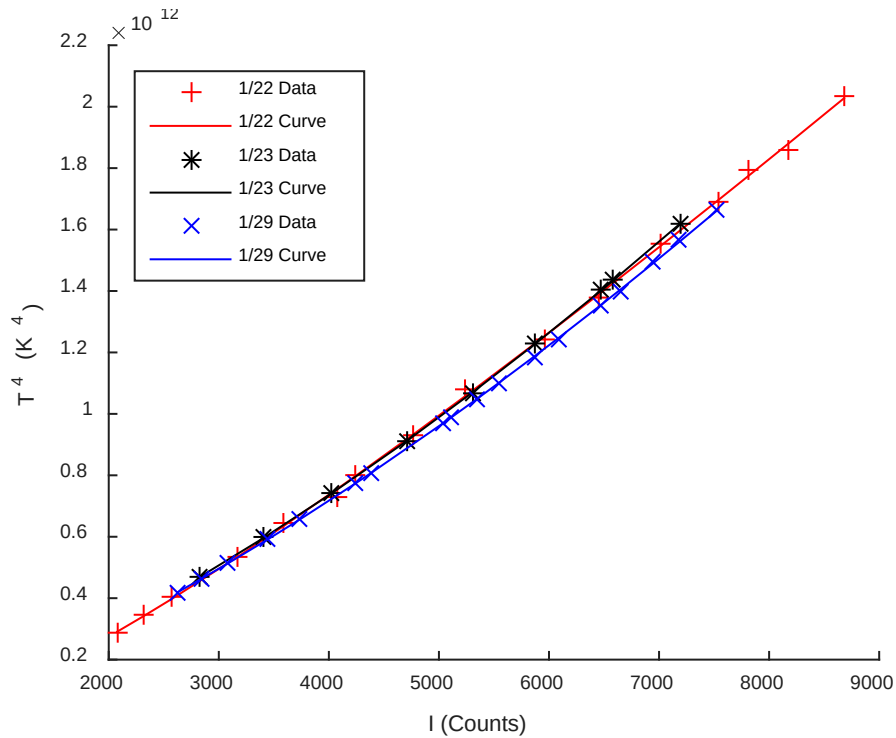


Figure 3-26: Combustion condition IR calibration temporal comparison

Alternatively, this increase in measured radiative transfer from the test plate could be caused by other radiative heat transfer mechanisms. First, the introduction of coolant gas flow for the experimental conditions requires a higher freestream temperature to achieve the same surface temperature. This difference in experimental and calibration freestream temperatures for the lowest surface temperature in Figure 3-25 (1014 K) was approximately 200 K. Other surface temperatures revealed freestream temperature differences ranging from 100-170 K. This same phenomenon also appears in data gathered prior to coolant baffle implementation, shown in Figure 3-27. The increase in freestream temperature for experimental conditions increases the

temperature of surrounding materials, raising irradiation incident upon the test plate. Assuming a constant value for test plate reflectivity, increased irradiation would increase reflected radiation, raising the number of photons striking the IR camera. Additionally, possible oxide and soot buildup on the sapphire window could emit, further increasing the radiation detected by the IR camera. Fortunately, the contributions of these mechanisms can be accounted for using an in situ calibration method.

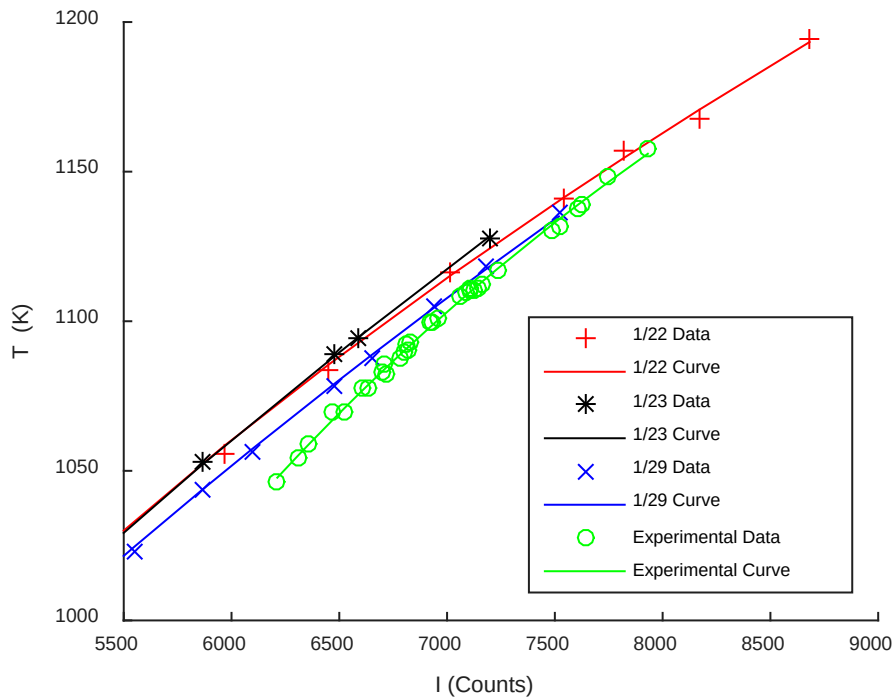


Figure 3-27: Early combustion condition IR calibration validation

3.3.2. PSP Setup

The PSP testing method required different equipment than thermal testing. A schematic of the PSP setup is shown in Figure 3-28 and a picture in Figure 3-29 shows the actual equipment as placed for testing. Approximate incidence angles for the LED light sources are also shown in Figure 3-29. Equipment required strictly for PSP testing included an Imperx Bobcat CCD camera, a 400 nm LED illumination source, and a total pressure probe placed at the

FCR inlet. The total pressure probe provided test section total pressure necessary for pressure field calculation with the PSP method. A piece of screen with circular wire elements 0.305 mm (0.012”) in diameter comprising a square mesh with four diameter spacing (1.22 mm, 0.048”) was also placed at the FCR inlet to break up large turbulent flow structures. The screen was chosen to assure 1 % turbulence at the test plate leading edge assuming no additional turbulent trips between the screen and test plate. A thick black line at the far left of Figure 3-28 schematically represents the location of the screen. This screen was introduced to combat an apparent swirl velocity in the freestream revealed through coolant jet flow direction observations during PSP testing, discussed in Section 3.4.1.

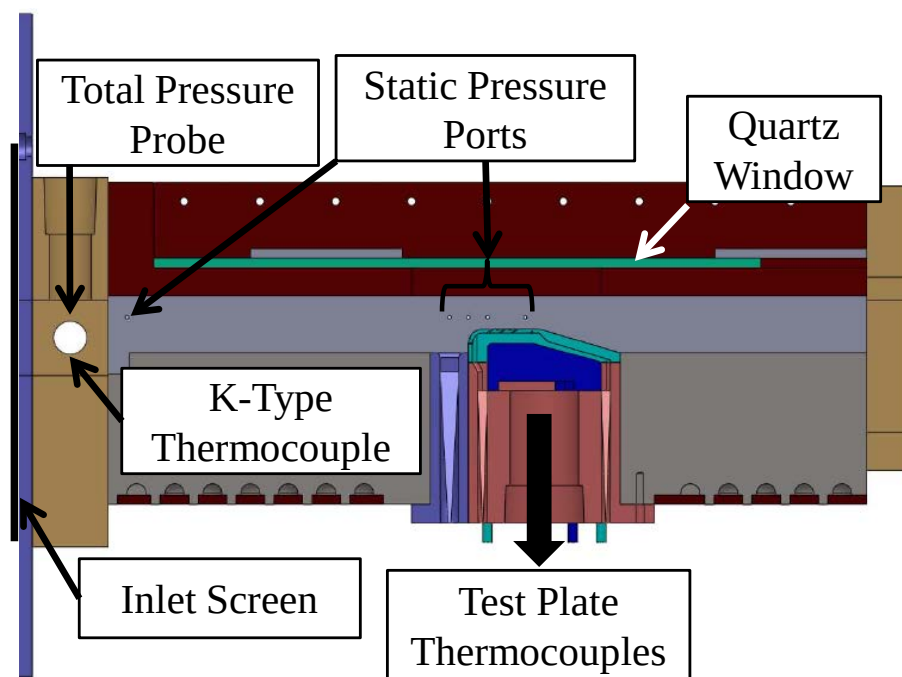


Figure 3-28: PSP test setup instrumentation schematic

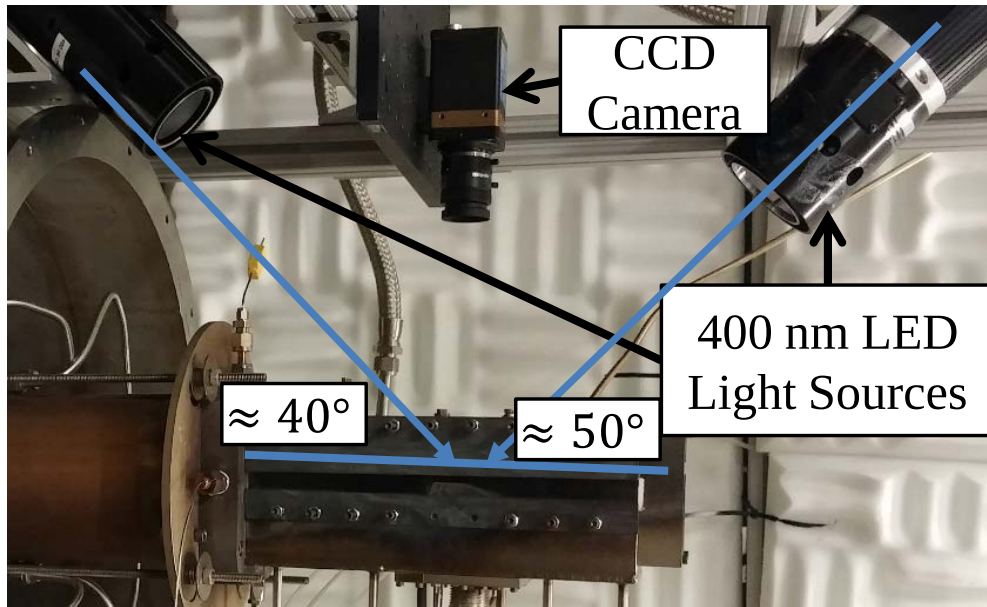


Figure 3-29: PSP test setup

To maximize the test plate image within the camera field of view, a 35 mm lens and spacer rings providing 15 mm standoff between the lens and camera were used. A rectangular quartz window also provided increased test plate optical access over the circular sapphire window used for IR testing. This window also permitted placement of the 400 nm excitation light at an oblique angle to minimize reflected excitation light entering the camera. An optical filter was also placed on the lens to minimize measurement of excitation light and maximize registration of signal light wavelengths.

Further, the small size of the hardware and optical access windows posed a significant challenge to collecting images with sufficient spatial resolution and excitation light intensity. Several excitation light positions were considered prior to the final configuration as approximated in Figure 3-28. Combinations of camera placement and lens focal length were also examined to maximize spatial resolution with the Bobcat camera. All efforts to maximize spatial resolution and excitation light intensity resulted in the oblique light placement, camera

placement normal to the test plate surface, and the combination of a 35 mm focal length lens with a 15 mm standoff between lens and camera.

3.3.2.1. Camera Spatial Calibration

The optical camera used for this study was an Imperx Bobcat CCD camera with a resolution of 804 x 604 pixels. In order to accurately resolve position on the curved test plate leading edge, a three part spatial calibration was accomplished for the Bobcat. First, an image of the test plate was acquired with illumination appropriate for PSP tests and no gas flow (Figure 3-30). Film cooling hole locations were identified based on pixel location. Second, a printed grid was adhered to the test plate surface and another image was acquired under the same conditions as the first image (Figure 3-31). Gridline intersections were identified and their pixel locations recorded. Finally, a spatial map for each pixel's location in Cartesian coordinates was generated using the pixel locations from the grid image. Using the known distances between gridlines, each pixel in the grid image was assigned a position. These positions were offset according to the location of the single row of film cooling holes and then normalized by film cooling hole diameter to provide pixel locations in terms of x/d and y/d .

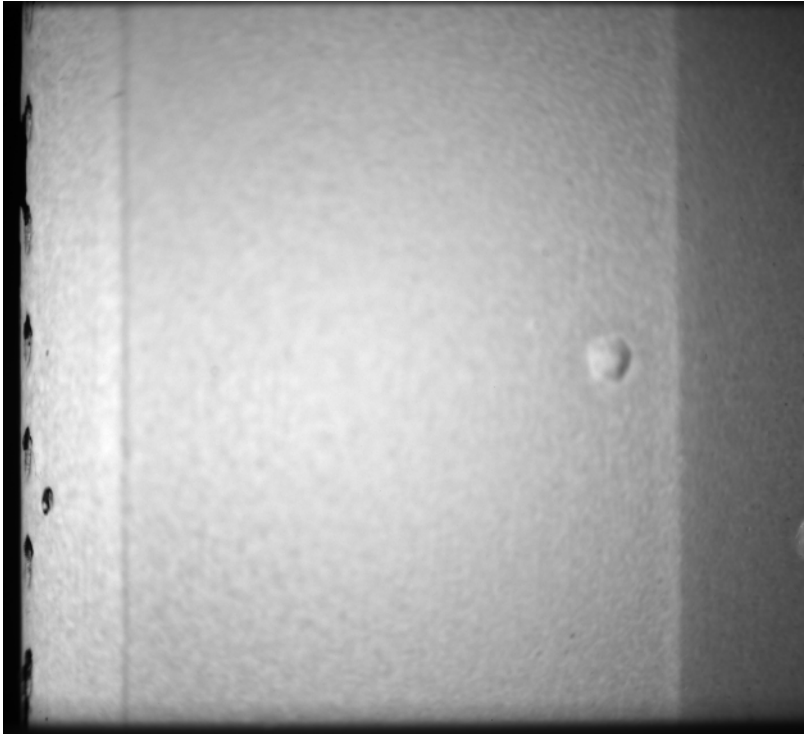


Figure 3-30: Bobcat spatial calibration no grid image

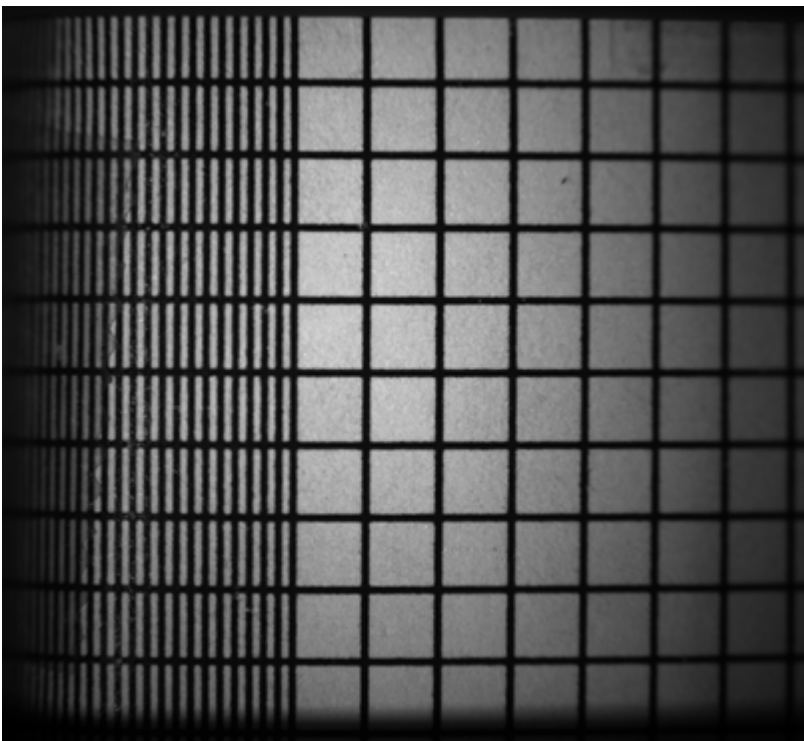


Figure 3-31: Bobcat spatial calibration grid image

3.4. Test Rig Troubleshooting

Several difficulties were encountered with testing equipment over the course of this investigation. Each of these problems required development of a solution, many of which altered the test rig hardware or test procedures. The largest obstacles dealt with coolant flow uniformity, overheating the chilled water supplied to cool the WSR, and flow leaks and unknown material limits that were exposed during high temperature tests. A brief description of these problems and the solution method used for each problem is presented in the following sections.

3.4.1. Coolant Flow Uniformity

While conducting PSP tests with the one row showerhead test plate, a bizarre coolant flow pattern was observed. The raw images of Figure 3-32 show a bias in coolant jet placement towards the test plate centerline. Additionally, the ejected coolant distribution appeared to favor the right side of the test plate as the coolant inlet flow rate increased. Selected film cooling holes were then plugged to reduce variation in the coolant jets ejected from the remaining open holes. Three cooling holes whose inlets were located in the center of the test plate were left open, and the remaining five holes were sealed using a silicone-based bathroom caulk. This solution was sufficient to provide reasonable uniformity in the three remaining coolant jets. It is important to note that the conditions of Figure 3-32 are not matched between the two images. Each image should be considered individually for uniformity of flow across the span of cooling jets.

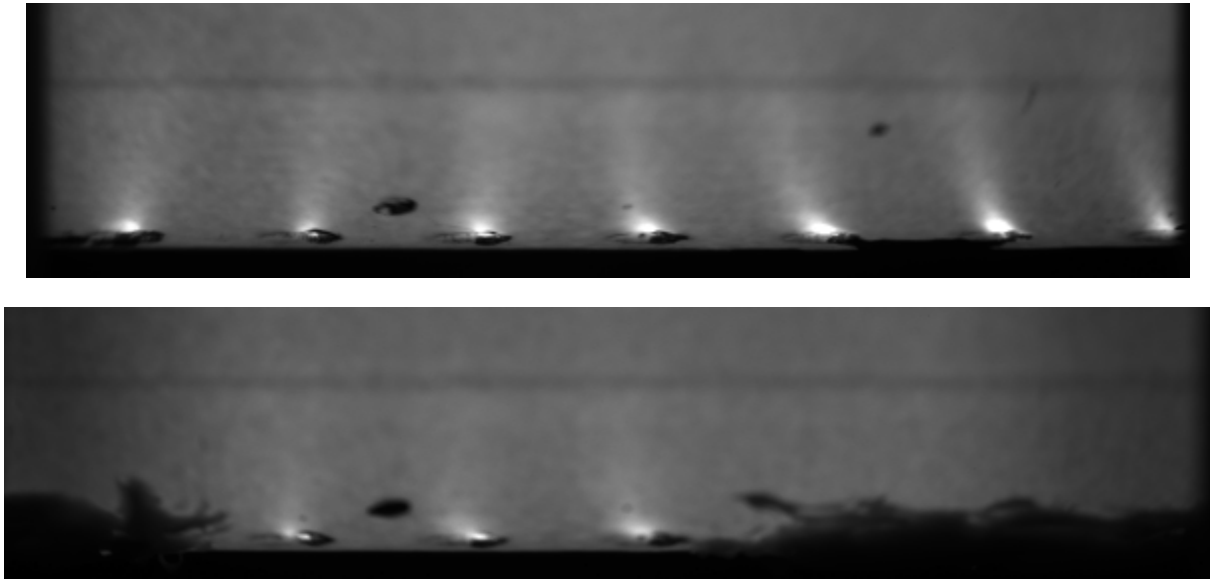


Figure 3-32: PSP coolant jets with all holes open (upper) and three holes open (lower)

After transitioning to high temperature thermal testing with the one row fan-shaped test plate, a different coolant flow effect was observed. At high coolant inlet flow rates, a cool region was observed on the upstream left side of the test plate, shown top left in Figure 3-33. The black ellipses represent where 8 of the 16 flowing film cooling holes lie. It is important to note that this image was observed live during testing and merely represents the number of photons striking the IR camera sensor. This skewed cooling effect was initially attributed to poor uniformity in coolant mass flow exiting the individual cooling holes. The experimental conditions were for a blowing ratio of 1.5 with excess internal coolant flow to feed an extra 3 rows of 16 cooling holes each. For reference, Figure 3-34 displays the IR window viewing area. The 6 middle cooling holes are also highlighted.

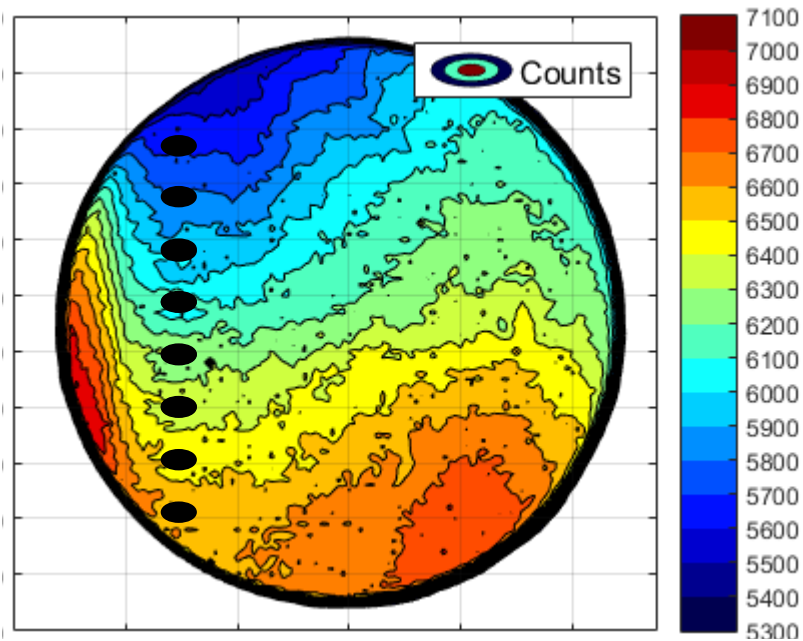


Figure 3-33: All cooling holes high temperature radiative emission

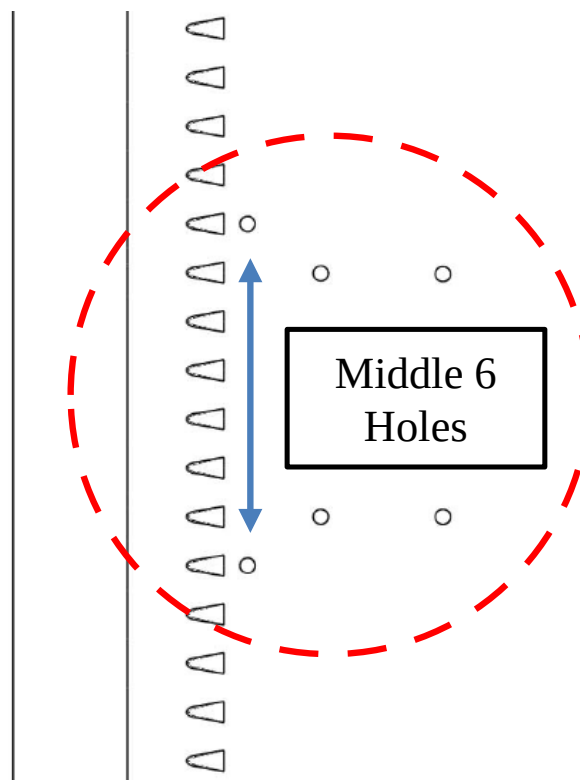


Figure 3-34: Nominal IR window viewing area

To reduce the available coolant flow area, all but six holes located nearest the test plate centerline were sealed. With only three holes open to either side of the test plate centerline, this

non-uniform cooling effect was less prominent at comparable coolant inlet flow rates, but it still persisted and completely obscured any film cooling effect. Figure 3-35 shows radiative emission contours measured with only six film cooling holes open. Again, the dashed rectangle shows where the open cooling holes lie. Experimental conditions were comparable to those for the case of Figure 3-33 except the total coolant gas flow supplied was decreased by the amount to feed 10 cooling holes at the appropriate blowing ratio.

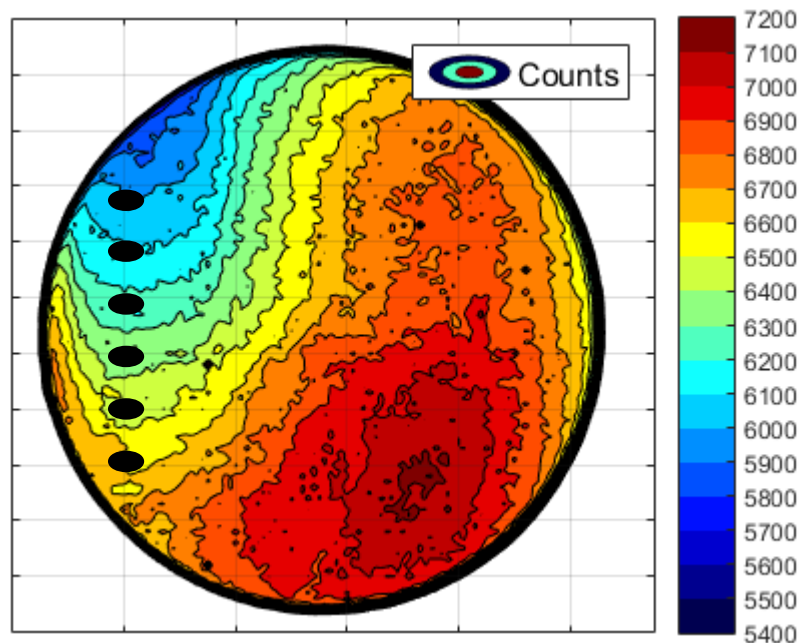


Figure 3-35: Six cooling holes high temperature radiative emission

Several brief tests were conducted to determine the root cause of this variation. Spatial variations in test plate surface emissivity and viewing window transmissivity were ruled out by turning off the coolant flow while maintaining the same freestream temperature. Using a plenum-fed cooling scheme produced fair uniformity across the test plate span, but increasing the coolant inlet flow rate skewed the coolest area of the test plate further and further to the apparent cool region at viewer's top left in Figure 3-35. The test plate was then removed to examine the uniformity of the coolant flow exiting the diffuser channel at the inlet to the main cooling block and prior to the passage beneath the test plate.

Using a tuft made of tissue paper attached to safety wire, the flow profile across the span of the diffuser channel entering the main cooling block was qualitatively characterized. At high flow rates, flow was observed to exit the diffuser in a location corresponding to the cool area at viewer's top left in Figure 3-33 while at low flow rates a coherent jet exited the center of the diffuser channel. Potential turbulence effects caused by a bend in the gas feed tube leading up to the diffuser inlet were speculated to cause this variation. Additionally, adjusting the inlet tube position allowed "aiming" of the coolant jet, suggesting that an overly aggressive diffusion angle failed to maintain attached flow through the diffuser channel. The diffusion half-angle was nearly 21.5 degrees and can be seen in Figure 3-36. Regardless of the root cause of the problem, a solution was required to slow the flow and enhance spreading.

Several solutions were considered and tested. The first solution would have placed a series of baffle plates within the diffuser channel to break up any turbulent flow structures and enhance spreading. However, the diffuser channel was not machined to specifications, so creating a part to fit within the diffuser and maintain a reasonable seal between the baffles and diffuser walls would be prohibitively difficult. Next, a single baffle plate was placed at the exit of the diffuser. The first plate used too many holes of too large a diameter, so flow was relatively undisturbed by the plate and still exited the left side of the diffuser. A subsequent plate used fewer, smaller holes to restrict exit flow area to less than the inlet flow area. While initially unsuccessful because it failed to adequately seal the irregularly shaped diffuser exit, retesting with a silicone seal around the plate edges revealed a flow profile fairly evenly spread across the span of the diffuser.

To overcome the irregular shape of the diffuser exit, the final baffle plate design was placed atop the main cooling block and completely blocked the diffuser channel exit by fitting up

against the cooling insert. An image of the final part and its place in the main cooling block assembly is shown in Figure 3-36. The test plate is not shown but sits directly atop the footprint of the baffle plate. The baffle plate features 13 holes of 0.76 mm (0.03 in) diameter forcing coolant to redistribute across the flow path as discrete jets. Based on the distance between the baffle plate and the internal surface of the test plate, a maximum spacing between the baffle holes was determined to ensure merging of the discrete jets prior to impacting the internal test plate surface. This approach was chosen to optimize coolant flow uniformity across the span of the coolant channel with minimal machining and redesign effort.

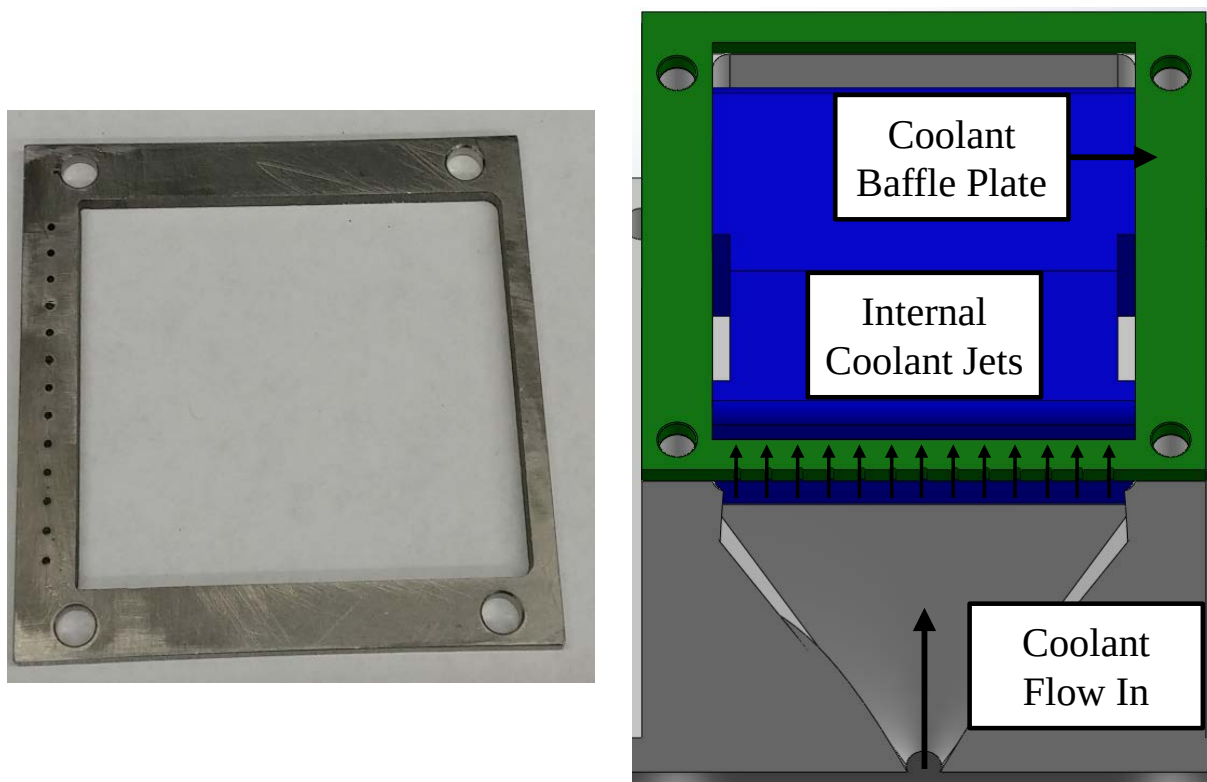


Figure 3-36: Diffuser channel baffle plate

To demonstrate the improvement in coolant distribution afforded by the baffle plate, Figure 3-37 displays radiative emission contours for the same high temperature conditions as in Figure 3-33. Total coolant supplied was identical to the conditions for Figure 3-35. The baffle

plate design of Figure 3-36 effectively distributed coolant across the internal channel, ensuring there were no localized regions of elevated cooling effect. The cooling bias towards the left hand side of the test plate (top of Figure 3-37) disappeared entirely, and the film cooling holes became readily apparent. Observed trends matched expected trends very well: radiative emission was fairly uniform across the test plate span for a given axial position, and emission was lowest immediately surrounding the film cooling holes.

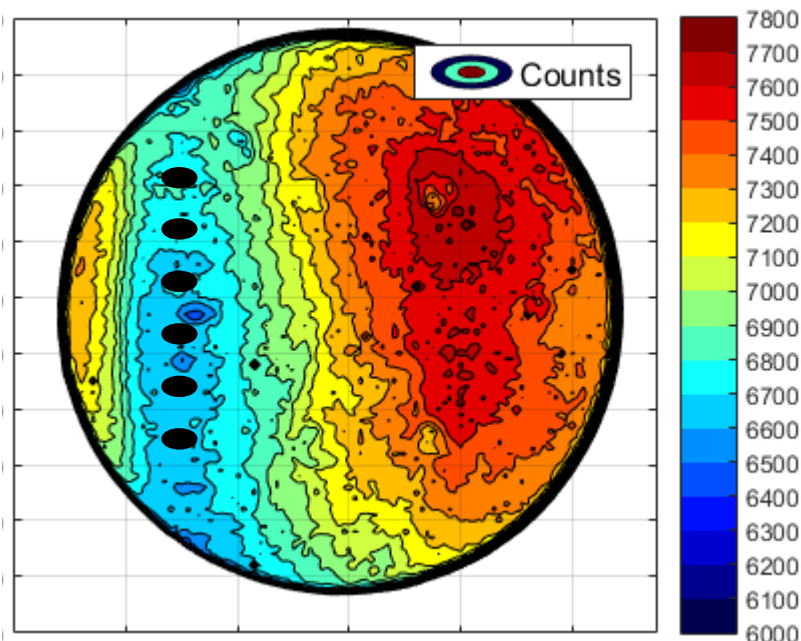


Figure 3-37: Six cooling holes with coolant baffle plate high temperature radiative emission

However, all the parts within the cooling block assembly did not fit together perfectly. Figure 3-38 shows available flow paths within the cooling block assembly, highlighting how the cooling insert (shown in blue) did not mate perfectly with the cooling block (shown in gray). This created a small gap where flow could potentially bypass the coolant baffle plate. Because the total flow area across the baffle plate was less than the cross-sectional area of the coolant inlet feed, it stands to reason that some of the coolant flow would likely seek a path of less

resistance, such as the alternate flow path depicted in Figure 3-38. No solutions to this problem are readily apparent aside from remanufacturing the entire cooling block assembly.

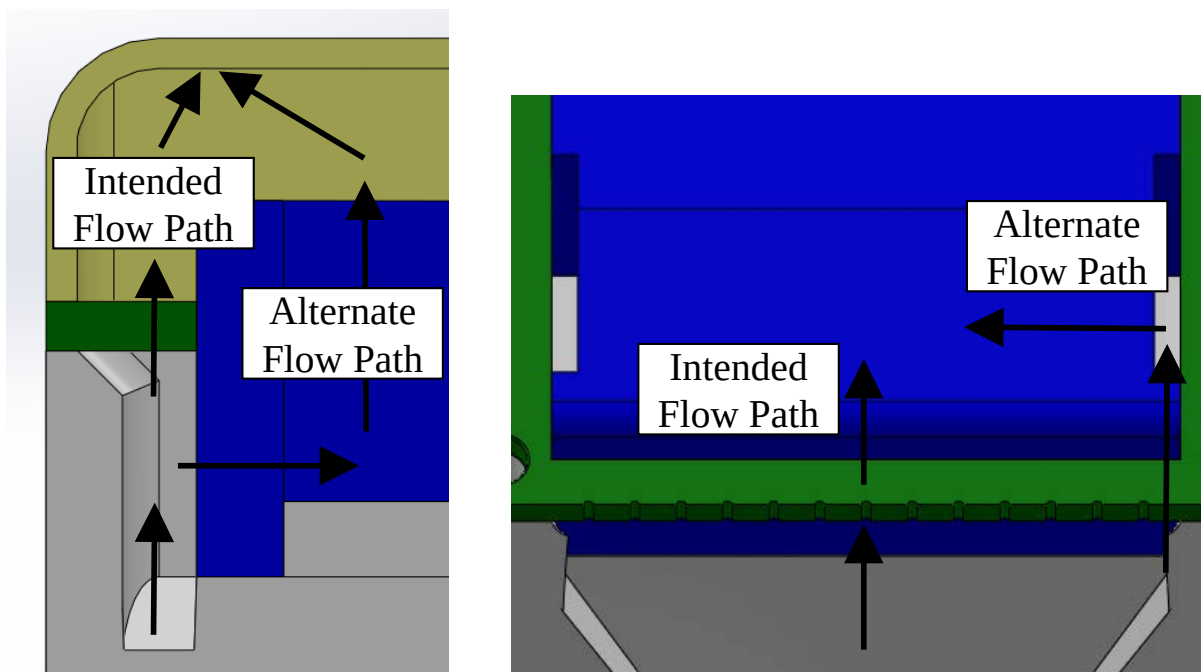


Figure 3-38: Possible internal coolant flow paths

3.4.2. Cooling Water Overheating

During several high temperature tests, the cooling water for the WSR reached dangerously high temperatures and an emergency shutdown was performed. Because the water temperature is not measured until approximately 0.3 m (1 ft) downstream of the exit point on the rig, a rapid climb in observed water temperatures already in excess of 340 K was assumed to be the result of localized boiling within the coolant channel. Peak water exit temperature reached 360 K on at least one occasion before the WSR was turned off. When an emergency shutdown was required, the maximum cooling water flow rates for both the upper and lower toroids were typically about 0.4 gallons per minute (GPM). Later consultation with lab technicians revealed that one of the two pumps for the AFIT chilled water system may have been inoperable at the time. Testing conducted on a later date provided cooling water flow rates as high as 0.7 GPM

with an average flow rate around 0.55-0.6 GPM. If the maximum cooling water flow achieved does not exceed 0.5 GPM before starting a test, inquiries should be made on the status of the chilled water system.

Prolonged difficulties with cooling the WSR were encountered. On one occasion, the cooling water flow for the upper toroid was observed rising and falling as much as 0.1 GPM above and below an average flow rate of 0.6 GPM while temperatures were observed occasionally spiking from ~325 K up to ~345 K. A visual inspection of the WSR revealed that the southern cooling passage had overheated (~1025 K) and was glowing red-hot, as shown in Figure 3-39. After an emergency shutdown and cooling period, some form of debris was observed at the exit of the cooling passage. Purging the passage with high pressure air expelled an alarming quantity of debris comprised of small flakes similar to lime scale or hard water buildup. All WSR cooling passages were later purged with high pressure water flow to prevent future flow obstruction. This flush expelled small pieces of debris from all passages and one much larger piece from the southern passage of the upper toroid. The process has begun to implement some form of filtration system for the AFIT chilled water system to help protect sensitive equipment in the future.

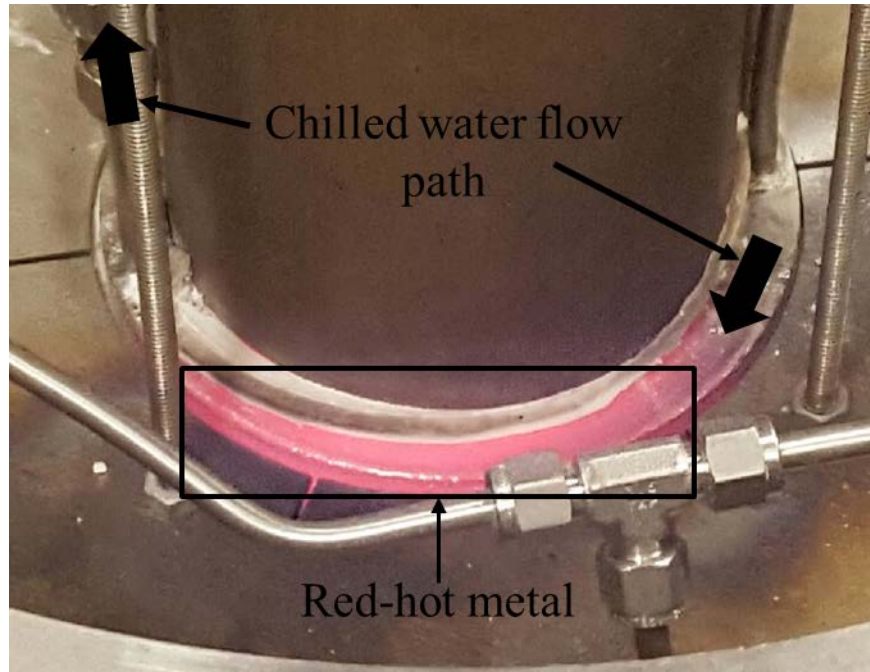


Figure 3-39: Overheated WSR upper toroid

Further, a significant water hammer effect was observed with low cooling water flow rates. Water hammer is essentially a pressure wave travelling through a fluid caused by a sudden change in fluid momentum [55]. The noticeable vibration of the copper cooling water tubes attached to the WSR was assumed to be caused by water boiling within the cooling channels and then rapidly condensing further downstream. This caused oscillations of up to 0.1 GPM in the cooling water flow rate, equating to 20-25% of the total cooling flow rate. While the hammering was very slight with the WSR off, it was greatly exacerbated when the WSR reached its high temperature operating conditions. The presence of this water hammer as well as its increased effect during high temperature WSR operation further indicates localized boiling within the coolant water channels. The volume of debris removed from the cooling passage could potentially restrict water flow, and would further support the hypothesis of boiling within the coolant water channels.

3.4.3. High Temperature Flow Leaks

Thermal expansion associated with high temperature testing presents a unique challenge. Fasteners must mate parts securely to prevent flow leakage but must also allow parts to expand or otherwise endure immense stress caused by thermal expansion. On several occasions, a gap was observed between the FCR main block and the baseplate and is shown in Figure 3-40. Greater thermal expansion of the metal side plate and front block was present because both parts are much thinner than the main block and do not have active cooling. This greater expansion of the front block and side plate raised the upper bracket, pulling the main block with it and creating a gap between the main block and baseplate. Because the upper bracket primarily serves to maintain spacing between the FCR main and front blocks, a temporary solution was devised by removing the heads of the screws attaching the upper bracket to the front block. However, a more permanent solution using longer screws and springs should be implemented to ensure a compressive load is applied to the upper bracket to anchor the FCR main block.

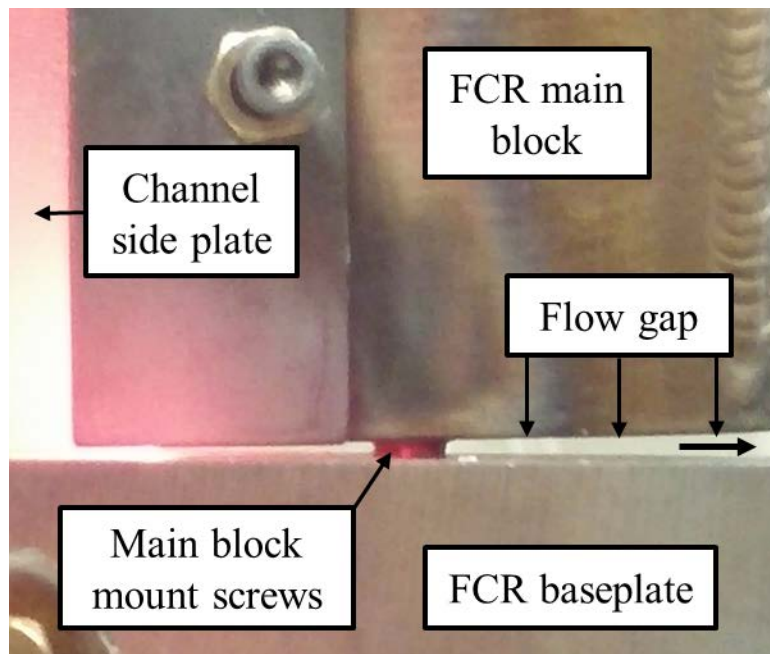


Figure 3-40: Thermal expansion induced flow gap

3.4.4. Material Limits

While thermal material limits were of little concern for near ambient tests, high temperature testing encountered material properties issues with the flow straightener. These devices were cast by lab technicians in AFRL/RQTC and were intended for use in their experimental rig. It was previously unknown to AFIT FCR investigators that these flow straighteners were made of different materials catering to specific experimental regimes examined by AFRL/RQTC. However, when exploring the operating limits of the WSR, it was found that these flow straighteners have been made of two different materials. Unknown at the time, a fused silica flow straightener with a melting temperature of 1810 K was present while the WSR core temperature was raised to 1900 K. The flow straightener began to melt and erode, ejecting red-hot pieces of glass out the top of the FCR and destabilizing WSR combustion. This destroyed flow straightener, shown in Figure 3-41, was removed, and a new flow straightener cast out of alumina was acquired, providing a melting temperature of 2065 K. When acquiring new flow straighteners in the future, whether made at AFIT or by technicians from AFRL/RQTC, special care should be taken to ensure that the material's melting temperature is sufficient for the anticipated test conditions.

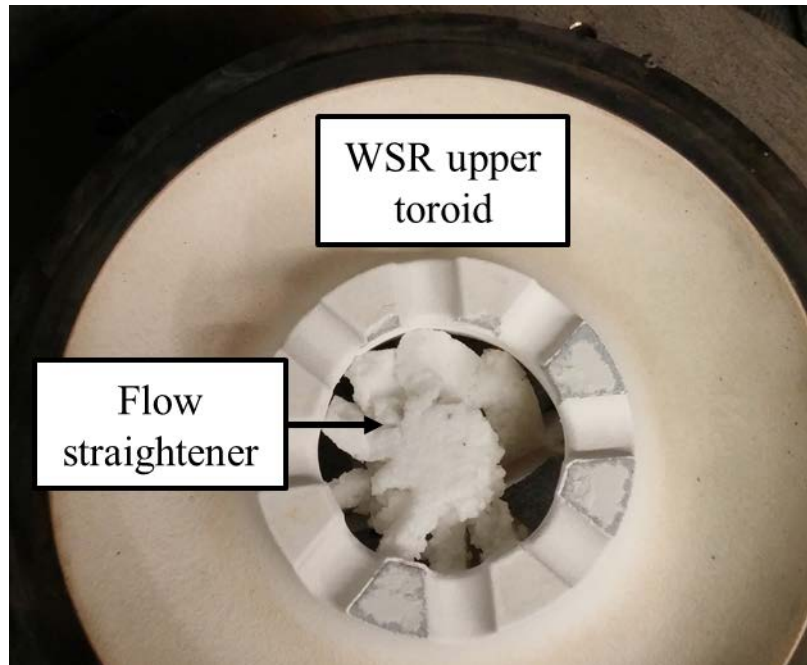


Figure 3-41: Destroyed fused silica flow straightener

3.5. Leading Edge Film Cooling Tunnel

The LEFCT consisted of an open loop wind tunnel fed by a centrifugal compressor. An upstream settling chamber and flow screens straightened the flow and allowed turbulent flow energy to decrease such that the freestream turbulence intensity was approximately $Tu = 0.7\%$ for the clean tunnel configuration [56]. Turbulence generating grids could be installed to raise the freestream turbulence intensity, but were not used for this investigation. The LEFCT also features the capacity to heat or cool the freestream flow via flow work imparted by the compressor or a chilled water heat exchanger respectively.

The model tested in the LEFCT was a large scale, non-metallic model representative of a turbine airfoil leading edge. The model geometry, illustrated in Figure 3-42, was identical to the model used by Ekkad et al. [28]. While the model could be constructed of a low conductivity material for thermal η measurements, the foam typically used for such experiments was not well suited for use with PSP. Because mass transfer methods such as PSP are unaffected by thermal

effects such as conduction through the model, a conductive model made of Corian was used for the PSP experiments described in this thesis. This material provided a smoother, more uniform surface for paint application than would otherwise be obtained with low conductivity foam.

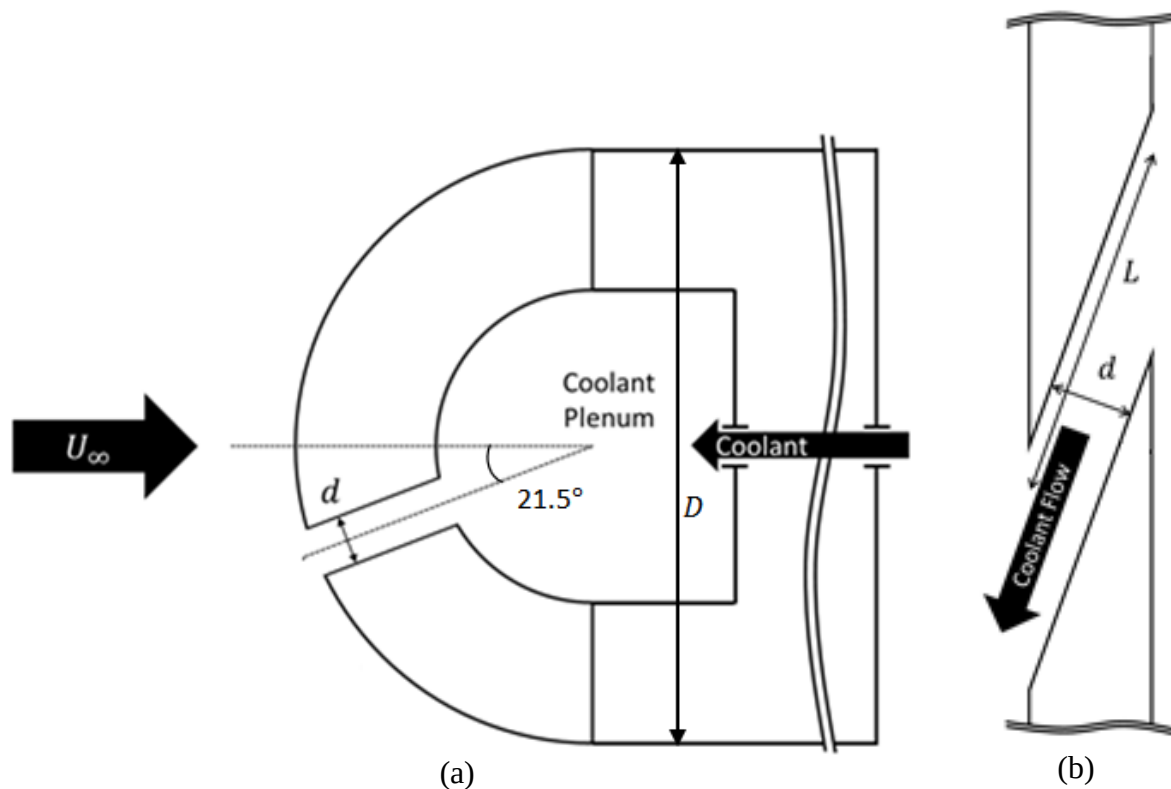


Figure 3-42: LEFCT test model schematics: (a) top view and (b) coolant hole side view (images from Wiese et al. [39])

Several key differences existed between the FCR and LEFCT test articles. The most noticeable difference between the two was their physical sizes. The FCR test plate was a one-tenth scale model of the LEFCT test article. Size limitations in the FCR made it impossible to center the test plate in the test section, requiring the quarter cylinder with tapered afterbody shape instead of a semi-cylinder with a flat afterbody. It should also be noted that the LEFCT model utilized a plenum-fed cooling scheme as compared to the channel flow arrangement present in the FCR. Any thermal effects from this difference should have negligible effects on

the PSP measurements, but, as previously noted, this internal flow difference could cause a significant aerodynamic effect. Such an aerodynamic effect would most likely be observed through the location of the coolant jet, and thus the areas of peak cooling effectiveness.

While the physical size of the FCR and LEFCT models were very different, their scaled geometries were nearly identical. Table 3-1 provides a listing of all the relevant geometric parameters for the two models. The only dimension that varied between the two models was L_h/d . While this variation could alter the heat transfer characteristics in the near-hole region, the use of a mass-transfer measurement method obviates heat transfer concerns. The impact of hole length on the jet velocity profile is of greater concern, as jet velocity plays an important role in how the coolant mixes with freestream flow.

Table 3-1: Comparison of FCR and LEFCT model dimensions

Dimension	FCR model	LEFCT model
D/d	18.7	18.7
L_h/d	7.31	11.69
α	20°	20°
γ	21.5°	21.5°

3.6. Testing Procedures

To perform the experimentation necessary to achieve the objectives for this work, two separate testing methods were used: thermal testing using IR thermography and mass-transfer testing using the Pressure Sensitive Paint (PSP) method. Thermal testing provided ϕ data necessary for scaling film cooling performance from near ambient to engine representative conditions. Experiments using the PSP method provided η data for studying model size effects. Both thermal and PSP methods were required to make effectiveness predictions, as both η and ϕ

are required to make use of Eqs. (2-44) and (2-46). A discussion of the procedures for both thermal and PSP testing follows with emphasis on data acquisition methods.

3.6.1. Thermal Testing Method

Thermal testing was conducted in two separate FCR operating regimes. The first modeled realistic turbine engine conditions with combustion products supplied by the WSR forming the freestream flow. The second approximated near ambient conditions and relied on heated air instead of WSR combustion products for freestream flow.

The experimental approach used in scaling ϕ from near ambient to engine representative conditions used engine representative conditions as a baseline. Data were collected at a variety of blowing ratios and excess internal coolant flow rates at engine representative conditions, and the relevant parameters (M , DR , I , and Re_c or Re_∞) were then matched at near ambient conditions and data was recorded. Excess internal coolant flow was determined based on rows of excess coolant, but future testing should rely on matching an internal coolant Reynolds number instead. The “rows excess coolant” metric refers to the amount of coolant needed to supply a single row of 16 film cooling holes with gas flow for a given blowing ratio. Because real gas turbine airfoils incorporate many rows of cooling but the examined test plate does not, the excess internal coolant was normalized to provide insight on how many downstream rows of cooling holes could be fed at the given condition.

On testing days, the data acquisition process began by following the detailed startup procedures in Appendix B. Procedures for both operating regimes are included. The rig was allowed time to heat up, usually about 30-40 minutes with WSR combustion or approximately 1 hour without WSR combustion. Near ambient tests required a longer warmup period because the air feed lines and heater required more time to reach a steady condition. After warmup, film

coolant was turned on and data for a repeatability condition was recorded. This condition featured $Re_{\infty,D} \approx 3800$, $DR \approx 1.65$, and $M \approx 1.05$ with two rows excess coolant flow. Engine representative conditions used air for the film coolant gas while near ambient conditions utilized carbon dioxide to provide DR values greater than would be possible if solely relying on thermal differences between freestream and coolant flow. The planned test matrix was then collected followed by another acquisition of the repeatability condition prior to rig shutdown (procedures also in Appendix B).

In between experimental points, time was allowed for the rig to reach steady state. The “thermal soak time” necessary to reach steady state differed for the two operating conditions. Based on the recommendations of Shewhart [47], a minimum of five minutes was allowed to elapse between each change in film coolant flow rate for WSR combustion conditions. The rate of change for test section thermocouples (test plate surface, backside, and internal channel) was also tracked, and steady state was deemed satisfied when this rate slowed to less than 1 K per minute. No data were recorded to quantify the time necessary to reach steady state at near ambient conditions, but a minimum of ten minutes elapsed between data points. The rate of temperature change was also observed, and data were not recorded until temperature change slowed to less than 0.3 K per minute. While these metrics and Shewhart’s [47] recommendations are useful, a detailed study of time necessary to achieve equilibrium between test points is still warranted.

Temperature data were acquired in the same manner for both operating regimes. Ten data points were acquired for each condition using a LabView program with data sampling at a nominal rate of 0.5 Hz. Data acquisition was limited by the refresh rate of the LabView program, indicating that a dedicated data acquisition processor would be of great benefit.

Further, the LabView software utilized was incapable of capturing successive data points with a single prompt, so each data point was collected individually. This contributed to inconsistent data acquisition frequency across test points. An overhaul of the LabView program by a skilled LabView programmer to ensure efficient handling of data acquisition and ensuing calculations is warranted. Further, incorporation of a dedicated data acquisition processor, such as a National Instruments chassis with data acquisition cards would adequately meet the needs for future experimentation.

IR thermography data was collected on a standalone system immediately following thermocouple data acquisition. Frames were collected at a rate of 60 Hz for two seconds and averaged to provide a single frame used for analysis. Future software modifications should seek to trigger the IR camera remotely, providing a single trigger to simultaneously capture thermocouple, mass flow rate, and IR thermography data.

3.6.2. PSP Method

In support of the objectives for predicting effectiveness values and investigating physical model size, the PSP mass-transfer method of measuring adiabatic effectiveness was developed for use in the FCR. The testing procedures and data reduction are described in detail. Obstacles faced while initially applying the PSP method are also described.

The test setup described in Section 3.3.2 was utilized to collect images necessary to calculate adiabatic effectiveness. The one row showerhead test plate discussed in Section 3.1.3 was the only cooling configuration examined during PSP testing. Four different coolant gas species were used to provide a range of density ratio values, including carbon dioxide, argon, nitrogen, and helium. Software packages necessary for image collection and pressure field calculation included ProAcquire and OMS ProImage, both supplied by Innovative Scientific

Solutions, Inc. (ISSI). Final data processing to calculate η fields was accomplished using a MATLAB script developed in-house for this study.

Consistent with the discussion of Section 2.5.3, a binary PSP was used to minimize temperature dependence in pressure calculations. Use of the binary PSP minimized temperature related errors, enabling temperature variations necessary to match density ratio when using different coolant gas species. Two separate pressure fields were required to calculate η , as shown in Eq. (2-37). The mixed pressure field featured coolant injection with a foreign gas species while the air pressure field utilized air injection at a blowing ratio equivalent to that of the foreign gas injection case. The air pressure field accounts for coolant jet dynamic pressure effects. For this study, the coolant jet dynamic pressure was assumed to be negligible and the air pressure field was generated using wind on condition of freestream flow without any coolant gas flow. However, further examination of this assumption is warranted to ensure experimental accuracy.

Pressure field calculation requires three different types of images, as presented in discussion accompanying Eq. (2-38). All images were acquired with laboratory lighting turned off to reduce background light intensity. First, a series of “dark” or “background” images were collected with no freestream or coolant gas flow and no illumination. Next, the wind off images were collected by turning on the excitation light with no freestream or coolant gas flow. Wind on images were collected with the excitation light on after establishing freestream and coolant gas flow. As previously stated, the wind on images for the air and foreign gas pressure fields differed in that the wind on condition for the air pressure field featured freestream flow and no coolant gas flow. Representative samples for each of the described image types are presented in Figure 3-43. Viewed top to bottom, left to right, the images are background, wind off, wind on

with nitrogen coolant, and wind on with only freestream flow. The test plate and cooling holes are clearly visible in all but the background image, and the regions of cooling effect immediately downstream of the cooling holes are readily apparent in the helium ejection image. All images used for pressure field calculation were an average of 100 individual frames. This large number of frames was chosen to reduce noise from pixel-to-pixel variation in time.

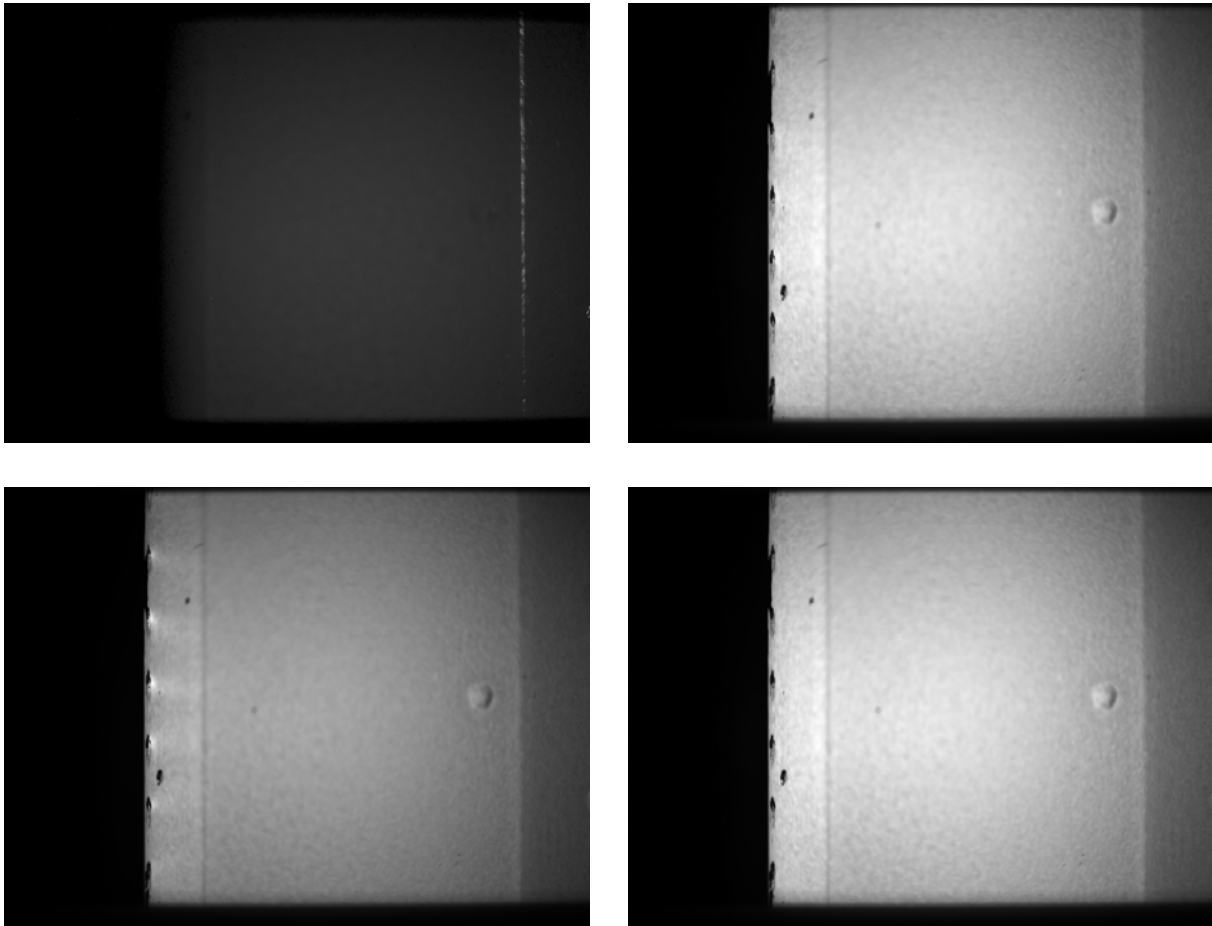


Figure 3-43: Representative raw PSP images: background (top left), wind off (top right), wind on with coolant flow (lower left), and wind on freestream only (lower right)

Ideally, the images necessary for a single pressure field would be collected in rapid succession with wind on, wind off, and dark images for each pressure field. Practically speaking, a single background and wind off image may suffice for a collection of wind on images as long as temperature and atmospheric pressure remain fairly constant for the duration

of image collection. Single background, wind off, and freestream only wind on images were shared between several points with foreign gas injection for this study. New background images were collected hourly or whenever a noticeable change in ambient lighting occurred, such as turning on a computer monitor. Wind off and freestream only wind on images were collected every time coolant gas species was changed. Pressure fields were calculated using the desired wind on image and the next ensuing wind off image. Thus, images for a single air only pressure field were collected for all wind on points of a given coolant gas species on a given day.

After image collection was completed, a series of data reduction processes was accomplished. First, the respective image series were averaged to generate a single frame. Next, images were grouped for pressure field calculation in OMS ProImage. A MATLAB script was developed to automate creation of the .ims files used for pressure field calculation. OMS ProImage then calculated pressure fields using an ISSI supplied binary paint calibration file, images supplied for comparison according to Eq. (2-38), and freestream and atmospheric properties. Filtering was also performed in OMS ProImage to help reduce noise in the pressure fields. A Gaussian filter using a three pixel diameter performed two passes on each pressure field. Finally, η was calculated for each condition using Eq. (2-37) and the appropriate pressure fields for foreign gas injection and freestream flow only. A sample result of these calculations is shown as a plot of η contours in Figure 3-44.

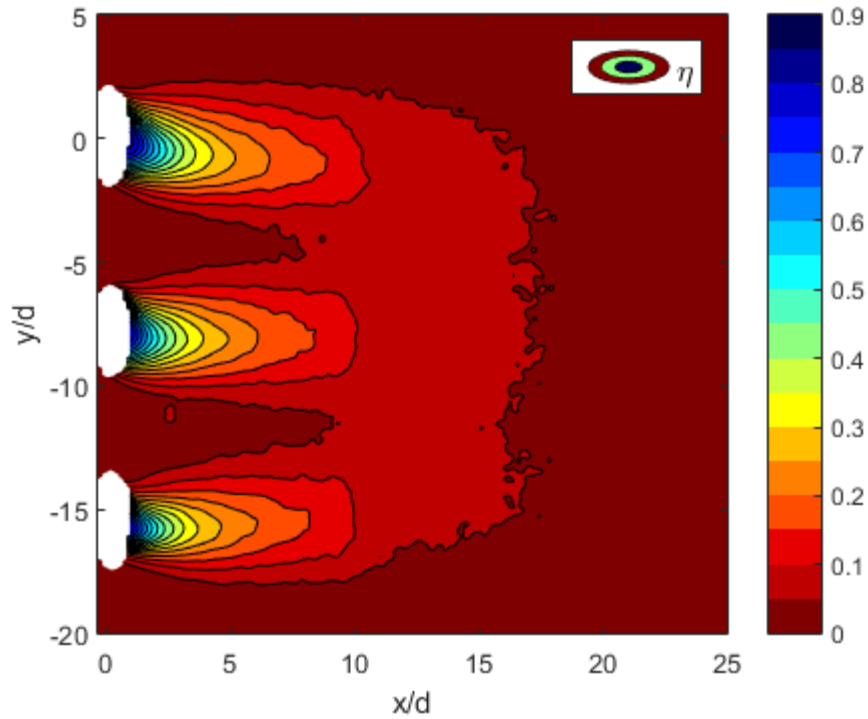


Figure 3-44: Representative PSP η contours

3.7. Data Reduction Methodology

Basic measurements acquired included various temperatures from thermocouples, a count of photons emitted from the test plate, and freestream and coolant mass flow rates. These fundamental measurements enabled calculation of relevant film cooling performance metrics and parameters. A discussion of the acquisition of these measurements and the subsequent calculations for performance metrics follows.

3.7.1. Key Temperatures and Flow Rates

Temperatures are required for calculation of many performance metrics and characteristic parameters. Useful temperatures include coolant and freestream gas temperatures and metal temperatures at various locations. The experiments for this study required different instrumentation setups to accommodate the two different temperature regimes. The high

temperature setup will be addressed first and any differences for the near ambient setup will then be discussed.

Coolant gas temperature was desired at four positions within the internal coolant channel consistent with Figure 3-18 and Figure 3-19, but internal channel thermocouple #4 was damaged and did not provide measurements. These 0.51 mm diameter K type thermocouples were anchored to the cooling insert and bent 90 degrees to extend their tips approximately halfway into the space between the cooling insert and test plate (~2.5 mm, 0.1 inch). The value for T_c was determined as the average of internal channel thermocouples C1 and C2 (Figure 3-18 and Figure 3-19).

Freestream gas temperature was initially determined by placing a 0.508 mm diameter K type thermocouple into the freestream flow through a pressure port in the test section, but this thermocouple eventually failed due to prolonged exposure to the high temperature gas flow. This test section thermocouple provided readings approximately equal to the B type thermocouple at the FCR inlet, shown previously in Figure 3-4. Thus, the B type thermocouple at the FCR inlet was used to measure T_∞ because of its capacity to operate in the high temperature environment. Near ambient tests relied on the previously mentioned test section thermocouple insertion method with a K-Type thermocouple to measure T_∞ .

Test plate surface temperatures were measured using 0.508 mm diameter K type thermocouples laser welded flush to the test plate surface. The laser welding was performed by Precision Joining Technologies. Metal temperatures along the backside of the test plate surface (the surface adjacent to the internal coolant channel) were also determined using 0.508 mm diameter K type thermocouples. The backside thermocouples were placed immediately adjacent to wires for the surface thermocouples and anchored using Ni-Cr ribbons spot welded in place.

The spot welding was initially performed using AFRL equipment in a combustion lab in Building 490, but it was found that AFIT possesses a spot welder capable of making these attachments. Spatially resolved surface temperatures were calculated using the IR calibration method described in Section 3.3.1.1.

The various mass flows were controlled using the mass flow controllers described in Table 3-2. Each flow path is listed along with the appropriate mass flow controller and full scale range expressed in Standard Liters Per Minute (SLPM) of nitrogen. Each of these flow controllers also included some form of feedback on actual flow rate. However, only the freestream flow rate was recorded through the LabView software. All other flow rates were observed and recorded based on feedback from the appropriate control unit. These flow rates included the boundary layer bleed and coolant supply and exhaust flows. The film coolant mass flow rate was determined as the difference between coolant gas supply and exhaust flow rates.

Table 3-2: Mass flow controllers

Flow Path	Temperature Regime	Controller Brand	Controller Model	Full Scale Flow Rate (SLPM)
Boundary Layer Bleed	Both	MKS	1559A	200
Coolant Supply	High	MKS	MC20A	50
Coolant Supply	Low	MKS	Alta 1480A	10
Coolant Exhaust	High	MKS	MC20A	50
Coolant Exhaust	Low	MKS	Alta 1480A	30
Freestream	Both	Fox Thermal Instruments, Inc.	FT2	3400

3.7.2. Characteristic Parameters and Performance Metrics

Various film cooling characteristic parameters are reported consistent with their definitions provided in Section 2.1.2. However, actual calculation of several parameters was performed using a modified form of those equations. Thermophysical properties for gases were estimated by linear interpolation using a table of properties over a large temperature span at a pressure of 101325 Pa. Properties for air and pure gases (argon, carbon dioxide) were obtained using the Thermophysical Properties of Fluid Systems application within the National Institute of Standards and Technology Chemistry WebBook [57] while properties for combustion products were determined using NASA Glenn's Chemical Equilibrium with Applications program [58] for propane-air combustion at an equivalence ratio of 0.8. Definitions of reported characteristic film cooling parameters as calculated are included below with appropriate discussion regarding temperatures for evaluation of thermophysical properties, area measurements, and mass flow rate measurements.

Reported parameters include those particular to film cooling, such as the density ratio, blowing ratio, momentum flux ratio, and advective capacity ratio, and those common in fluid dynamics experiments, such as the Reynolds number and Prandtl number. Although the definitions of these parameters are also presented in Section 2.1.2, they are presented again in terms of measured quantities and constants. Density ratio is calculated assuming equal pressure for the freestream and coolant gas flows, reducing Eq. (2-10) to

$$DR = \frac{\rho_c}{\rho_\infty} = \frac{P_c}{R_c T_c} \frac{R_\infty T_\infty}{P_\infty} = \frac{R_\infty T_\infty}{R_c T_c} \quad (3-4)$$

This expression for DR requires only temperatures and gas constants. The coolant and freestream temperature values are determined according to the discussion of Section 3.7.1.

Blowing ratio is also rewritten to utilize known mass flow rates and flow path areas

$$M = \frac{\rho_c U_c}{\rho_\infty U_\infty} = \frac{\dot{m}_c A_\infty}{A_c \dot{m}_\infty} = \frac{\dot{m}_c}{\frac{\pi}{4} d^2 N_{holes}} \frac{H_\infty W_\infty}{\dot{m}_\infty} \quad (3-5)$$

where H_∞ and W_∞ represent the height and width of the freestream channel over the test plate, $\dot{m}$ represents a mass flow rate, and N_{holes} represents the number of film cooling holes. Film coolant and freestream mass flow rates are determined according to the discussion of Section 3.7.1. Momentum flux ratio and advective capacity ratio are now expressed in terms of blowing ratio and density ratio.

$$I = \frac{M^2}{DR} \quad (3-6)$$

$$ACR = M \frac{c_{p,c}}{c_{p,\infty}} \quad (3-7)$$

where $c_{p,c}$ and $c_{p,\infty}$ are evaluated at T_c or T_∞ , respectively, for the appropriate gas or gaseous mixture.

Prandtl number and Reynolds number are calculated for both the freestream and coolant gas flows. All thermophysical properties are evaluated at T_c or T_∞ as appropriate. In addition, Reynolds number is rewritten to make use of measured mass flow rates

$$Pr = \frac{\mu c_p}{k} \quad (3-8)$$

$$Re = \frac{\rho U x}{\mu} = \frac{\dot{m} x}{A \mu} \quad (3-9)$$

Further, the freestream Reynolds number may be calculated using two different length scales—the airfoil leading edge diameter, D , and the cooling hole diameter, d . Cooling hole diameter is used for the freestream Reynolds number except where otherwise specified. The coolant Reynolds number always uses cooling hole diameter for the length scale.

The primary performance metrics in this study include overall effectiveness, heat flux, heat transfer coefficient, adiabatic wall temperature, and adiabatic effectiveness. Overall effectiveness is calculated consistent with the definition of Eq. (2-6) using spatially resolved test plate surface temperatures, the coolant gas temperature, and the freestream gas temperature. The calculation of T_w is described in Section 3.3.1.1 while the measurement of T_c and T_∞ is described in Section 3.7.1.

Heat flux through the test plate is calculated assuming steady, one-dimensional conduction through the test plate thickness

$$q''_{cond} = -k \frac{dT}{dx} \approx -k(T) \frac{\Delta T}{\Delta x} = -k(T) \frac{T_b - T_w}{t} \quad (3-10)$$

where $k(T)$ is a curve fit to manufacturer supplied thermal conductivity data for Hastelloy X, t represents the test plate thickness, and the subscripts b and s refer to the test plate backside and surface respectively. The expression used for $k(T)$ was developed by Lynch [48] and is given as

$$k(T) \approx 0.0194T + 4 \quad (3-11)$$

Internal heat transfer coefficient is calculated using Newton's Law of Cooling assuming steady convective heat transfer from the test plate to the internal coolant flow. Rearranging Newton's Law of Cooling, Eq. (2-1), the internal heat transfer coefficient is given as

$$h_i = \frac{q''_{cond}}{T_b - T_c} \quad (3-12)$$

where T_c is defined using the measurement from thermocouple C3 shown in Figure 3-18 and Figure 3-19.

The adiabatic wall temperature and external heat transfer coefficient were estimated using the method of Greiner et al. [30]. A representative example featured in Table 3-3 utilized two high temperature points collected at $M = 1.05$ and $DR \approx 1.55$ (#'s 16 and 17 from Table

4-1). Freestream temperature was nearly constant with only a 3 K difference between test points (1502 K and 1499 K). The points differed in that one featured a plenum-fed cooling scheme while the other received an excess coolant flow equivalent to that supplied for a single row of cooling holes at the same blowing ratio. Apparent and corrected values for heat transfer coefficient and adiabatic wall temperature were calculated consistent with Eqs. (2-17) and (2-18) (Section 2.5.1), respectively, and are displayed in Table 3-3. A representative plot demonstrating the extrapolation is shown below in Figure 3-45. Data are displayed grouped by thermocouple because heat flux varies spatially and is only known at the discrete thermocouple locations.

Table 3-3: Apparent and corrected adiabatic wall temperature and heat transfer coefficient for $M = 1.05$, $DR \approx 1.55$ high temperature condition

TC #	x/d	y/d	$\overline{T}_w$ (K)	$\overline{q}''$ (kW/m ²)	$T_{aw,app}$ (K)	T_{aw} (K)	h_{app} (W/m ² ·K)	h_{CP} (W/m ² ·K)
1	5	-14	1156	273	1208	1209	5258	5137
2	5	14	1165	254	1205	1205	6440	6327
3	10	-10	1164	287	1469	1491	943	823
4	10	10	1175	342	1227	1228	6526	6375
6	20	10	1184	412	1322	1326	2985	2808

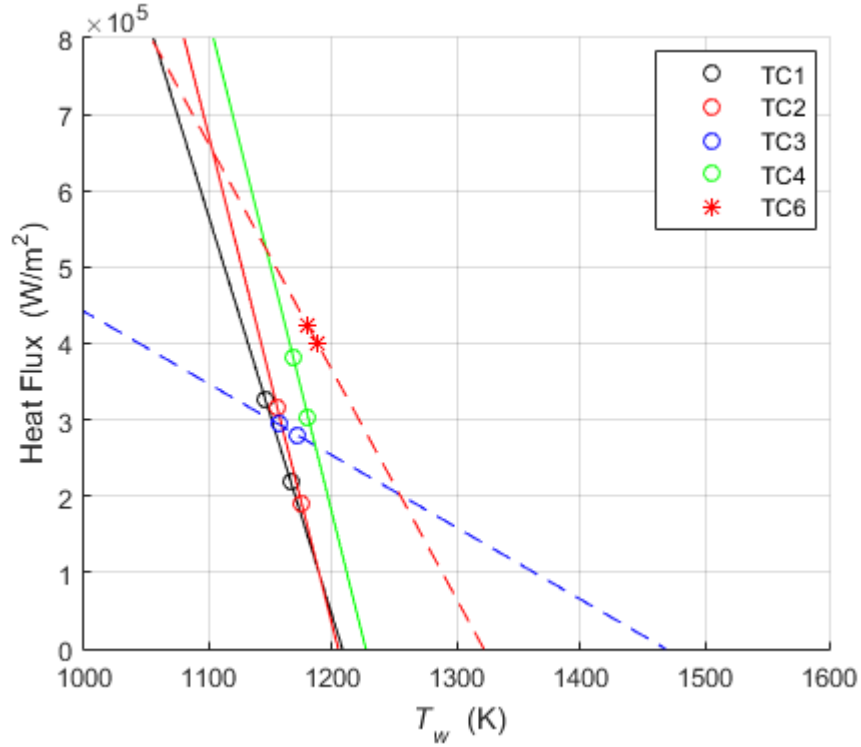


Figure 3-45: Adiabatic wall temperature extrapolation for $M = 1.05$, $DR \approx 1.55$ high temperature condition

The T_{aw} and h data of Table 3-3 were calculated according to the methods and equations described in Section 2.5.1. The two T_{aw} values reflect the apparent value ($T_{aw,app}$) observed from the extrapolation and a corrected value (T_{aw}) accounting for the curvature actually encountered when observing q'' as a function of T_w for a sufficiently large T_w range. Eq. (2-17) was used to calculate T_{aw} from $T_{aw,app}$. Assuming the actual T_w vs. q'' function is concave up, T_{aw} should be greater than $T_{aw,app}$, consistent with the trends of Table 3-3. The h values similarly reflect the observed value from the extrapolation (h_{app}) and a corrected value (h_{CP}). Eq. (2-18) was used to calculate h_{CP} , reflecting the equivalent heat transfer coefficient value if all thermophysical properties are evaluated at the freestream temperature. Because $T_\infty > T_{aw}$, using T_∞ in Newton's Law of Cooling instead of T_{aw} requires $h_{CP} < h_{app}$ to maintain a constant q'' value. This trend is also observed in Table 3-3.

3.8. Experimental Uncertainty

Experimental uncertainty must be assessed to quantify the accuracy of performance metrics calculated from experimental data. This uncertainty may stem from measurement accuracy, stability in time, or the variability in measurements acquired at identical conditions, referred to as accuracy, precision, and repeatability, respectively. Accuracy is assessed with an uncertainty propagation. An uncertainty propagation and repeatability analysis will be discussed for both high temperature combustion conditions and near ambient, non-combusting conditions.

Each representative value used for this uncertainty analysis is an average of ten points. Data acquisition was consistent with the methods described in Section 3.6.1. Data sampling frequency was limited by the refresh rate of the LabView data acquisition software (approximately 0.5 Hz) and resulted in inconsistent timing between individual measurements. All temperature measurements in the following sections, with the exception of T_{∞} at engine representative conditions, were obtained using K-type thermocouples with a manufacturer specified accuracy of $\pm 0.75\%$. Measurements of T_{∞} for engine representative conditions were obtained using a B-type thermocouple with an accuracy of $\pm 0.5\%$.

3.8.1. Precision Analysis

Precision in a given variable was assessed using the repeatability condition of $Re_{\infty} = 3800$, $DR = 1.65$, and $M = 1.05$ with two rows of excess internal coolant flow. Engine representative conditions used air for film coolant gas while near ambient conditions relied upon carbon dioxide to augment DR values to levels greater than those attainable relying strictly on thermal differences between freestream and coolant gases.

Relevant parameters and measurements for repeated engine representative conditions are presented in Table 3-4. The points are grouped by date with parenthetical reference to their

respective numbers within the experimental conditions listing of Table 4-1. Good agreement across and within days is seen for all nondimensional parameters presented. While freestream and coolant gas temperatures exhibit a slight downward trend that is undesired, it is not a cause for great concern. Decreasing freestream temperature over time may be attributed to erosion of the ceramic transition section described in Section 3.1.1 and shown in Figure 3-6. As the insulation thickness decreases, resistance to heat transfer also decreases and T_∞ decreases as the freestream gas loses more energy over the same distance of travel.

Table 3-4: High temperature repeatability conditions

Point	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_\infty$ (10^{-3} kg/min)	$\dot{m}_{c,in}$ (10^{-3} kg/min)	$\dot{m}_{c,out}$ (10^{-3} kg/min)	M (-)	I (-)	ACR (-)	Re_c (-)	Re_∞ (-)
3/11-1 (14)	917	1498	1.67	777	10.82	9.06	1.05	0.66	0.87	309	204
3/11-2 (19)	918	1490	1.66	770	10.82	9.06	1.06	0.68	0.88	308	203
3/12-1 (20)	910	1481	1.66	770	10.82	9.06	1.06	0.68	0.88	310	204
3/12-2 (11)	910	1470	1.65	782	10.82	9.06	1.04	0.66	0.87	310	208
3/17-1 (15)	901	1464	1.66	777	10.82	9.06	1.05	0.67	0.87	312	207
3/17-2 (13)	911	1471	1.65	778	10.82	9.06	1.05	0.67	0.87	310	207

Data points considered for the near ambient conditions repeatability analysis are presented in Table 3-5. Parenthetical references found in the “Point” column are a link with Table 4-3 which includes all near ambient conditions with carbon dioxide coolant gas. The spread in nondimensional parameters is greater than for the high temperature conditions of Table 3-4, but consistency in temperature values is much improved with the exception of T_c for point 4/26-2.

Table 3-5: Near ambient repeatability conditions

Point	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_\infty$ (10^{-3} kg/min)	$\dot{m}_{c,in}$ (10^{-3} kg/min)	$\dot{m}_{c,out}$ (10^{-3} kg/min)	M (-)	I (-)	ACR (-)	Re_c (-)	Re_∞ (-)
3/25-1 (51)	332	363	1.66	299	4.08	3.42	1.04	0.65	0.91	281	210
3/25-2 (49)	333	363	1.65	304	4.08	3.42	1.02	0.63	0.89	280	213
4/20-1 (56)	332	363	1.66	285	4.08	3.42	1.09	0.71	0.95	281	200
4/20-2 (54)	333	364	1.66	287	4.08	3.42	1.08	0.70	0.94	280	201
4/26-1 (52)	333	364	1.66	296	4.08	3.42	1.05	0.66	0.91	280	208
4/26-2 (57)	338	364	1.64	283	4.08	3.42	1.09	0.73	0.96	276	198

The values for precision are presented in Table 3-6 for primary measurements. These measurements include T_∞ , T_c , T_w , and T_b . Using the range of temperatures from the six measurements of the repeatability conditions in each operating regime, mean temperatures and standard deviations for each temperature measurement were found. Using the Student's t distribution, a 95% confidence interval was calculated for the mean temperature as shown in Eq. (3-13)

$$\delta T = t_{\alpha=0.025} \frac{s}{\sqrt{6}} \quad (3-13)$$

where $t_{\alpha=0.025}$ represents the Student's t inverse cumulative distribution function evaluated for 95% confidence and five degrees of freedom, s represents the sample standard deviation, and $\sqrt{6}$ represents the square root of the number of samples. This δT value was reported as the precision for a given temperature measurement. These values will be used in addition to instrument accuracies to perform an uncertainty propagation for calculated parameters.

Table 3-6: Temperature precision uncertainty values

Measurement	Engine Representative		Near Ambient	
	Mean T (K)	δT (K)	Mean T (K)	δT (K)
T_{∞}	1478.8	13.56	363.3	0.58
T_w	1127.4	10.88	345.6	1.43
T_b	1104.6	10.80	344.5	1.46
T_c	911.3	6.42	333.4	2.41
$T_{c,3}$	927.5	9.31	334.3	2.24

3.8.2. Uncertainty Propagation

The constant odds, root-sum-square method described by Moffat [59] was used to determine uncertainty in calculated performance metrics

$$\delta Z = \left[\sum_{i=1}^N \left(\frac{\partial Z}{\partial X_i} \delta X_i \right)^2 \right]^{\frac{1}{2}} \quad (3-14)$$

where Z is the parameter of interest; X is a variable of the parameter, Z ; and δ represents the uncertainty of the variable or parameter of interest.

While uncertainty could be calculated for every single measurement, a representative sampling of experimental conditions was used to determine experimental uncertainty. Maximum, minimum, and mean values were used to calculate uncertainty, representing the range of experimental conditions for both the engine representative and near ambient test matrices. Cases warranting further investigation due to excessive uncertainty include suggestions to reduce measurement uncertainty.

Uncertainty propagation calculations are presented according to the two regimes of FCR conditions: high temperature combustion and near ambient conditions. High temperature combustion conditions are presented first with near ambient conditions next. The uncertainty propagation performed here represents the minimum uncertainty in calculated parameters arising from manufacturer specified instrument accuracy. All temperature uncertainty values (δT) are

calculated as a sum of the manufacturer specified instrument accuracy and the precision error described in Section 3.8.1.

3.8.2.1. High Temperature Uncertainty Propagation

Table 3-7 includes representative surface and backside temperature measurements acquired during combustor conditions. The conditions were selected from the high temperature test matrix (Table 4-1) to characterize the range of calculated parameters due to temperature variations as experimental conditions such as M , DR , and $\dot{m}_c$ were changed. Each of these columns (Highest, Lowest, and Average T_w) consists of temperature measurements taken from a single experimental condition. The values displayed in bold in the bottom row labeled “Representative Values” are an average of the values in the rows above and were used as representative temperature values for uncertainty propagation calculations.

Table 3-7: Representative surface and backside wall temperatures – high temperature conditions

Position	Highest T_w		Lowest T_w		Average T_w	
	T_w	T_b	T_w	T_b	T_w	T_b
1	1166.4	1155.9	1014.5	968.8	1101	1074.1
2	1175.1	1166	1029.2	987.2	1111.5	1086.1
3	1171.5	1158.1	1029.3	1004.4	1117.1	1097.1
4	1180.5	1166.1	1048.6	1020.1	1131.8	1106.5
5	1180.7	Broken	1051.3	Broken	1135	Broken
6	1187.9	1169	1070.2	1037.3	1146.7	1120.8
Representative Values	1177	1166	1040.5	1003.6	1123.9	1096.9

The other critical temperature measurements included freestream and coolant gas temperatures. The values for the freestream and coolant gas temperatures are displayed in Table 3-8. The grouping of values in Table 3-8 corresponds exactly to the groupings in Table 3-7. In other words, the values for T_c , T_w , T_b , and T_∞ for the highest T_w case were all collected simultaneously.

Table 3-8: Representative gas temperatures – high temperature conditions

Position		Highest T_w	Lowest T_w	Average T_w
T_i	1	1011	730	883.5
	2	1009.1	722	872.1
	3	1030.2	762.8	900.9
T_c		1010.1	726	877.8
T_∞		1502.4	1472	1503.9

Using the representative values from Table 3-7 and the T_c and T_∞ values from the highest, lowest, and average T_w columns of Table 3-8, Eq. (3-14) was used to perform an uncertainty propagation analysis for all parameters of interest. The first variable considered was overall effectiveness, ϕ . Uncertainty values for overall effectiveness are shown in Table 3-9. The values for each variable as well as the absolute value of its sensitivity coefficient $\left(\frac{\partial \phi}{\partial x_i}\right)$ are displayed to help evaluate sources of error. Temperature uncertainties were calculated as the root-sum-square of the manufacturer specified thermocouple accuracy (0.5% or 0.75% as appropriate for B- and K-Type thermocouples) and the appropriate precision uncertainty value as described in Section 3.8.1. While little can be done to reduce uncertainty in ϕ due to instrument accuracy, increasing the difference between the coolant and wall temperatures does reduce $\delta\phi$ appreciably. However, the maximum uncertainty value of $\delta\phi = 0.032$ presented in Table 3-9 will be used to evaluate the significance of experimental results as it provides a more conservative viewpoint.

Table 3-9: Overall effectiveness uncertainty – high temperature conditions

Variable	Value			δX_i			$\left \frac{\partial \phi}{\partial X_i} \right $		
	High	Low	Avg	High	Low	Avg	High	Low	Avg
T_∞ (K)	1502.4	1472	1503.9	15.50	15.43	15.51	$6.89 \cdot 10^{-4}$	$5.65 \cdot 10^{-4}$	$6.28 \cdot 10^{-4}$
T_w (K)	1177	1040.5	1123.9	14.01	13.39	13.77	$2.00 \cdot 10^{-3}$	$1.30 \cdot 10^{-3}$	$1.60 \cdot 10^{-3}$
T_c (K)	1010.1	726	877.8	7.12	6.06	6.60	$1.30 \cdot 10^{-3}$	$8.0 \cdot 10^{-4}$	$1.0 \cdot 10^{-3}$
ϕ	0.661	0.578	0.607	0.032	0.021	0.025			

Thermal conductivity was calculated using Eq. (3-11) with T evaluated as the average between the surface and backside wall temperatures. This δk value was then used for uncertainty calculations for both heat flux and internal heat transfer coefficient. Values used to calculate uncertainty in thermal conductivity of the Hastelloy X test plate are displayed in Table 3-10. While thermocouple accuracy is acceptable, calculating k using the average of the test plate surface and backside temperatures presents potential difficulties from oversimplifying temperature variation effects. Additional discussion on this issue follows.

Table 3-10: Hastelloy X thermal conductivity uncertainty – high temperature conditions

Variable	Value			δX_i			$\left \frac{\partial k}{\partial X_i} \right $
	High	Low	Avg	High	Low	Avg	
T_w (K)	1177	1040.5	1123.9	14.01	13.39	13.77	0.0097
T_b (K)	1166	1003.6	1096.9	13.88	13.17	13.58	0.0097
$k \left(\frac{W}{m \cdot K} \right)$	26.69	23.81	25.52	0.19	0.18	0.19	

Bohan [51] also calculated heat flux using a near identical instrumentation setup. However, he used an integration approach to arrive at an expression for heat flux, given as

$$q'' = \frac{\frac{a_1}{2}(T_1^2 - T_2^2) + a_0(T_1 - T_2)}{x_2 - x_1} \quad (3-15)$$

where a_0 and a_1 are curve fit coefficients for thermal conductivity, expressed as

$$k(T) = a_1 T + a_0 \quad (3-16)$$

Evaluating Eq. (3-16) with T as the average of two temperatures and substituting the result into Eq. (3-15) for heat flux yields the same result as Eq. (3-10) so long as consistent notation is utilized.

$$q'' \approx -k(T) \frac{T_2 - T_1}{x_2 - x_1} = \frac{\frac{a_1}{2}(T_1^2 - T_2^2) + a_0(T_1 - T_2)}{x_2 - x_1} \quad (3-17)$$

Thus, Bohan's expression for heat flux is consistent with the expression utilized here and previously by Lynch [48] and Shewhart [47]. No further recommendations are available to account for temperature variation through the test plate thickness aside from distributing temperature measurement instrumentation through the thickness of the plate.

Heat flux was calculated using Eq. (3-10) with thermal conductivity calculated as described above. Relevant values for the heat flux uncertainty calculations are displayed in Table 3-11. Maximum, minimum, and mean uncertainty values are presented to illustrate the wide range in uncertainty for heat flux measurements. Experimental results will be evaluated using the $\delta q''$ value of $398 \text{ kW}/\text{m}^2$. The lower uncertainty for the lowest experimental temperatures reveals that a combination of lower metal temperatures and larger surface-to-backside temperature differences produces more accurate heat flux results. Increasing test plate thickness would contribute to increasing ΔT between the internal and external test plate surfaces, thus reducing $\delta q''$. This is consistent with previous work done by Lynch [48] on the same test rig. The temperature values used by Lynch were comparable to those for the minimum T_w condition, resulting in heat flux uncertainty values of 37 % for the flat portion of the test plate

and 15% on the thicker sloped after body section. However, Lynch did not account for precision uncertainty in his uncertainty propagation. If $\delta q''$ is calculated using only instrument accuracy for the lowest T_w condition, the $\delta q''$ values are comparable to those reported by Lynch [48].

Table 3-11: Heat flux uncertainty – high temperature conditions

Variable	Value			δX_i			$\left \frac{\partial q''}{\partial X_i} \right $		
	High	Low	Avg	High	Low	Avg	High	Low	Avg
$k \left(\frac{W}{m \cdot K} \right)$	26.69	23.81	25.52	0.19	0.18	0.19	$1.04 \cdot 10^4$	$2.74 \cdot 10^4$	$1.94 \cdot 10^4$
T_w (K)	1177	1040.5	1123.9	14.01	13.39	13.77	$2.10 \cdot 10^4$	$1.87 \cdot 10^4$	$2.01 \cdot 10^4$
T_b (K)	1166	1003.6	1096.9	13.88	13.17	13.58			
t_w (mm)	1.27			$2.14 \cdot 10^{-4}$			$2.20 \cdot 10^8$	$5.12 \cdot 10^8$	$3.91 \cdot 10^8$
$q'' \left(\frac{kW}{m^2} \right)$	279	650	496	418	370	398			
% error	150 %	56.9 %	80.1 %						

Internal heat transfer coefficient was calculated using Eq. (3-12). Table 3-12 presents the maximum, minimum, and mean uncertainty values for the h_i values calculated over the course of this study. Values displayed for T_i correspond to the third internal channel thermocouple position of Table 3-8. The bulk of the uncertainty in h_i stems from the difficulty in obtaining an accurate heat flux measurement. Any comparisons of h_i values will use the mean δh_i value of 104 % to evaluate significance.

Table 3-12: Internal heat transfer coefficient uncertainty – high temperature conditions

Variable	Value			δX_i			$\left \frac{\partial h_i}{\partial X_i} \right $		
	High	Low	Avg	High	Low	Avg	High	Low	Avg
$q'' \left(\frac{kW}{m^2} \right)$	279	650	496	418	370	398	$7.5 \cdot 10^{-3}$	$4.2 \cdot 10^{-3}$	$5.1 \cdot 10^{-3}$
T_w (K)	1177	1040.5	1123.9	14.01	13.39	13.77	15.71	11.93	13.20
T_i (K)	1030.2	762.8	900.9	12.10	10.92	11.50			
$h_i \left(\frac{kW}{m^2 \cdot K} \right)$	2.09	2.77	2.55	3.16	1.56	2.05			
% error	151 %	56.5 %	80.5 %						

3.8.2.2. Near Ambient Uncertainty Propagation

The uncertainty described for the non-combusting, near ambient FCR conditions follows the same format as for the combusting, high temperature FCR conditions in Section 3.8.2.1. Table 3-13 and Table 3-14 include representative temperature measurements acquired during non-combusting, near ambient conditions. The assortment of temperatures includes metal temperatures from the test plate surface and backside as well as gas temperatures from the internal coolant channel and freestream flows. These temperatures indicate the highest, lowest, and average temperatures encountered during the entire test matrix of points acquired with non-combusting conditions. It is worthwhile to note that additional thermocouples failed during the course of near ambient testing, including surface thermocouple 1 and backside thermocouples 4 and 5 (see Figure 3-18 for detail).

Table 3-13: Representative surface and backside wall temperatures – near ambient conditions

Position	Highest T_w		Lowest T_w		Average T_w	
	T_w (K)	T_b (K)	T_w (K)	T_b (K)	T_w (K)	T_b (K)
1	Broken	374.05	Broken	340.71	Broken	349.26
2	376.17	375.04	342.17	341.25	351.32	350.08
3	375.76	374.82	342.24	341.40	351.29	350.31
4	376.30	Broken	342.52	Broken	351.72	Broken
5	375.87	Broken	342.47	Broken	351.57	Broken
6	376.60	375.57	343.00	342.34	352.15	351.37
Representative Values	376.14	374.87	342.48	341.43	351.61	350.26

Table 3-14: Representative gas temperatures – near ambient conditions

Position		Highest T_w	Lowest T_w	Average T_w
T_{int} (K)	1	357.66	329.38	336.64
	2	357.71	329.45	336.63
	3	359.03	330.49	337.24
T_c (K)		357.68	329.42	336.64
T_∞ (K)		395.11	361.32	371.13

All temperature measurements in Table 3-13 and Table 3-14 were obtained using K-type thermocouples with a manufacturer specified accuracy of $\pm 0.75\%$. Additional detail on thermocouple placement within the FCR and data reduction procedures may be found in Section 3.7.

Uncertainty in overall effectiveness is shown in Table 3-15. Test plate surface and backside temperatures are derived from the bottom row labeled “Mean” in Table 3-13. In the same fashion as in Section 3.8.2.1, an uncertainty value for overall effectiveness of $\delta\phi = 0.172$ was calculated. Remarkably, this $\delta\phi$ value is even higher than for the high temperature conditions. The root cause of this issue lies in the sensitivity coefficients. Reducing the temperature magnitudes increased the value of the sensitivity coefficients enough to increase $\delta\phi$ by nearly a factor of 5.

Table 3-15: Overall effectiveness uncertainty – near ambient conditions

Variable	Value			δX_i			$\left \frac{\partial \phi}{\partial X_i} \right $		
	High	Low	Avg	High	Low	Avg	High	Low	Avg
T_∞ (K)	395.11	361.32	371.13	3.03	2.78	2.80	$1.32 \cdot 10^{-2}$	$1.28 \cdot 10^{-2}$	$1.26 \cdot 10^{-2}$
T_w (K)	376.14	342.48	351.61	3.88	3.70	3.73	$2.67 \cdot 10^{-2}$	$3.13 \cdot 10^{-2}$	$2.90 \cdot 10^{-2}$
T_c (K)	357.68	329.42	336.64	3.73	3.65	3.67	$1.35 \cdot 10^{-2}$	$1.85 \cdot 10^{-2}$	$1.64 \cdot 10^{-2}$
ϕ	0.507	0.591	0.566	0.122	0.139	0.172			

Uncertainty in thermal conductivity decreased slightly to $\delta k = 0.48 \%$, as shown in Table 3-16. Temperature values used for these calculations correspond to a positional average of the maximum temperatures shown in Table 3-14. The fixed nature of the sensitivity coefficients for k means that an increase or decrease in temperature will result in an accompanying increase or decrease in δk . Thus, as temperature decreases, δk decreases.

Table 3-16: Hastelloy X thermal conductivity uncertainty – near ambient conditions

Variable	Value			δX_i			$\left \frac{\partial k}{\partial X_i} \right $
	High	Low	Avg	High	Low	Avg	
T_w (K)	376.14	342.48	351.61	3.88	3.70	3.73	0.0097
T_b (K)	374.87	341.43	350.26	3.92	3.74	3.77	0.0097
$k \left(\frac{W}{m \cdot K} \right)$	11.29	10.64	10.81	0.054	0.051	0.052	
% error	0.47 %	0.48 %	0.48 %				

Table 3-17 displays heat flux uncertainty values for near ambient conditions. Uncertainty in heat flux was dramatically greater for near ambient conditions as compared to high temperature conditions. This is because ΔT between the external and internal test plate surfaces is much smaller. Heat flux uncertainty at near ambient conditions could be readily reduced by

increasing test plate thickness or using a lower conductivity test plate material, either of which would produce a larger ΔT through the test plate thickness.

Table 3-17: Heat flux uncertainty – near ambient conditions

Variable	Value			δX_i			$\left \frac{\partial q''}{\partial X_i} \right $		
	High	Low	Avg	High	Low	Avg	High	Low	Avg
$k \left(\frac{W}{m \cdot K} \right)$	11.29	10.64	10.81	0.054	0.051	0.052	812.4	633.2	786.8
T_w (K)	376.14	342.48	351.61	3.88	3.70	3.73	$8.89 \cdot 10^3$	$8.37 \cdot 10^3$	$8.51 \cdot 10^3$
T_b (K)	374.87	341.43	350.26	3.92	3.74	3.77			
t_w (mm)	1.27			$2.14 \cdot 10^{-4}$			$7.22 \cdot 10^6$	$5.30 \cdot 10^6$	$6.70 \cdot 10^6$
$q'' \left(\frac{kW}{m^2} \right)$	9.17	6.73	8.50	49.1	44.1	45.0			
% error	535 %	655 %	751 %						

Table 3-18 displays uncertainty values for internal heat transfer coefficient data at near ambient conditions. As with the high temperature uncertainty data in Section 3.8.2.1, the difference in percent error for q'' and h_i is quite small. However, the percentage uncertainty in h_i is unacceptably high at over 500 %. The solutions to reduce heat flux uncertainty should also serve to reduce internal heat transfer coefficient uncertainty.

Table 3-18: Internal heat transfer coefficient uncertainty – near ambient conditions

Variable	Value			δX_i			$\left \frac{\partial h_i}{\partial X_i} \right $		
	High	Low	Avg	High	Low	Avg	High	Low	Avg
$q'' \left(\frac{kW}{m^2} \right)$	9.17	6.73	8.50	49.1	44.1	45.0	0.062	0.090	0.075
T_w (K)	376.14	342.48	351.61	3.88	3.70	3.73	35.3	54.8	48.7
T_i (K)	359.03	330.49	337.24	5.00	4.88	4.92			
$h_i \left(\frac{W}{m^2 \cdot K} \right)$	569.1	607.1	642.9	$3.05 \cdot 10^3$	$3.97 \cdot 10^3$	$5.00 \cdot 10^3$			
% error	536 %	653 %	751 %						

3.8.3. Repeatability Analysis

The ability to consistently receive the same result when performing the same experiment is an important issue that must be addressed when considering the significance of experimental results. For this research effort, a single experimental condition was identified for repeated data collection. The condition selected for both high temperature and near ambient conditions was a blowing ratio $M = 1$ with 2 rows excess coolant flow and no heat added to the coolant gas through the OmegaLux heaters described in Section 3.2.1. Repeatability was considered primarily for overall effectiveness, although raw temperature measurements are also briefly considered.

3.8.3.1. High Temperature Conditions

As previously stated, the repeated condition for this analysis was a blowing ratio of $M = 1$ with 2 rows of excess coolant supplied and no coolant heating. This normalization of excess coolant flow is discussed in Section 3.6.1. For high temperature conditions, air was used as the coolant gas. On every day of testing, this condition was both the first and last test point collected.

Repeatability was considered in two respects: day-to-day variability and variability within a given day. Day-to-day variability is presented by comparing either the first points collected on each day or the final points collected. Figure 3-46 shows span-averaged overall effectiveness values for the first points collected on each day of high temperature testing. Span averaging was performed over the middle two holes on the test plate. The data of the first two days agree extremely well in magnitude with slight deviations in profile. While the data show a slight increase with time, the magnitude of this change does not appear to exceed the calculated uncertainty of $\delta\phi = 0.032$. The IR calibration discussion of Section 3.3.1.1 did reveal increased test plate emission with increasing run time. Accordingly, this trend of increasing $\bar{\phi}$ with time will be considered when evaluating experimental results.

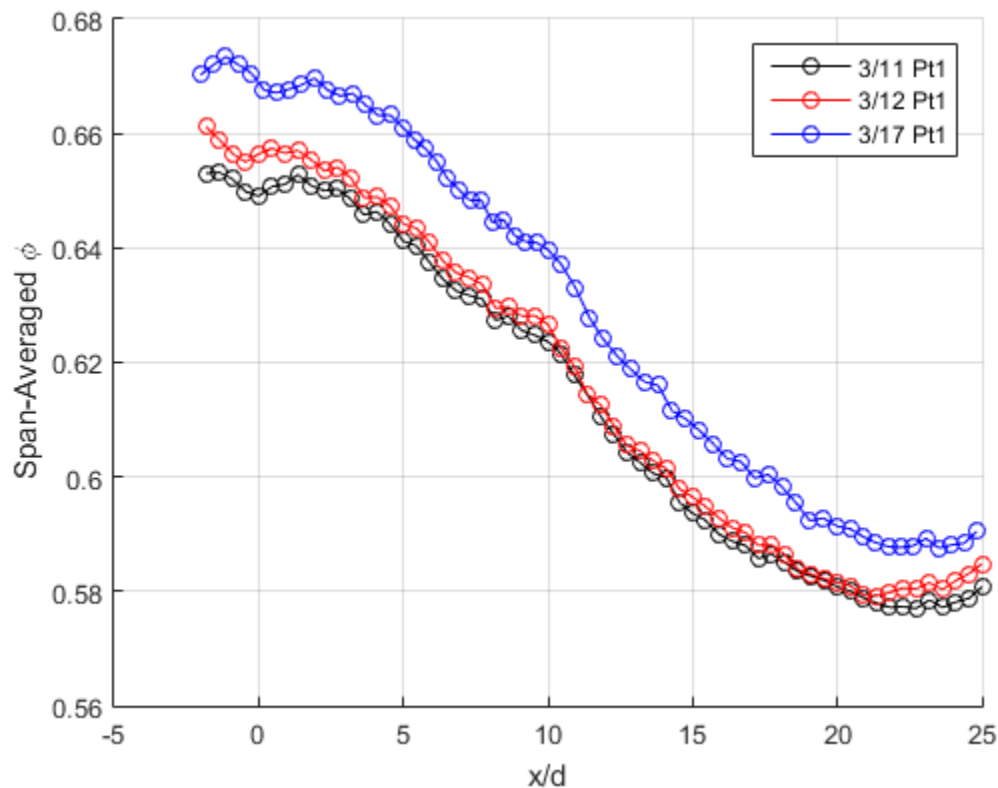


Figure 3-46: Day-to-day high temperature ϕ variability – first daily points

Data for the points collected at the end of a day's testing are presented in Figure 3-47. There is excellent agreement in profile, but the trend of increasing $\bar{\phi}$ with time is still apparent. The magnitude of change still appears to lie within the calculated uncertainty band ($\delta\phi = \pm 0.032$), but the overall values for the final points (Figure 3-47) appear greater than for the initial points (Figure 3-46). While the IR calibration discussion of Section 3.3.1.1 revealed that test plate emissivity varies in time, this would only cause T_w to appear greater, causing an apparent reduction in ϕ . Further, the calibration used for all IR image processing used the entire range of combustion condition data, limiting the impact of test plate emissivity variation in time. Change within a given day also appears to be within the $\delta\phi$ uncertainty band, but a more exhaustive study of $\bar{\phi}$ repeatability with more points collected over a greater time span is warranted. While test plate emissivity does appear to increase slightly with time, this effect was accounted for by performing an in situ calibration using data from the experimental points (see Section 3.3.1.1).

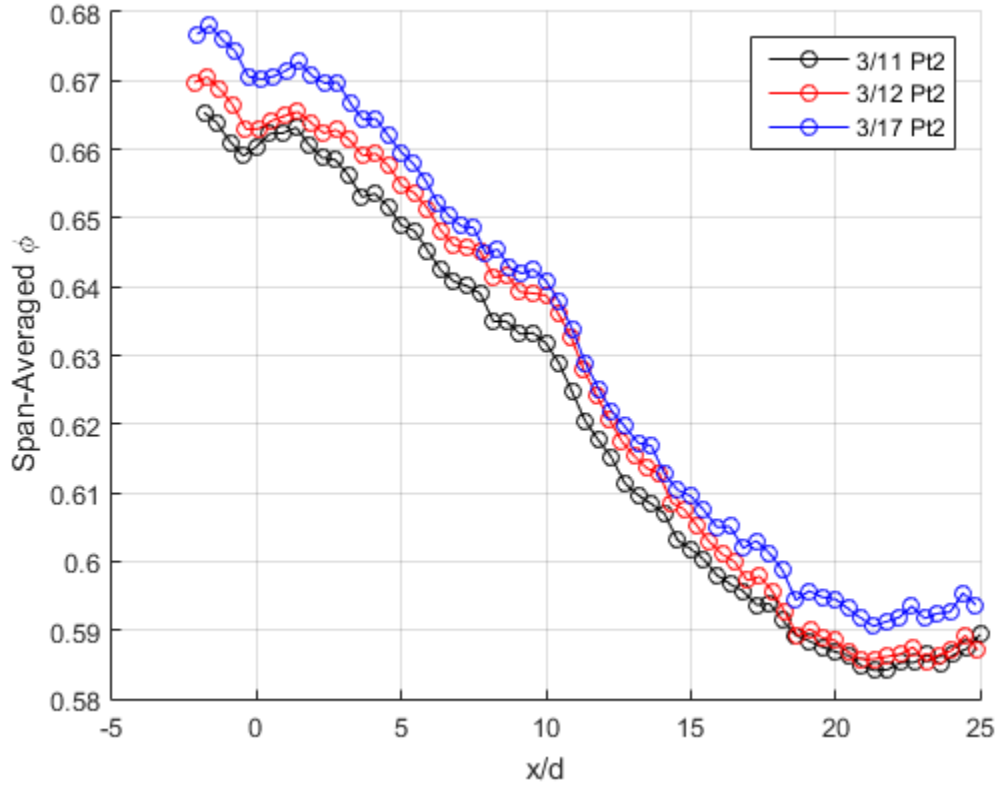


Figure 3-47: Day-to-day high temperature ϕ variability – final daily points

Variation within a day's testing is addressed by calculating a value for $\Delta\bar{\phi} = \bar{\phi}_2 - \bar{\phi}_1$ over the full x/d range as shown in Figure 3-48. No discernible trend of the presented $\Delta\bar{\phi}$ data with time is apparent, but the positions for greatest $\Delta\bar{\phi}$ values vary from day to day. Data from 3/11 features a peak $\Delta\bar{\phi}$ value of $\Delta\bar{\phi} \approx 0.012$, while data from 3/12 features peak $\Delta\bar{\phi}$ of $\Delta\bar{\phi} \approx 0.018$ and data from 3/17 features peak $\Delta\bar{\phi}$ of $\Delta\bar{\phi} \approx 0.006$. Data from 3/11 has greatest $\Delta\bar{\phi}$ values ($\Delta\bar{\phi} \approx 0.01$) over the range $0 < x/d < 20$ while data from 3/12 and 3/17 features their lowest $\Delta\bar{\phi}$ values ($\Delta\bar{\phi} \approx 0.007, -0.005$ respectively) over this same range. The cause of this phenomenon is not readily apparent. The data in Figure 3-48 do conclusively show that $\bar{\phi}$ values collected within a single day lie within the calculated uncertainty range of $\delta\phi = \pm 0.032$.

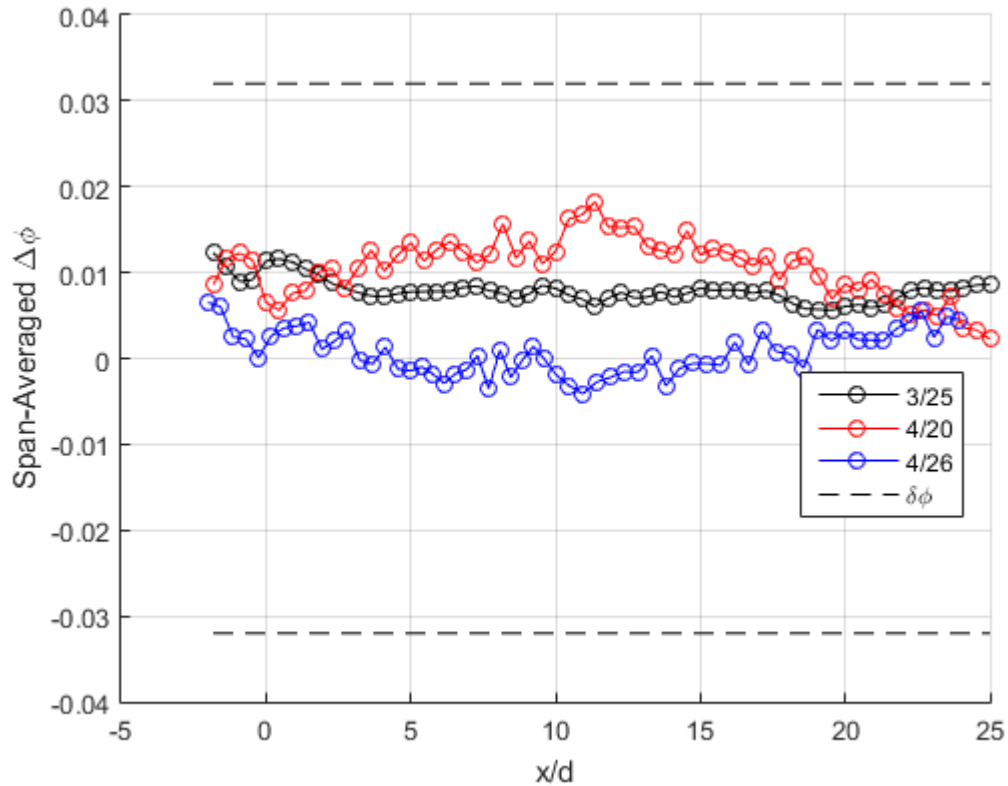


Figure 3-48: Single day high temperature ϕ variability

This high temperature repeatability analysis indicates that the FCR is capable of producing near-identical results over time. When considering the uncertainty in the data ($\delta\phi = 0.032$), there is no conclusive difference between the data points considered. However, the data were only collected over a 3 day testing span with 14 total hours of FCR combustion run time. An analysis carried out with more data points collected over a longer period of time would provide a much better quantification of FCR repeatability at high temperature conditions.

3.8.3.2. Near Ambient Conditions

The near ambient conditions selected for repeatability runs was selected to match the high temperature repeatability conditions as closely as possible. This required using carbon dioxide as the coolant gas with conditions set identically to the high temperature repeatability

condition ($Re_{\infty,D} = 3800$, $DR = 1.65$, $M = 1$ with 2 rows excess coolant). A discussion on this excess coolant normalization scheme is provided in Section 3.6.1.

A comparison of $\bar{\phi}$ values for the first points collected daily is shown in Figure 3-49. These $\bar{\phi}$ values were also generated by span averaging across the two middle holes on the test plate. Profiles agree quite well for all data points in Figure 3-49, but there is some disagreement in $\bar{\phi}$ magnitudes. Data points collected on 3/25 and 4/26 lie within approximately $\Delta\bar{\phi} = 0.003$ of each other, but the data from 4/20 differs by an apparent bias of approximately $\Delta\bar{\phi} = 0.02$ across the entire x/d range considered. However, this difference lies within the uncertainty calculated for near ambient conditions in Section 3.8.2.2, $\delta\phi = 0.172$. Further, no conditions outlined in Table 3-5 suggest a significant difference in experimental conditions for Point 4/20-1 that would generate the apparent bias observed.

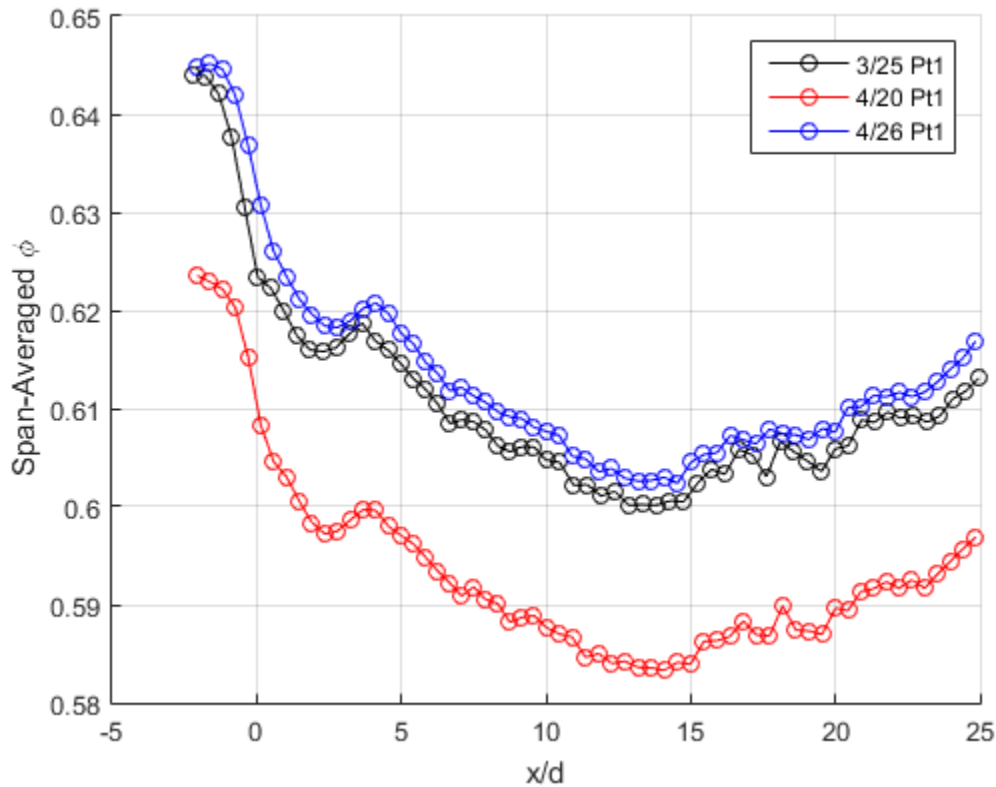


Figure 3-49: Day-to-day near ambient ϕ variability – first daily points

Figure 3-50 displays $\bar{\phi}$ data for the final points collected on a daily basis. Agreement is not quite as strong between the points on 3/25 and 4/26 as for the first daily points, and the two points differ by approximately $\Delta\bar{\phi} = 0.015$ near $x/d = -2$. This difference gradually decreases with axial position until $x/d = 15$, after which point the two data sets agree quite well. Data from 4/20 differs from the other two days by approximately $\Delta\bar{\phi} = 0.025$ over the full x/d range considered. However, the profile for 4/20 matches the 4/26 profile very well with a near constant offset between the two datasets. Nothing in the data of Table 3-5 suggests a difference sufficient to create the $\Delta\bar{\phi} = 0.025$ difference between the data of 4/20 and 4/26. In fact, the reduced values of T_c and Re_∞ for the data of 4/26 would suggest data from that day should differ. Regardless, none of the variations of Figure 3-50 exceed the near ambient uncertainty of $\delta\phi = 0.172$.

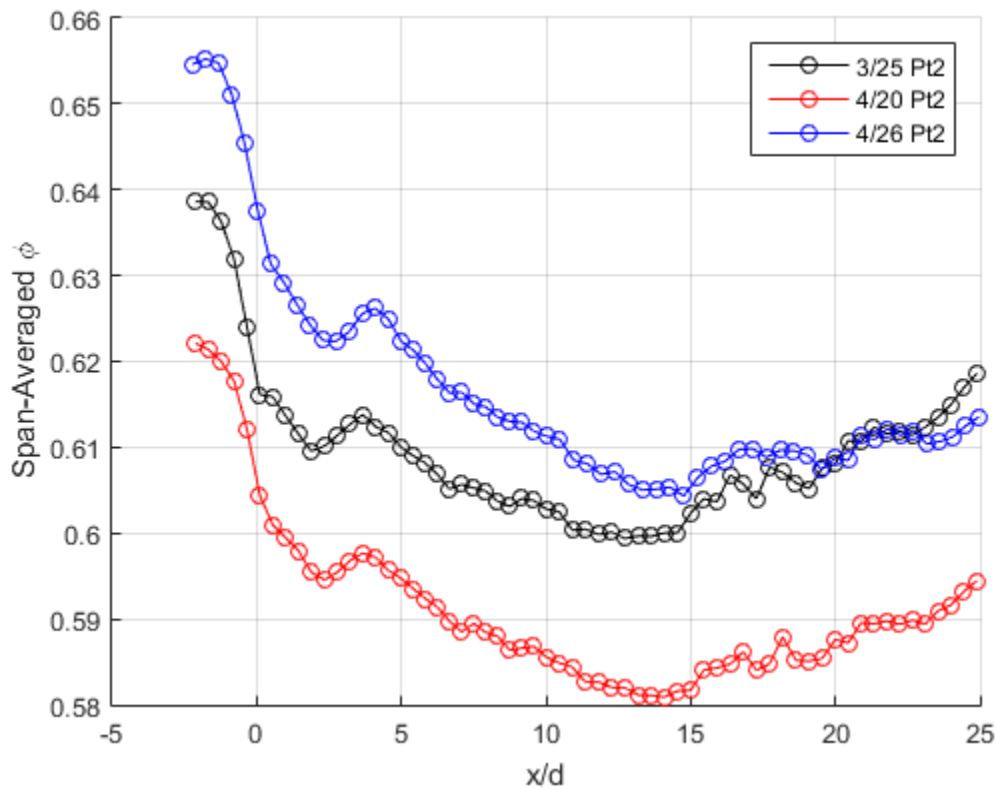


Figure 3-50: Day-to-day near ambient ϕ variability – final daily points

An assessment of variation within a single day of near ambient testing is presented in Figure 3-51. The greatest values of $\Delta\bar{\phi} = 0.01$ indicate strong agreement between the two repeatability runs collected on any given day. In fact, the data in Figure 3-51 indicate agreement between daily repeatability runs within $\Delta\bar{\phi} = \pm 0.006$ over a vast majority of the x/d range considered. More striking is the difference in $\Delta\bar{\phi}$ profile shapes for the various days. A relatively constant $\Delta\bar{\phi}$ value for data from 4/20 makes a strong argument for uniformity at all axial positions. Axial variations in $\Delta\bar{\phi}$ indicate lack of uniformity between the test points at various positions along the test plate. The cause of such a variation is unknown.

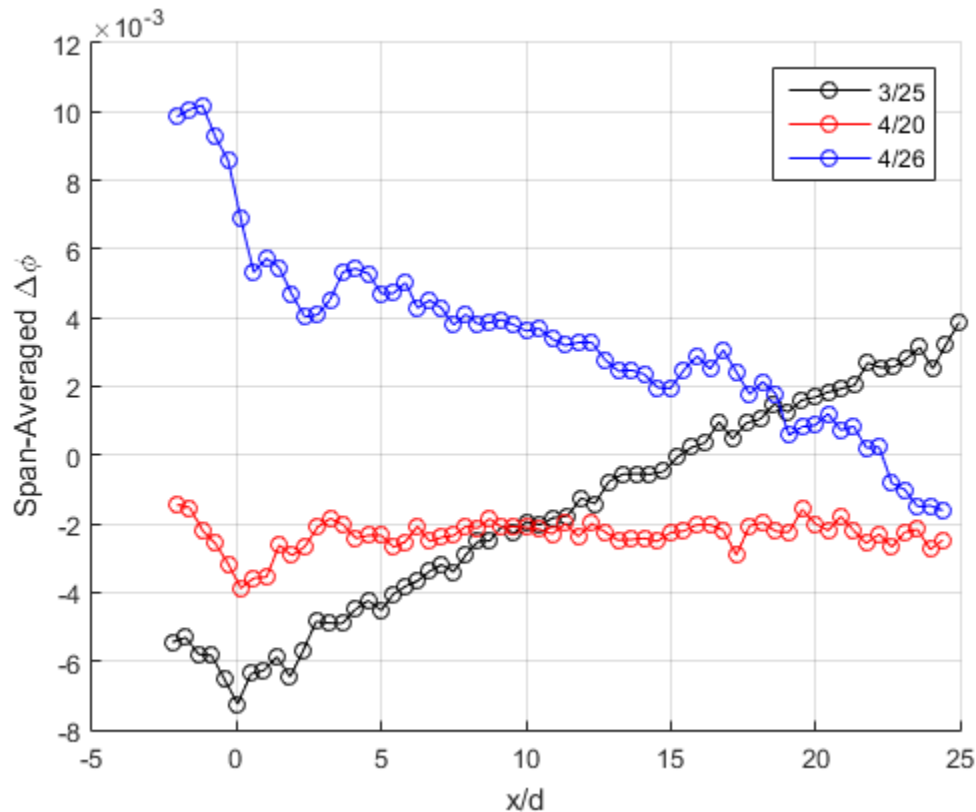


Figure 3-51: Single day near ambient ϕ variability

The data considered for near ambient repeatability indicate a strong capacity for replication of results on a given day. However, the data also demonstrate that collecting data over the span of several days may yield unintended variations that could at worst compromise

and at best complicate comparisons. No trend in overall effectiveness with respect to time is observed, but the relatively small data set collected warrants further analysis to better characterize FCR variations in time. Overall, none of the variations in $\bar{\phi}$ at near ambient conditions yielded values great enough to conclude a significant difference in results.

Overall, the repeatability analysis indicates acceptable capability for result replication over time. Greater variations in daily $\bar{\phi}$ values at high temperature conditions indicates a need for more careful experimental control and execution while large day-to-day $\bar{\phi}$ variations at near ambient conditions may warrant a change in experiment design. With these conclusions in mind, it is recommended that all conditions required for comparisons be collected on a single day within a single FCR run. Extra care should be taken during high temperature testing to ensure uniformity in freestream conditions and steady temperature values are achieved.

4. Results and Discussion

This work focused primarily on evaluating an approach for scaling film cooling performance from near ambient to engine representative temperatures. The steps necessary to complete this assessment are reflected in the primary research objectives: investigation of physical model size, comparison of overall effectiveness distributions measured at matched engine representative and near ambient conditions, and evaluation of effectiveness predictions. The achieved progress of comparing overall effectiveness values from separate temperature regimes and developing PSP and thermal near ambient experimental capabilities represent steps necessary in the build up to accomplish the larger research objectives. The following discussion presents a summary of experimental conditions; analysis of trends within both the engine representative and near ambient operating regimes; comparison of overall effectiveness values obtained at engine representative and near ambient conditions; and analysis of preliminary PSP results. The analysis of engine representative, near ambient, and PSP results directly addresses the technical challenge of ensuring adequate data quality before drawing tackling the primary research objectives.

4.1. Experimental Conditions

A summary of measured experimental conditions is displayed in Table 4-1, Table 4-2, and Table 4-3. The first column in each table enables quick reference for a given experimental point. Experimental points are sorted by blowing ratio with heavier horizontal lines grouping points by blowing ratio. The extra coolant column displays the amount of coolant supplied in excess of that required for the given blowing ratio normalized by the film coolant flow for a row of 16 cooling holes at the given film coolant blowing condition (see Section 3.6.1). Dark

horizontal lines provide groupings of points by blowing ratio. Freestream Reynolds numbers are reported using the coolant hole diameter, d , as the length scale ($Re_{\infty,d}$). Multiplying these Re_{∞} values by the ratio of leading edge cylinder diameter to coolant hole diameter, $D/d = 18.7$, yields Re_{∞} based on the leading edge cylinder diameter, $Re_{\infty,D}$. Assuming an average $Re_{\infty,d}$ value of 210, nominal values for $Re_{\infty,D}$ are approximately 4000.

4.2. Engine Representative Conditions

The process of accomplishing the objective of matching engine representative and near ambient film cooling performance began with data collection at engine representative conditions. Data collection at engine representative conditions entails challenges requiring specialized equipment and methods. The AFIT FCR provides capabilities enabling this process, but the data obtained must still be critically analyzed to ensure performance trends adhere to expectations based on published literature. This section examines data collected at engine representative conditions. The cooling scheme utilized was the single row of fan-shaped, laidback cooling holes described in Section 3.1.3. Results are analyzed first as a single baseline case, then with comparisons of blowing ratio and internal coolant flow rate.

4.2.1. Baseline Engine Representative Result

A single result obtained at engine representative conditions was selected for detailed analysis. This analysis is intended to evaluate significant features in the test plate surface temperature distribution, understand the phenomena driving those features, ensure adequate span-wise cooling effect uniformity, and ensure observable cooling effect. All these facets relate directly to the technical challenge of ensuring acceptable data quality.

Table 4-1: Air coolant high temperature experimental conditions

#	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_\infty$ (10^{-3} kg/min)	$\dot{m}_{c,in}$ (10^{-3} kg/min)	$\dot{m}_{c,out}$ (10^{-3} kg/min)	Extra Coolant (rows)	M (-)	I (-)	ACR (-)	Re_c (-)	Re_∞ (-)
1	982	1501	1.56	779	5.40	4.54	2	0.52	0.17	0.43	146	204
2	918	1503	1.67	779	10.82	9.78	3.5	0.62	0.23	0.51	181	204
3	903	1471	1.66	780	10.82	9.15	2	0.99	0.59	0.82	296	207
4	857	1471	1.75	811	10.82	9.06	2	1.01	0.58	0.82	322	216
5	726	1472	2.07	811	24.27	22.51	5	1.01	0.49	0.80	357	215
6	839	1502	1.83	788	19.88	18.12	4	1.04	0.59	0.84	326	206
7	834	1467	1.79	786	24.27	22.51	5	1.04	0.60	0.85	327	209
8	878	1504	1.75	780	15.20	13.45	3	1.04	0.62	0.85	315	204
9	916	1469	1.64	784	10.82	9.06	2	1.04	0.66	0.86	309	209
10	774	1464	1.93	784	24.27	22.51	5	1.04	0.56	0.84	343	209
11	910	1470	1.65	782	10.82	9.06	2	1.04	0.66	0.87	310	208
12	840	1479	1.80	779	24.27	22.51	5	1.05	0.61	0.86	326	206
13	911	1471	1.65	778	10.82	9.06	2	1.05	0.67	0.87	310	207
14	917	1498	1.67	777	10.82	9.06	2	1.05	0.66	0.87	309	204
15	901	1464	1.66	777	10.82	9.06	2	1.05	0.67	0.87	312	207
16	1010	1502	1.52	776	1.76	0.00	0	1.05	0.73	0.88	291	203
17	975	1499	1.57	776	6.14	4.38	1	1.05	0.71	0.88	297	204
18	916	1505	1.68	773	10.82	9.06	2	1.06	0.67	0.87	309	202
19	918	1490	1.66	770	10.82	9.06	2	1.06	0.68	0.88	308	203
20	910	1481	1.66	770	10.82	9.06	2	1.06	0.68	0.88	310	204
21	909	1471	1.65	770	10.82	9.03	2	1.08	0.70	0.89	315	205
22	831	1470	1.81	780	24.27	22.42	5	1.10	0.67	0.90	345	207
23	834	1469	1.80	779	24.27	22.33	5	1.16	0.74	0.94	361	207
24	868	1505	1.77	774	16.22	14.16	2.6	1.23	0.86	1.01	374	203
25	868	1504	1.77	786	16.22	13.60	2	1.55	1.36	1.27	477	206
26	831	1496	1.84	788	21.63	19.00	2.7	1.55	1.32	1.26	492	207
27	1004	1503	1.53	778	2.64	0.00	0	1.57	1.62	1.31	438	204
28	829	1476	1.82	785	27.04	23.53	2.5	2.07	2.37	1.69	655	208
29	837	1500	1.83	786	21.63	18.12	2	2.08	2.37	1.69	653	206
30	997	1504	1.54	781	3.50	0.00	0	2.08	2.82	1.74	584	205
31	832	1474	1.81	778	27.04	22.66	2	2.62	3.79	2.13	817	206
32	993	1504	1.55	773	4.38	0.00	0	2.63	4.48	2.20	732	203
33	813	1476	1.85	787	32.45	25.85	1.5	3.90	8.21	3.16	1251	209

Table 4-2: Argon coolant near ambient experimental conditions

#	Gas	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_\infty$ (10^{-3} kg/min)	$\dot{m}_{c,in}$ (10^{-3} kg/min)	$\dot{m}_{c,out}$ (10^{-3} kg/min)	M (-)	I (-)	ACR (-)	Re_c (-)	Re_∞ (-)
1	Ar	358	395	1.52	441	2.82	2.37	0.47	0.14	0.24	118	291
2	Ar	357	394	1.52	449	3.12	2.62	0.52	0.18	0.27	133	297
3	Ar	336	364	1.49	283	4.06	3.74	0.53	0.19	0.27	90	198
4	Ar	336	363	1.49	292	4.06	3.71	0.57	0.22	0.29	100	204
5	Ar	357	393	1.52	451	3.42	2.87	0.57	0.21	0.29	147	298
6	Ar	336	363	1.49	288	4.06	3.69	0.60	0.25	0.31	105	202
7	Ar	352	388	1.52	442	0.96	0.00	1.01	0.67	0.52	259	296
8	Ar	357	393	1.52	440	3.40	2.44	1.02	0.68	0.52	256	291
9	Ar	336	363	1.49	288	0.66	0.00	1.06	0.76	0.55	184	202
10	Ar	356	393	1.52	447	3.58	2.55	1.08	0.76	0.55	276	295
11	Ar	336	363	1.49	283	2.30	1.64	1.08	0.79	0.56	184	198
12	Ar	337	363	1.49	298	0.70	0.00	1.08	0.79	0.56	194	209
13	Ar	353	389	1.52	433	1.02	0.00	1.09	0.79	0.56	273	288
14	Ar	356	393	1.52	451	3.74	2.67	1.10	0.80	0.57	285	298
15	Ar	336	363	1.49	288	2.42	1.73	1.12	0.84	0.58	194	202
16	Ar	336	363	1.49	302	2.53	1.80	1.13	0.85	0.58	204	212
17	Ar	354	389	1.51	441	1.07	0.00	1.13	0.84	0.58	287	294
18	Ar	337	363	1.49	291	0.73	0.00	1.17	0.92	0.60	204	204
19	Ar	354	390	1.52	441	1.46	0.00	1.54	1.57	0.79	391	293
20	Ar	337	363	1.49	293	1.00	0.00	1.59	1.69	0.82	278	205
21	Ar	355	390	1.52	443	1.52	0.00	1.59	1.66	0.82	406	295
22	Ar	337	363	1.49	294	1.03	0.00	1.64	1.80	0.84	288	206
23	Ar	355	390	1.52	441	1.57	0.00	1.65	1.80	0.85	419	293
24	Ar	337	364	1.49	293	1.07	0.00	1.70	1.94	0.88	298	205
25	Ar	356	395	1.53	460	1.98	0.00	2.00	2.61	1.03	528	304
26	Ar	336	365	1.50	294	1.30	0.00	2.06	2.82	1.06	364	205
27	Ar	357	395	1.53	456	2.03	0.00	2.07	2.81	1.06	540	301
28	Ar	337	363	1.49	290	1.34	0.00	2.14	3.09	1.10	372	203
29	Ar	336	365	1.50	289	1.34	0.00	2.15	3.08	1.11	374	202
30	Ar	358	395	1.52	447	2.07	0.00	2.15	3.04	1.11	549	295
31	Ar	337	363	1.49	291	1.37	0.00	2.19	3.24	1.13	382	204
32	Ar	337	364	1.49	289	1.41	0.00	2.26	3.44	1.17	392	202
33	Ar	337	364	1.49	299	1.69	0.00	2.63	4.65	1.36	472	209
34	Ar	355	390	1.52	446	2.53	0.00	2.64	4.59	1.36	678	296
35	Ar	355	390	1.52	436	2.50	0.00	2.66	4.67	1.37	668	290
36	Ar	355	390	1.52	441	2.58	0.00	2.72	4.89	1.40	692	293
37	Ar	337	364	1.49	295	1.76	0.00	2.78	5.17	1.43	492	207
38	Ar	337	364	1.49	286	1.73	0.00	2.80	5.28	1.45	482	200

Table 4-3: Carbon dioxide coolant near ambient experimental conditions

#	Gas	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_\infty$ (10^{-3} kg/min)	$\dot{m}_{c,in}$ (10^{-3} kg/min)	$\dot{m}_{c,out}$ (10^{-3} kg/min)	M (-)	I (-)	ACR (-)	Re_c (-)	Re_∞ (-)
39	CO2	336	370	1.67	350	4.67	4.24	0.57	0.20	0.50	179	241
40	CO2	332	363	1.66	301	4.08	3.71	0.58	0.20	0.50	157	211
41	CO2	333	365	1.67	295	3.95	3.57	0.59	0.21	0.51	156	206
42	CO2	337	371	1.67	346	4.67	4.20	0.63	0.24	0.56	196	239
43	CO2	333	365	1.67	294	3.95	3.53	0.65	0.26	0.57	173	205
44	CO2	333	363	1.66	297	4.08	3.65	0.68	0.28	0.59	181	208
45	CO2	337	371	1.68	340	4.67	4.16	0.70	0.29	0.61	212	235
46	CO2	333	363	1.66	295	4.08	3.61	0.74	0.33	0.65	198	207
47	CO2	333	363	1.66	287	4.08	3.57	0.83	0.41	0.72	214	201
48	CO2	332	364	1.66	295	3.89	3.26	0.99	0.59	0.86	264	207
49	CO2	333	363	1.65	304	4.08	3.42	1.02	0.63	0.89	280	213
50	CO2	334	370	1.68	345	4.67	3.91	1.03	0.63	0.90	321	238
51	CO2	332	363	1.66	299	4.08	3.42	1.04	0.65	0.91	281	210
52	CO2	333	364	1.66	296	4.08	3.42	1.05	0.66	0.91	280	208
53	CO2	335	370	1.67	347	4.89	4.10	1.05	0.66	0.92	328	240
54	CO2	333	364	1.66	287	4.08	3.42	1.08	0.70	0.94	280	201
55	CO2	329	361	1.67	287	4.08	3.42	1.08	0.70	0.94	283	202
56	CO2	332	363	1.66	285	4.08	3.42	1.09	0.71	0.95	281	200
57	CO2	338	364	1.64	283	4.08	3.42	1.09	0.73	0.96	276	198
58	CO2	335	369	1.67	346	5.12	4.30	1.11	0.74	0.97	344	239
59	CO2	332	363	1.66	285	4.28	3.59	1.12	0.76	0.98	288	200

Surface temperature contours for a sample condition (point # 14 in Table 4-1) are shown in Figure 4-1. This point represents the high temperature repeatability condition, and the experimental conditions were $M = 1.06$, $DR = 1.66$, and $I = 0.68$ with air as the coolant gas and two rows of excess coolant flow. This definition for excess coolant flow would supply two rows of 16 holes each with enough coolant to ensure the specified blowing ratio from all film cooling holes. Further details on experimental conditions are shown in Table 4-1 and may be referenced in Row 14. This point is chosen as a representative example of high temperature conditions for the purpose of assessing overall data quality for the high temperature measurements.

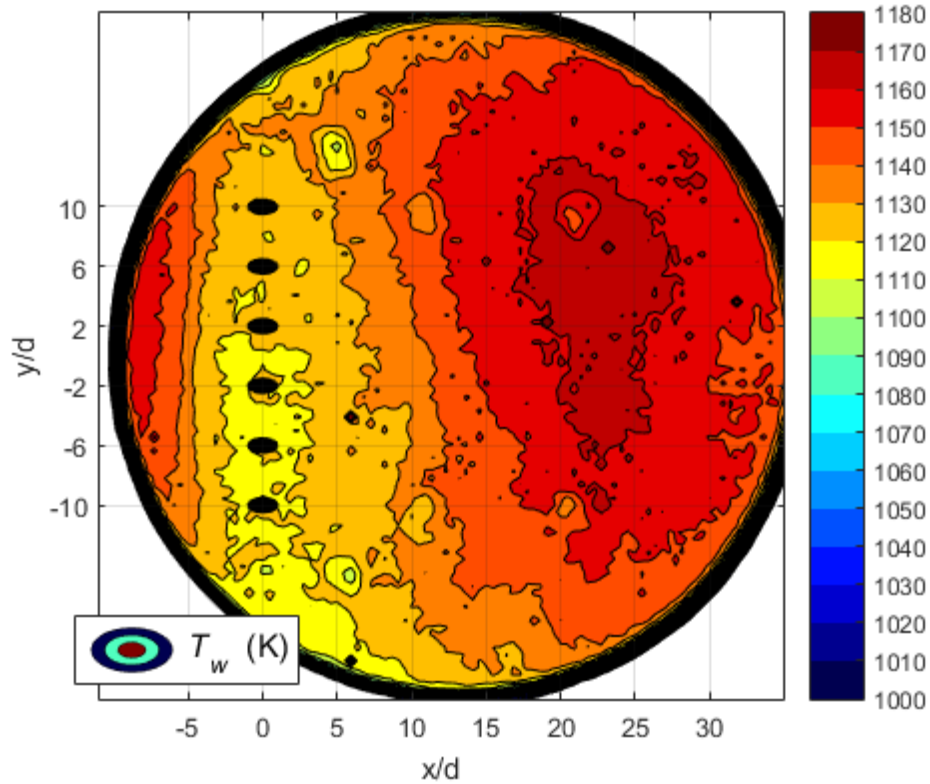


Figure 4-1: Sample high temperature condition (#14) surface temperature contours

Several notable features and trends are apparent in Figure 4-1. As expected, there is a region of decreased temperature immediately downstream of the single row of six cooling holes indicating film cooling effect. In the near-hole region ($x/d < 15$), surface temperature increases with increased x/d , which is consistent with increasing film temperature as the coolant mixes with freestream gas. Also, the region of high surface temperatures at $x/d \leq -5$ is consistent with freestream gas stagnating on the leading edge of the test plate. Elevated heat transfer into the test plate is expected in this region compared to the flat after body due to an increase in heat transfer coefficient as the freestream flow impinges on the test plate leading edge. To better reference these x/d locations with the physical hardware, a cross-section schematic is presented in Figure 4-2.

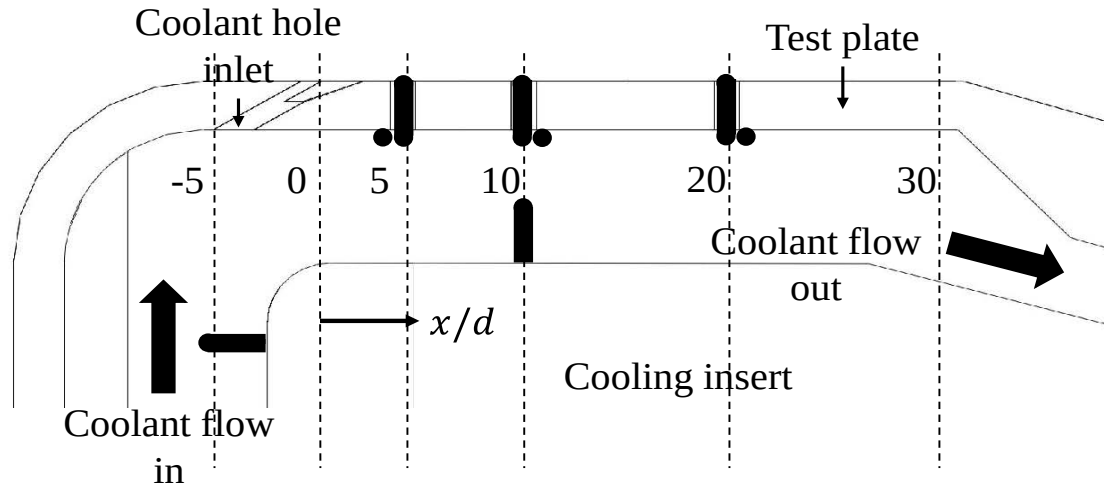


Figure 4-2: Test plate cross-section schematic with x/d positions marked

However, the high point in temperature centered at $x/d = 22$ and $y/d = 7$ followed by decreasing temperatures downstream reveals a trend counter to the expectation for increasing surface temperature as axial position increases. Decreasing freestream temperature with axial position would reduce the driving potential for heat transfer into the test plate and could explain the downstream decrease in surface temperature. However, the gradient in T_∞ with axial distance would need to be significant for test plate surface temperature to vary so noticeably over such a small distance. Based on a freestream temperature of 1498 K and an FCR exit temperature of 1199 K with a distance of 140 mm between measurement stations, the average temperature gradient through the FCR is 2.14 K/mm for this condition. Using this temperature gradient and the distance of 25.4 mm between T_∞ measurement and $x/d = 22$, the potential decrease in T_∞ is 54 K. A temperature decrease of this magnitude could conceivably result in the downstream T_w decrease observed. Further T_∞ measurements should be made to better resolve this temperature gradient and assess the impact of decreased flow area on static temperature in the FCR test section.

Alternatively, a combination of aerodynamic and heat transfer effects resulting from the test plate transition to the sloped afterbody at $x/d = 30$ and the accompanying increase in test plate thickness could cause a decrease in T_w . As the test plate thickness increases, heat flux should decrease, resulting in lower surface temperatures as observed. Also, the internal channel width decreases with this thickness increase. Because the coolant temperature is not known at this axial position within the channel, a precise evaluation of coolant velocity, and thus heat transfer coefficient, cannot be made. A region of flow separation may exist at the transition to the sloped afterbody, potentially drawing hot freestream gas down to the surface. However, contrary to the results observed, this would result in elevated T_w values. Further, no data has been collected to observe such a phenomenon.

However, the change in internal flow direction caused by the sloped afterbody would result in a region of indirect impingement cooling on the internal test plate surface near $x/d = 30$ (see Figure 4-2). Figure 4-3 displays a schematic highlighting the probable internal coolant impingement regions on the test plate. This region would act as a heat sink, drawing energy away from the surrounding material. With a highly conductive material like Hastelloy X, multi-dimensional conduction effects could potentially be significant enough to cause the observed decrease in T_w beginning at $x/d = 22$. A combination of effects from increased wall thickness, a streamwise decrease in T_∞ , and internal coolant impingement could cause the axial decrease in T_w observed for $x/d > 25$.

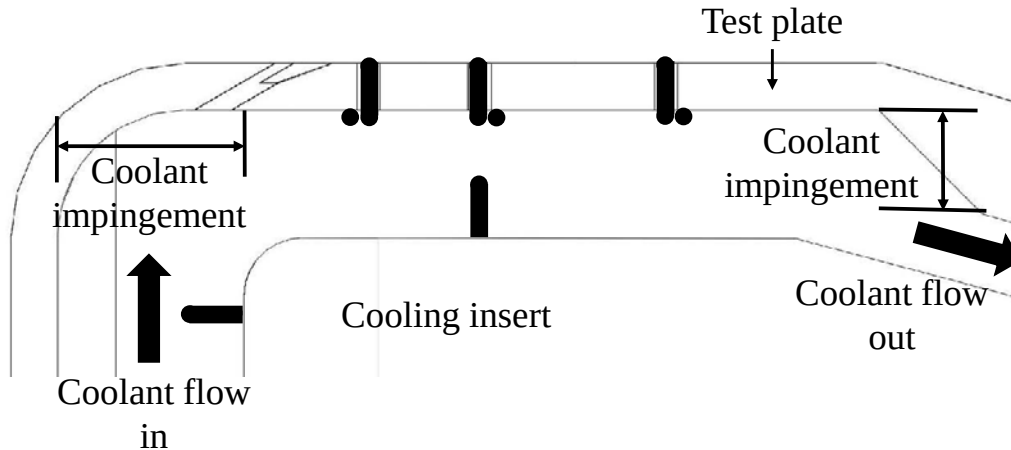


Figure 4-3: Probable test plate internal coolant impingement regions

Other noteworthy features present in Figure 4-1 include several small regions of lower temperature present at the thermocouple locations, lower temperatures upstream of and between the cooling holes than immediately downstream of the cooling holes, and lower temperatures for $y/d < 0$ as opposed to $y/d > 0$. The data indicate the thermocouple weld beads emitted less radiation than the rest of the surface, which may result from a difference in either emissivity or temperature. Because the thermocouples behaved differently, the locations used for calibration of the IR camera were not directly on the thermocouples. Further discussion of this issue and the IR calibration process can be found in Section 3.3.1.1. Non-uniform sidewall boundary conditions, swirl in the freestream flow, or leakage around the coolant baffle plate described in Section 3.4.1 could cause heat transfer variations across the test plate resulting in the skewed temperature distribution of Figure 4-1. In fact, the trend of increasing T_w with increasing y/d in the region $-5 < x/d < 0$ is consistent with internal coolant flow from $y/d < 0$ to $y/d > 0$ and gathering heat as it goes.

After impinging on the internal test plate surface near $x/d = -5$ (see Figure 4-3), the coolant flow turns and flows downstream toward the region of indirect impingement and cross-sectional area decrease near $x/d = 30$ and the coolant exhaust. This process of direct

impingement near $x/d = -5$ would result in very high turbulence levels and would disrupt boundary layer formation, resulting in two independent boundary layers forming on the internal test plate surface and the upper cooling insert surface. The combination of developing flow and very high turbulence levels should result in very high Nu values within the internal channel. Additionally, the mixing generated by the upstream impingement region may result in a recirculation region, increasing coolant dwell time and raising the coolant temperature considerably before entering the coolant hole inlets. This could result in an underestimate of T_c . Further, as the coolant travels downstream within the internal channel, the gas temperature continues to rise 20-30 K, reducing cooling potential at downstream locations.

Decreased temperatures upstream of and between the cooling holes must result from internal cooling effect because no film coolant was present at these locations. Low temperatures in the $-5 < x/d < 0$ region most likely resulted from increased internal heat transfer due to internal coolant impingement as depicted in Figure 4-3. The lower temperatures between cooling holes could result from a combination of increased heat transfer along the cooling hole passages and coolant impingement on the internal test plate surface, thus reducing the surface temperature between the holes. The fact that the lowest surface temperatures in Figure 4-1 reside outside of film cooled areas indicates that internal cooling makes the strongest contribution to reducing test plate surface temperatures.

All features found in Figure 4-1 are also present in the overall effectiveness contours of Figure 4-4. This is as expected because ϕ is calculated using fixed values for T_c and T_∞ , meaning that any variation in ϕ stems from a direct variation in T_w . Thus, ϕ enables comparison between test cases at different temperatures but provides no added value over that obtained from T_w data for feature identification with a single case. However, T_c does vary spatially within the

internal channel. The internal coolant gained about 30 K between the upstream and downstream internal channel thermocouple locations. Also, the upstream internal channel thermocouple located at $y/d = 14$ indicated a temperature about 3 K hotter than its counterpart at $y/d = -14$. These T_c trends are manifested in the ϕ contours of Figure 4-4 as increased ϕ for $y/d < 0$ and decreasing ϕ with increasing x/d . However, if ϕ were calculated using a spatially variable T_c value, the downstream ϕ values in Figure 4-4 would be greater than shown. Further, the hypothesized coolant recirculation region near $x/d = -5$ would cause an overestimate of T_c , causing inflated ϕ values.

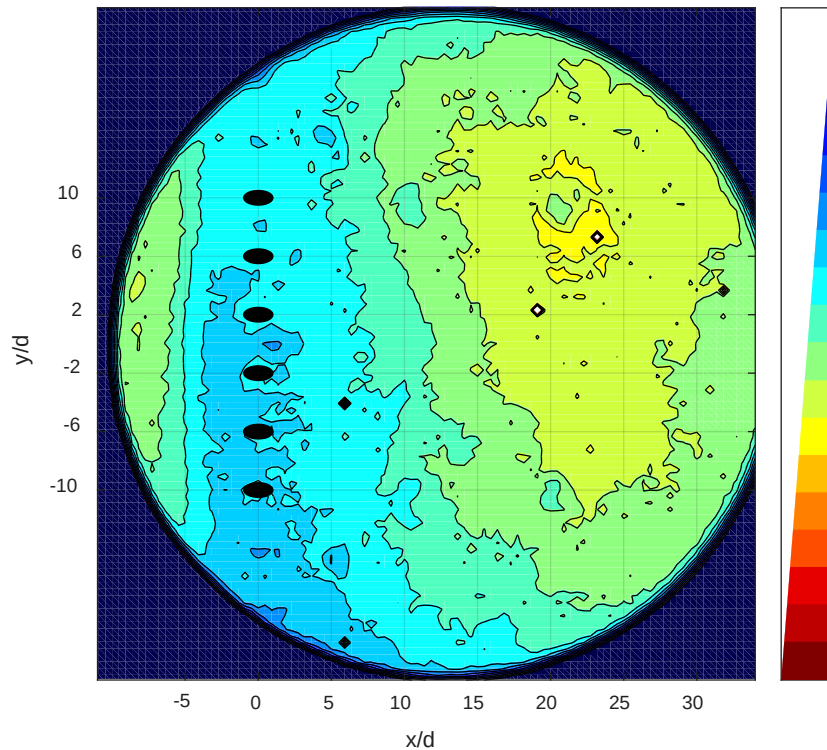


Figure 4-4: Sample high temperature condition (#14) overall effectiveness contours

4.2.2. Blowing Ratio Sweeps

Blowing ratio sweeps were conducted at high temperature test conditions with various internal coolant flow rates. The sweeps included cooling schemes that were plenum-fed,

incorporated varied excess coolant flow based on the blowing ratio examined, and held internal coolant flow approximately constant.

4.2.2.1. Plenum-Fed Cases

The sweep with the most consistent test conditions across the varied blowing ratios was the plenum-fed case. A listing of experimental conditions for comparison is shown in Table 4-4. The points correspond to Points 16, 27, 30, and 32 in Table 4-1 and feature four different blowing ratios: $M = 1.05, 1.57, 2.08, \text{ and } 2.63$. Density ratio was held fairly constant near 1.52 with an average coolant temperature near 1000 K and average freestream temperature of approximately 1503 K.

Table 4-4: Plenum-fed blowing ratio sweep high temperature experimental conditions

#	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_{c,in}$ (10^{-3} kg/min)	$\dot{m}_{c,out}$ (10^{-3} kg/min)	M (-)	I (-)	ACR (-)	Re_c (-)	Re_∞ (-)	Re_{c,D_h} (-)
16	1010	1502	1.52	1.76	0.00	1.05	0.73	0.88	291	203	33
27	1004	1503	1.53	2.64	0.00	1.57	1.62	1.31	438	204	50
30	997	1504	1.54	3.50	0.00	2.08	2.82	1.74	584	205	67
32	993	1504	1.55	4.38	0.00	2.63	4.48	2.20	732	203	84

Figure 4-5 displays overall effectiveness contours for the conditions listed in Table 4-4. Overall ϕ magnitudes are fairly well matched between all the cases, only differing by approximately 0.04. Although effectiveness in the near-hole region ($x/d < 10$) appears to decrease with increased blowing ratio, downstream ($x/d > 15$) effectiveness values increase with increased blowing ratio.

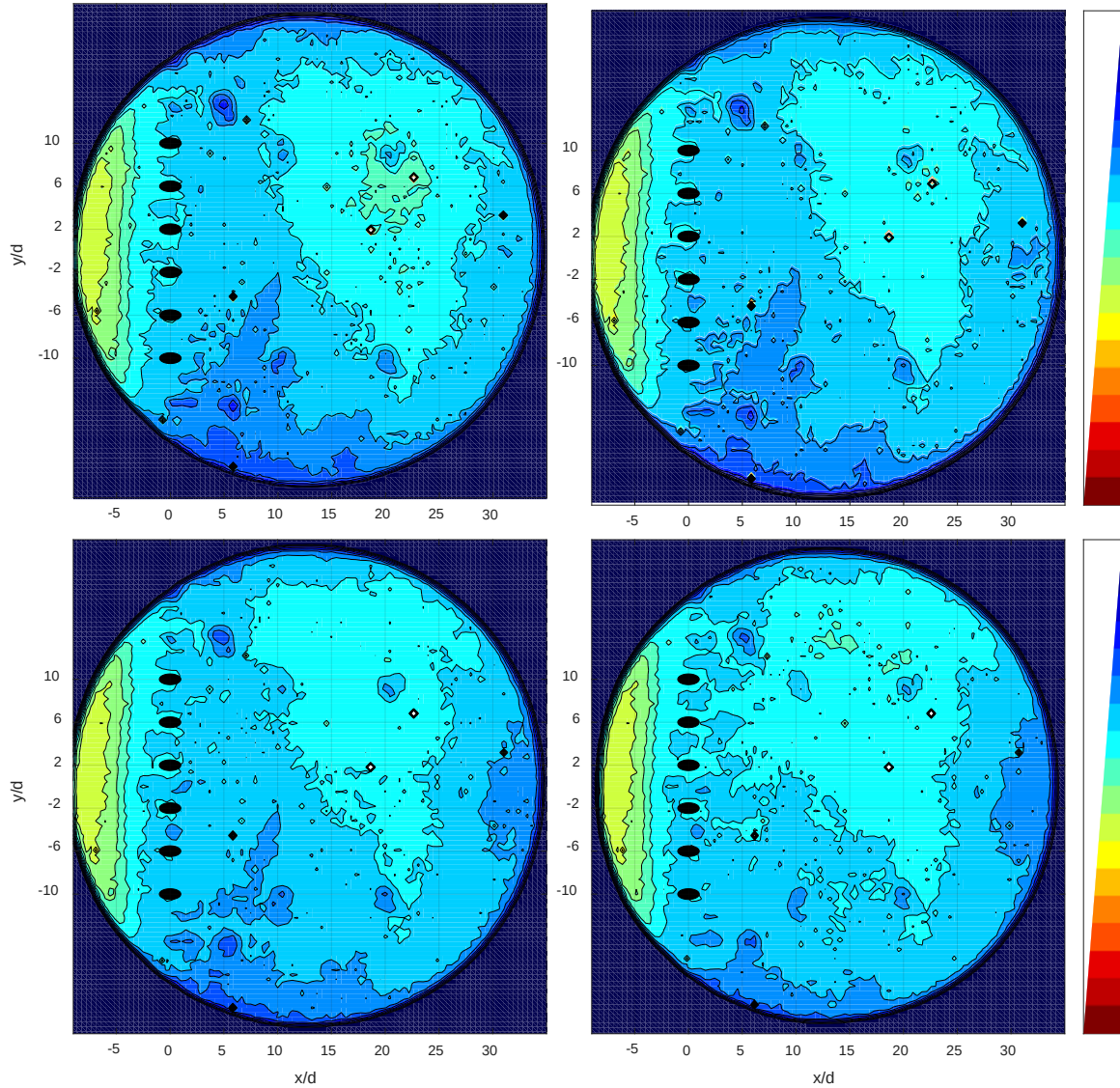


Figure 4-5: High temperature overall effectiveness contours with plenum-fed cooling: $M = 1.05$ (upper left), $M = 1.57$ (upper right), $M = 2.08$ (lower left), $M = 2.63$ (lower right)

Notable features in Figure 4-5 include the presence of an “arm” of increased effectiveness clearly visible between $5 < x/d < 10$ for $y/d < -2$ in the $M = 1.05$ and 1.57 cases with some trace present for the $M = 2.08$ case. This “arm” suggests that the internal coolant bypassed the coolant baffle plate intended to produce uniform internal flow and established a reverse flow from the origin of the “arm” to the entrance of the cooling holes. While this cannot be proven, the skewed distributions shown in Figure 4-5 combined with a logical assessment of

the flow paths available to the internal coolant flow (see Section 3.4.1) make it likely that there is at least some reverse flow within the internal channel for plenum fed cases. This skew in cooling effect towards $y/d > 0$ is also present for other cases, including the repeatability condition presented above in Figure 4-4. For this reason, it is hypothesized that more coolant flows beneath the test plate for $y/d < 0$ than $y/d > 0$. This may be caused by the additional flow paths described in Section 3.4.1. It is further noted that the repeatability condition features two rows of excess internal coolant flow at $M = 1.05$, indicating that this phenomenon should be present regardless of the internal coolant flow rate.

Several small dots are apparent on all the contour plots at $(x/d, y/d)$ coordinates (6, -4.5), (18, 2), (22.5, 6.5), and (31, 3). These dots are locations with outlying ϕ values well below neighboring points and shall be considered non-physical. The root cause of these aberrant values is unknown, although they could arise from bad pixels in the IR camera, blemishes on the IR window, or some form of debris with higher emissivity than the test plate surface. Bad pixels are highly unlikely, as not all measurements feature outlying values, and the outliers change location within the viewing frame even when the camera is not moved.

The trends described for Figure 4-5 become more apparent when considering span-averaged effectiveness values. Figure 4-6 displays ϕ values averaged over the span of the two middle cooling holes. A region of reduced effectiveness over $0 < x/d < 10$ for the $M = 2.63$ case points to coolant jet liftoff while the increase in $\bar{\phi}$ at $x/d = 10$ indicates jet reattachment. Figure 4-6 clearly illustrates that $\bar{\phi}$ for all four blowing ratios is approximately equal leading up to the end of the coolant holes. The film cooling effect is clearly visible from $0 < x/d < 10$, at which point $\bar{\phi}$ decays. Based on the magnitude of $\bar{\phi}$ increase downstream of the cooling holes and the relatively flat profiles shown in Figure 4-6, it is clear that the film cooling provides little cooling

effect. The bulk of the cooling effect is driven by internal cooling, consistent with the discussion in Section 4.2.1. It is also worth noting that the spread in $\overline{\phi}$ values is 0.015 at most. This does not exceed the uncertainty value ($\delta\phi = 0.032$) calculated in Section 3.8.2.1, meaning these results cannot conclusively be said to differ.

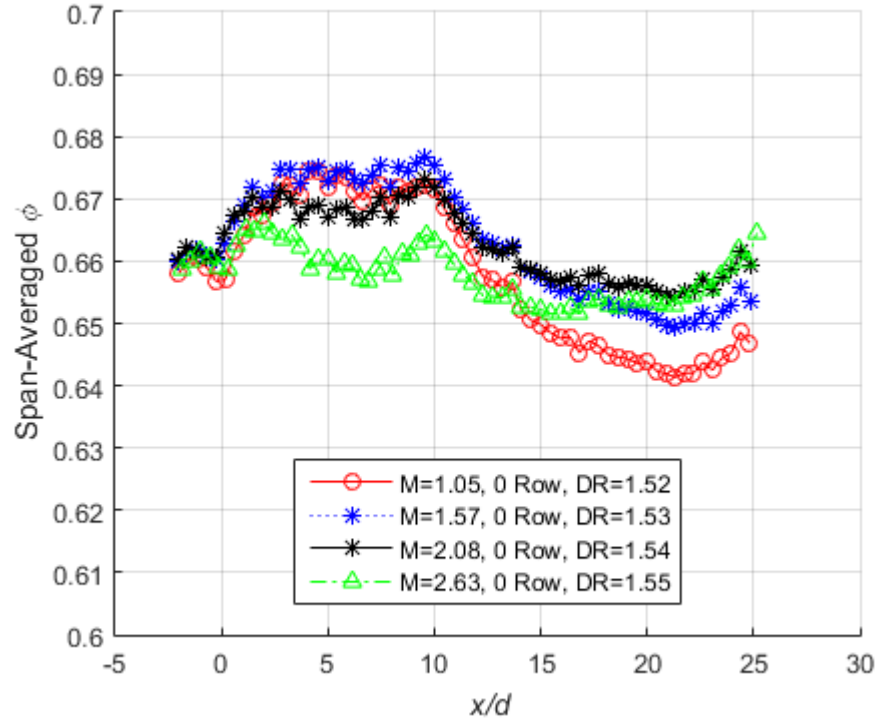


Figure 4-6: High temperature plenum-fed cooling span-averaged overall effectiveness

Although the higher blowing ratio cases of Figure 4-6 appear to feature slightly lower $\overline{\phi}$ values, Figure 4-7 shows that their $\overline{T_w}$ values are lower. Considering the maximum uncertainty in T_w due to the IR calibration process ($\Delta T_{w,max} = 10.3$ K), these $\overline{T_w}$ values can only be said to differ for $0 < x/d$ and $x/d > 15$. While this provides some insight on the effect of increasing internal coolant flow rate, the $\overline{T_w}$ values in the film cooled region of the test plate cannot be said to differ.

This reversal of expectation for $\overline{\phi}$ values must lie in the external heat transfer coefficient and adiabatic wall temperature values. As blowing ratio increased from $M = 1.05$ to $M = 2.63$,

the coolant mass flow rate increased and T_c decreased by approximately 20 K. In this case, increased M values would also lead to higher external and internal heat transfer coefficient values, both of which would independently increase heat flux. If h_f increased by a sufficient margin, it is possible that the benefits from reducing T_c and increasing $\dot{m}_c$ could be overshadowed. Alternatively, a smaller reduction in T_{aw} than T_c while maintaining a constant h_f value would also result in lower $\bar{\phi}$ values.

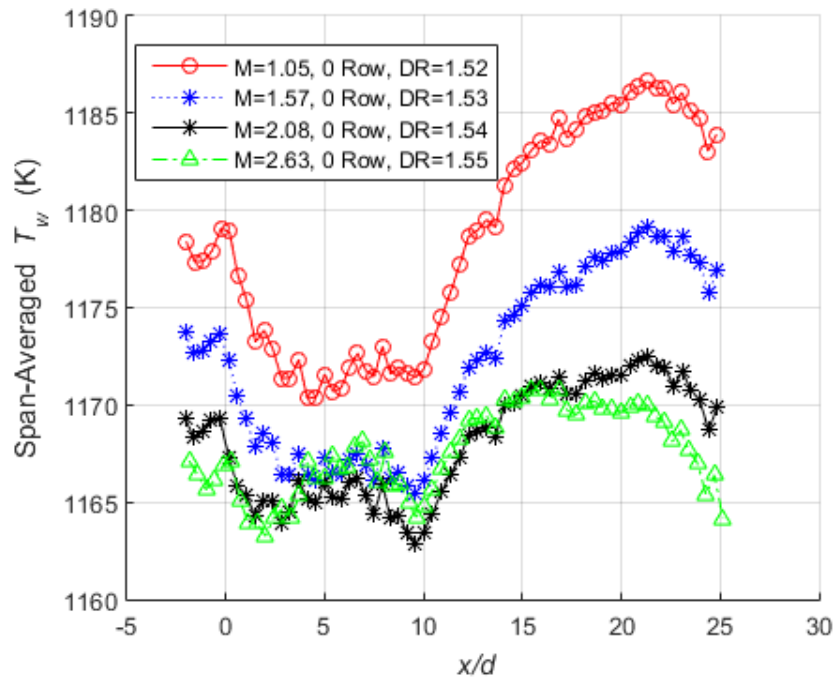


Figure 4-7: High temperature plenum-fed cooling span-averaged surface temperature

Making meaningful comparisons between plenum-fed cases at varied blowing ratios presented a significant challenge. Although the coolant flow rates are very small ($\sim 2\text{-}4$ g/min) and little impact on T_w is expected, the ~ 2 g/min increase in $\dot{m}_c$ from $M = 1.05$ to $M = 2.63$ generated a 20 K difference in T_c . This observed T_c dependence on $\dot{m}_c$ definitely impacts T_w but should not have any impact on ϕ values. However, the observed change in T_w (~ 10 K) is smaller than for T_c (20 K), indicating that ϕ should decrease with increasing M , consistent with the results of Figure 4-7. This unexpected result demands further investigation of T_c variations

with $\dot{m}_c$ as well as the impact of internal coolant flow rate on ϕ . Both of these challenging, complex issues are dealt with in the internal coolant flow rate investigation of Section 4.2.4.

4.2.2.2. Two Rows Excess Coolant

The effect of decreased effectiveness in spite of increased M or coolant flow rate is also apparent when comparing high temperature data points obtained at various blowing ratio values with a consistent two rows of excess coolant. Excess internal coolant flow was normalized by the film coolant flow rate for 16 coolant holes at the given blowing ratio to give excess flow as a number of rows. Although these T_c and DR variations are undesired and complicate making comparisons between points, they represent effects encountered when changing coolant mass flow for a real cooling scheme with a fixed inlet gas temperature. Thus the total coolant flow supplied for each point differed, resulting in varied T_c , DR , and h_i values. The array of variables under consideration means any conclusions drawn must be regarded as a combination of effects from M , T_c , DR , $\dot{m}_c$, and h_i variations. Table 4-5 shows conditions for a blowing ratio sweep with 2 rows of extra coolant supplied.

Table 4-5: Two rows excess coolant blowing ratio sweep high temperature experimental conditions

#	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_{c,in}$ (10^{-3} kg/min)	$\dot{m}_{c,out}$ (10^{-3} kg/min)	Extra Coolant (rows)	M (-)	I (-)	Re_c (-)	Re_∞ (-)	Re_{c,D_h} (-)
1	982	1501	1.56	5.40	4.54	2	0.52	0.17	146	204	105
18	916	1505	1.68	10.82	9.06	2	1.06	0.67	309	202	219
25	868	1504	1.77	16.22	13.60	2	1.55	1.36	477	206	339
29	837	1500	1.83	21.63	18.12	2	2.08	2.37	653	206	463
31	832	1474	1.81	27.04	22.66	2	2.62	3.79	817	206	581

Span averaged surface temperatures for the two rows excess coolant blowing ratio sweep are shown in Figure 4-8. Decreased surface temperatures for the high blowing ratio cases likely

result from a combination of increased coolant mass flow, decreased coolant temperature, and increased internal heat transfer coefficient. For these conditions, the coolant temperature decreases with increased blowing ratio from 982 K to near 835 K for the $M > 2$ cases. This 147 K change in T_c only generates a 95 K change in T_w , indicating that $\bar{\phi}$ values should decrease with increased blowing ratio. This is contrary to expectation and warrants further investigation.

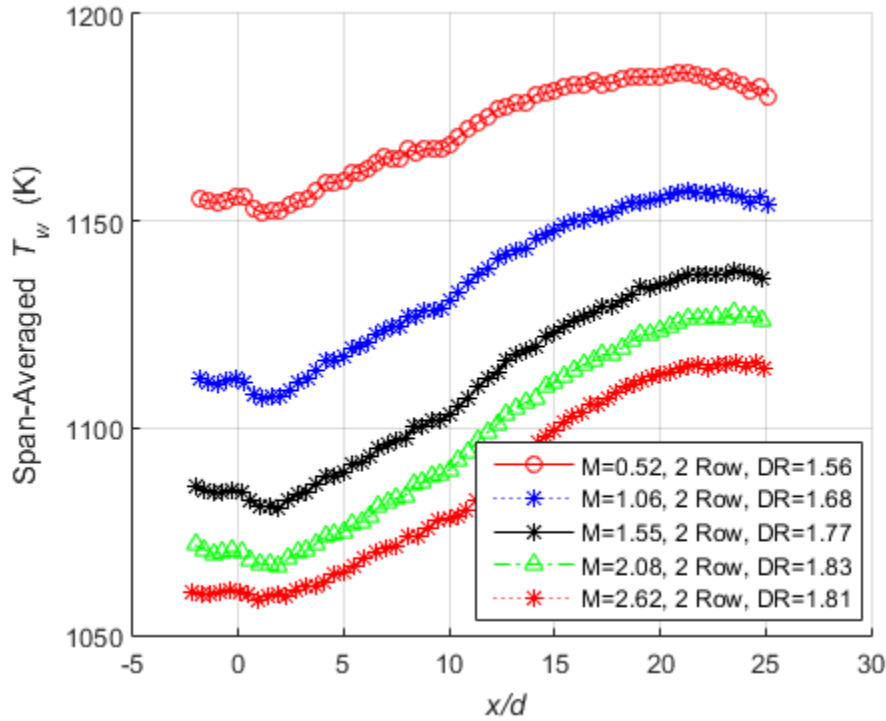


Figure 4-8: High temperature blowing ratio sweep, two rows excess coolant: span-averaged surface temperature

Because ϕ should increase with increased coolant flow, either the apparent change in T_c is too great or the change in T_w is too small. Based on the rigor of the IR calibration process, it seems more likely that the change in T_c values was less than as measured. This could result from a recirculation region immediately downstream of the internal coolant impingement region. Recirculation would increase coolant gas dwell time in a region of high heat transfer, causing T_c to increase between the point of measurement and the film cooling hole inlets or passing through the internal coolant channel. Additionally, the coolant baffle plate would serve to increase

turbulence in the internal coolant flow, increasing heat transfer into the coolant gas and raising T_c after the point of measurement. The discussion of alternate flow paths around the coolant baffle plate (Section 3.4.1) provides additional possibility for coolant temperature increase, as these alternate flow paths also serve to increase coolant dwell time. Further, freestream flow impingement on the external test plate surface provides a highly effective scheme for heat transfer into the coolant. Combining the effects of freestream impingement, internal recirculation, and increased turbulence from the coolant baffle plate, the coolant temperature could increase drastically between the point of measurement and the cooling hole inlets.

Alternatively, T_c measurements for low coolant flow rates could overestimate the actual gas temperature. This could arise from a failure to account for conductive and radiative heat transfer into the thermocouples used to measure T_c . While this would result in a T_c overestimate for all conditions, convective heat transfer from the thermocouples to the coolant gas would increase relative to conductive and radiative heat transfer into the thermocouples as the coolant flow rate rises, likely resulting in a larger T_c overestimate for low coolant flow rates. Further, the alternate coolant flow paths described in Section 3.4.1 could potentially accommodate a significant fraction of the total coolant flow for low coolant flow rates, resulting in further reduced convective heat transfer away from the coolant thermocouples and increasing the measured T_c value. Combined with the previous discussion on possible modes of increasing coolant temperature after the point of measurement, it is possible that if the magnitude of the T_c overestimate were great enough for low coolant flow rates, this could explain why T_c appears to decrease more than T_w .

The hypothesized error in T_c measurement may arise in part due to the type of thermocouple used. All test plate thermocouples were shielded K-Type thermocouples with an

Inconel sheath around the entire thermocouple body. This sheath would provide a conductive path, potentially skewing the measured T_c value. By utilizing an exposed bead thermocouple, the conductive path may be eliminated and a more accurate gas temperature obtained. While this comes with a reduction in strength and durability for the thermocouple, the potential gains in accuracy merit investigation in spite of potential for increased cost and maintenance.

Figure 4-9 displays span-averaged overall effectiveness for a blowing ratio sweep conducted for conditions with two rows of excess coolant flow (see Table 4-5). Because the internal coolant flow rate varies so widely for these conditions, an undesirably large range of DR values is present. A trend of decreasing effectiveness with increased blowing ratio is noted, although the $M = 1.06$ case does indicate slight improvement over $M = 0.52$ in the near-hole region ($x/d < 5$). Decreased $\bar{\phi}$ values with increased blowing ratio was a result of decreased surface temperature accompanied by a larger decrease in coolant temperature. The exact nature of this dynamic is not well understood, but it is consistent with the trend noted in the discussion of plenum-fed schemes in Section 4.2.2.1. Previous discussion also hypothesizes potential causes of this counterintuitive result. Further, the values of I shown in Table 4-5 indicate potential for coolant jet liftoff for the $M > 2$ cases. Coolant jet liftoff could also help explain the trend in $\bar{\phi}$ values present in Figure 4-9. Overall, the $\bar{\phi}$ profiles of Figure 4-9 indicate that combining effects from M , T_c , DR , $\dot{m}_c$, and h_i variations causes $\bar{\phi}$ to be approximately the same across the range of conditions considered. For each $\bar{\phi}$ variation caused by changing a single parameter, another parameter changing simultaneously opposes that $\bar{\phi}$ change, resulting in near constant $\bar{\phi}$ values.

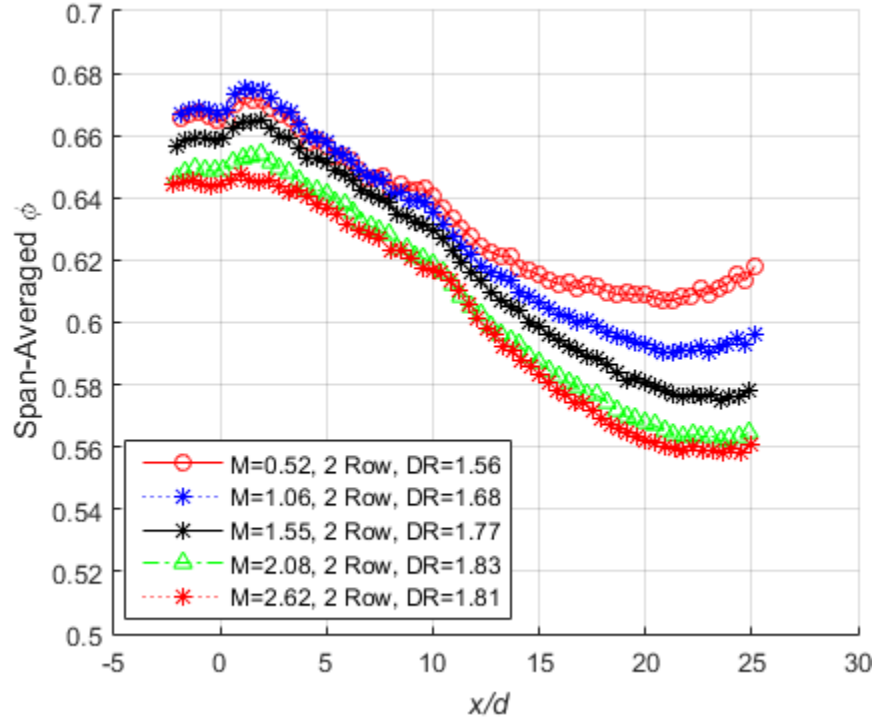


Figure 4-9: High temperature blowing ratio sweep, two rows excess coolant: span-averaged overall effectiveness

Additionally, the changes in $\bar{\phi}$ profile shape with increasing blowing ratio are of some interest. The highest blowing ratio cases exhibit less film cooling effect immediately downstream of $x/d = 0$ and feature a slightly more negative slope downstream of $x/d = 15$. If the driving potential for heat transfer (T_{aw}) is held fairly constant across all the cases of Figure 4-9, lower coolant temperatures would generate a greater energy difference between the internal and external flows. Cases with higher T_c values would provide a smaller “low energy reservoir” to absorb energy from the freestream, resulting in less surface temperature change with axial position. Lower T_c values would provide a larger “low energy reservoir,” resulting in lower upstream T_w values and requiring greater axial travel to raise the energy level of the coolant to a level where minimal axial change in T_w occurs. This would further manifest in ϕ data, as shown in the profiles of Figure 4-9. The observed decrease in $\bar{\phi}$ with M was highly unexpected, but

previous discussion of unmeasured coolant temperature increase and possible T_c overestimate at low coolant flow rates could potentially explain this result.

Internal heat transfer coefficient values were also calculated for the data points of Table 4-5. Figure 4-10 presents the h_i data for Table 4-5 as a function of internal coolant flow rate because $\dot{m}_c$ is directly related to h_i while M has no direct influence on internal heat transfer coefficient values. Two separate trends are noticed in the data of Figure 4-10 based on thermocouple location. Thermocouple locations outside the film-cooled region of the test plate (TC 1 and TC 2) feature increasing h_i values with increasing $\dot{m}_c$ values while thermocouple locations within the film-cooled region (TC 3, TC 4, and TC 6) demonstrate little h_i sensitivity to internal coolant flow rate. The relatively constant values of h_i within the film-cooled region for this comparison suggest that differences in $\bar{\phi}$ should be attributed to M , DR , T_c , or $\dot{m}_c$ variations.

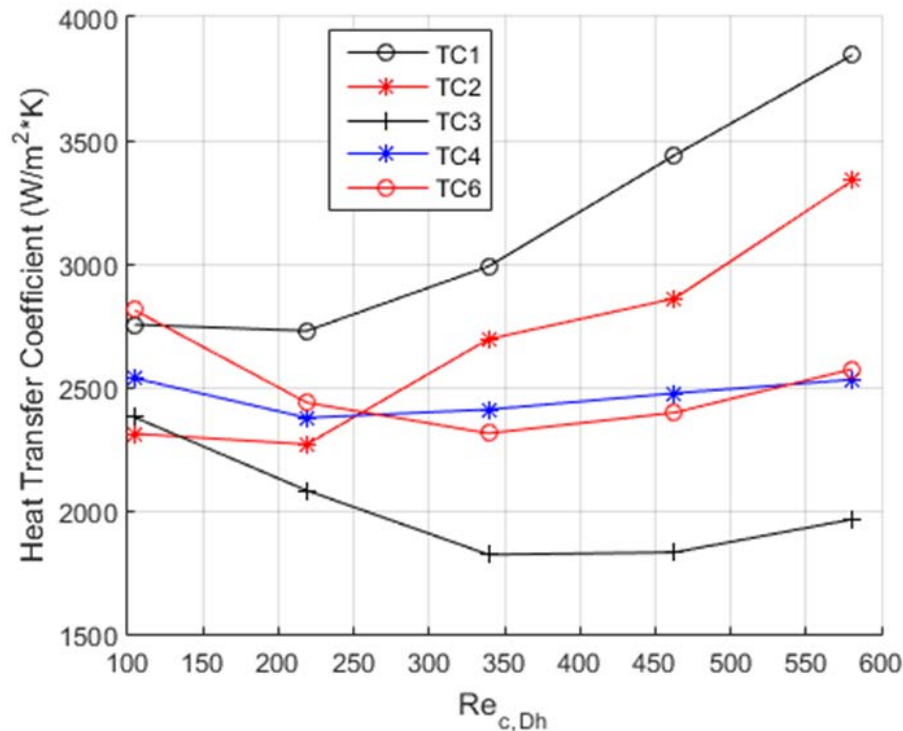


Figure 4-10: High temperature blowing ratio sweep, two rows excess coolant: internal heat transfer coefficient values

Two potential causes for the $\dot{m}_c$ sensitivity of Figure 4-10 are identified for locations 1 and 2: internal flow or external flow. Thermocouples 1 and 2 lie very near the region where coolant impinges on the internal test plate surface (see Figure 3-19), leading to a very strong h_i dependence on $\dot{m}_c$. Alternatively, increased film coolant flow accompanying the higher internal coolant flow could serve to reduce heat flux into the test plate, thus reducing the calculated h_i value. Significant span-wise variation at a given axial position is also noted. Thermocouples 1 and 2 differ by approximately 20 % at the lowest coolant flow rate ($\dot{m}_c \approx 5 \cdot 10^{-3}$ kg/min) while thermocouples 3 and 4 differ by nearly 25 % for the $\dot{m}_c \approx 22 \cdot 10^{-3}$ kg/min case. These differences suggest either poor span-wise uniformity of the internal coolant flow or swirl in the freestream flow causing increased heat flux on one side of the test plate. However, neither of these explanations address the fact that TC 1, which is not film cooled, experiences higher h_i values while TC 4, which is film cooled, also experiences higher h_i values.

The general increase in h_i with $\dot{m}_c$ indicates that $\bar{\phi}$ should increase with $\dot{m}_c$, lending credence to the hypothesis that coolant temperature measurements are in some way flawed. However, assuming the h_i trends of Figure 4-10 are accurate, it is possible that downstream $\bar{\phi}$ values could actually decrease due to reduced h_i values. This would also contribute to the $\bar{\phi}$ profile shapes noted in Figure 4-9. Upstream h_i values increase consistently with Re_{c,D_h} , but the downstream values either decrease or remain nearly constant. This difference in h_i trends would create a larger difference between upstream and downstream $\bar{\phi}$ values consistent with the results of Figure 4-9.

4.2.2.3. Matched Excess Internal Coolant

Because cases were not collected with exactly matched excess internal coolant flow to isolate blowing ratio effects, Table 4-6 displays experimental conditions for a film coolant flow sweep with approximately matched excess internal coolant flow. Excess internal coolant flow varies from 19 g/min up to 23.53 g/min (24% variation). All points feature an approximately matched internal coolant flow rate, density ratio, and coolant temperature with the exception of Point 10. Point 10 is shown last because it was collected without preheating the coolant.

Table 4-6: Approximately matched internal coolant flow blowing ratio sweep high temperature experimental conditions

#	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_{c,in}$ (10^{-3} kg/min)	$\dot{m}_{c,out}$ (10^{-3} kg/min)	Extra Coolant (rows)	M (-)	I (-)	Re_c (-)	Re_∞ (-)	Re_{c,D_h} (-)
12	840	1479	1.80	24.27	22.51	5	1.05	0.61	326	206	518
22	831	1470	1.81	24.27	22.42	5	1.10	0.67	345	207	522
26	831	1496	1.84	21.63	19.00	2.7	1.55	1.32	492	207	465
28	829	1476	1.82	27.04	23.53	2.5	2.07	2.37	655	208	582
31	832	1474	1.81	27.04	22.66	2	2.62	3.79	817	206	581
10	774	1464	1.93	24.27	22.51	5	1.04	0.56	343	209	545

Surface temperature values, shown in Figure 4-11, are in fairly good agreement, spanning a range of 20 K for the cases in best agreement. The $M = 1.04$ case yielded the lowest $\overline{T_w}$ values by up to 20 K. This is expected because the coolant temperature for this case was over 50 K cooler than any other case considered. The $M = 1.05$ case still stands alone with elevated $\overline{T_w}$ values for $x/d < 15$. These $\overline{T_w}$ profiles do not appear to have the expected trend of decreasing $\overline{T_w}$ with increasing M . No trend is observed even if ordering the points by coolant mass flow ($M = 1.55, 1.05, 1.1, 2.07, 2.62$). Further exploration, possibly with the aid of additional instrumentation, is necessary to better understand the lack of expected trends for these data points.

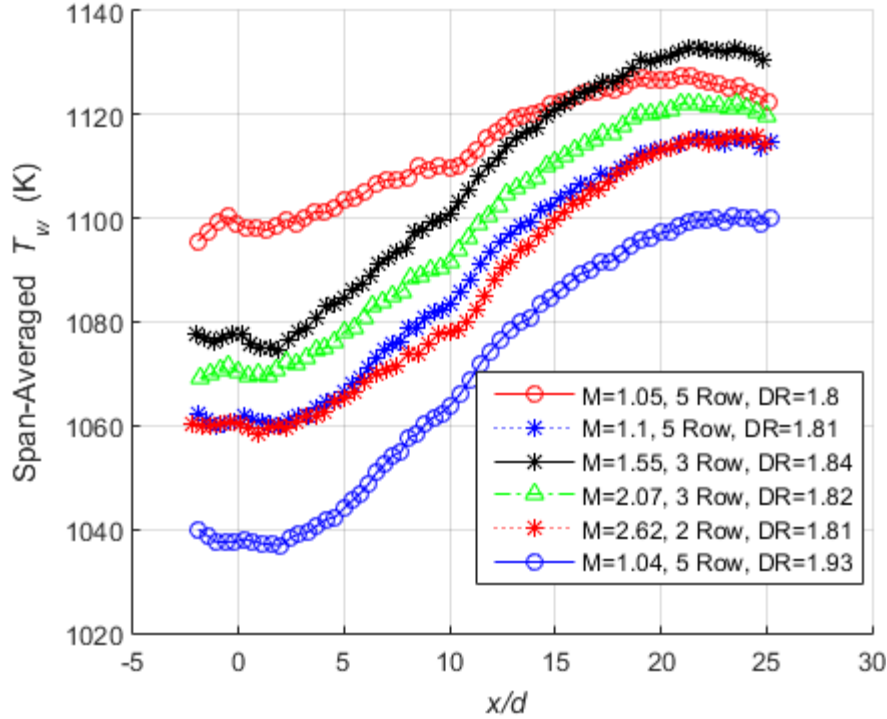


Figure 4-11: High temperature approximately matched coolant flow: span-averaged surface temperature

Figure 4-12 displays $\bar{\phi}$ values for the conditions shown in Table 4-6. The $M = 1.04$ case is shown last in the legend because the coolant temperature was intentionally set lower for comparison. With matched coolant flow, the $\bar{\phi}$ values for these cases span a maximum range of 0.05 and are in relatively good agreement. The expected trend of increasing effectiveness with blowing ratio is not observed. In fact, no trend between blowing ratio and $\bar{\phi}$ values is readily apparent. Potential time dependent effects may be responsible, including potential for elevated effectiveness values for points collected toward the end of a day's testing or towards the end of the overall data collection sequence (Section 3.8.3.1). However, these trends fail to address the relative disorder present in Figure 4-12. The last point collected in the sequence was $M = 1.1$, while the second point of the sequence was $M = 1.05$. Further, the $M = 1.05$ point was the second point collected during its daily sequence, which further suggests its $\bar{\phi}$ values should be

reduced according to the trends from the repeatability analysis of Section 3.8.3.1. The lack of expected trends for the data of Figure 4-12 suggests that future testing should incorporate all points for a comparison into a single day's testing to be run as a continuous sequence.

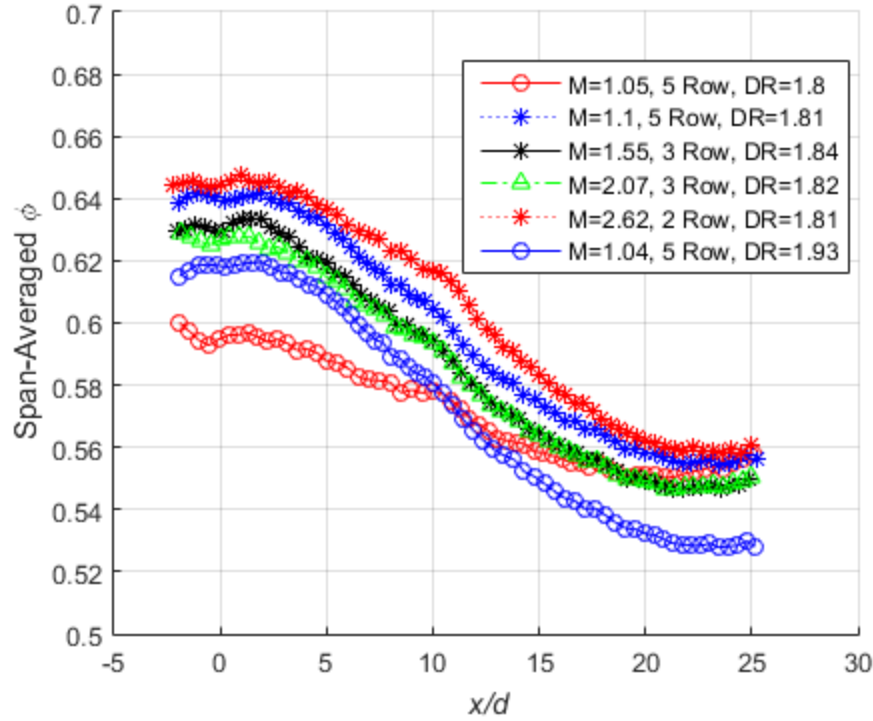


Figure 4-12: High temperature approximately matched coolant flow: span-averaged overall effectiveness

Cases at $M = 1.05$ and 1.04 vary more noticeably from the overall $\bar{\phi}$ trend in Figure 4-12, with both cases exhibiting decreased $\bar{\phi}$ values (approximately 0.03 and 0.01, respectively, below average of cases considered). The regions of decreased effectiveness are $x/d < 15$ and $x/d > 13$ for $M = 1.05$ and 1.04 respectively. The flatter profile of the $M = 1.05$ case is similar to the $M = 0.52$ point of Figure 4-8 and Figure 4-9. This is attributed to a smaller range in h_i and lower upstream h_i values, as shown in Table 4-7. The $\Delta h_{i,max}$ column displays the difference between the minimum and maximum h_i values for a given condition. As discussed in Section 4.2.2.2, a smaller spread in h_i values should result in a smaller range of $\bar{\phi}$ values, creating a flatter $\bar{\phi}$

profile. The reduced upstream h_i values would also cause the observed lower upstream $\bar{\phi}$ values for the $M = 1.05$ case. Because the range and magnitude of h_i values for the $M = 1.04$ case are roughly equivalent to that for other cases considered, the greater downstream decrease in $\bar{\phi}$ must be attributed to a combination of T_c and DR effects. Due to the elevated density ratio and lower coolant temperature for the $M = 1.04$ case, this difference in profile is better aligned with the “energy reservoir” discussion of Section 4.2.2.2.

Table 4-7: Approximately matched internal coolant flow blowing ratio sweep internal heat transfer coefficient values

#	M (-)	T_c (K)	T_∞ (K)	DR (-)	Re_{c,D_h} (-)	h_i (W/m ² · K)					$\Delta h_{i,max}$
						TC1	TC2	TC3	TC4	TC6	
12	1.05	840	1479	1.80	518	3537	3069	2056	2623	2514	1481
22	1.10	831	1470	1.81	522	4166	3694	2522	2658	2678	1644
26	1.55	831	1496	1.84	465	3907	3589	2049	2640	2766	1858
28	2.07	829	1476	1.82	582	3886	3714	1990	2566	2537	1896
31	2.62	832	1474	1.81	581	3963	3538	2192	2671	2694	1770
10	1.04	774	1464	1.93	545	3846	3337	1964	2531	2573	1882

Other unexpected results include the apparent lack of trend between $\bar{\phi}$ and M . No definite trend between M and $\bar{\phi}$ exists, although this could be obfuscated by the slight changes in Re_{c,D_h} . Cases for $M < 2$ feature Re_{c,D_h} values up to 20% smaller than the $M > 2$ cases. This should result in reduced effectiveness for these cases, but the $M = 1.1$ case still provides the second greatest $\bar{\phi}$ values shown in Figure 4-12. Because the difference in Re_{c,D_h} is relatively small, explaining $\bar{\phi}$ differences using the discussion of Section 4.2.2.2 is challenging. However, that discussion does provide possible explanation for decreased $\bar{\phi}$ values with increased Re_{c,D_h} . Of course, this completely overlooks the $M = 2.62$ case which provides the greatest $\bar{\phi}$ values presented.

Time dependent effects may contribute to the apparent lack of trend between M and $\bar{\phi}$. The repeatability discussion of Section 3.8.3.1 revealed a trend of increasing $\bar{\phi}$ in time. Points presented in Figure 4-12 were collected in the order $M = 1.55, 1.05, 2.07, 2.62, 1.04$, and 1.1 . Accounting for potential $\bar{\phi}$ increase in time, it stands to reason that the $M = 1.1$ case would outperform the $M = 1.05$ and 1.04 cases. However, disentangling time-dependent $\bar{\phi}$ effects from actual $\bar{\phi}$ changes due to M presents a real challenge. The best recommendation available is to repeat the data collection, collecting measurements for all necessary points consecutively to minimize any potential time-dependent effects.

Additionally, a decrease in $\bar{\phi}$ of approximately 0.02 for the $M = 2.07$, 2.5 row excess coolant flow case of Figure 4-12 was observed as compared to the $M = 2.08$, two row excess coolant flow case of. Comparing the h_i values for the $M = 2.07$ case in Table 4-7 with those for the $M = 2.08$ case in Figure 4-10, the increase in Re_{c,D_h} from 462 to 582 (26% increase) provided an increase in upstream h_i values of approximately 20%, consistent with expectation. However, h_i values at the downstream thermocouple locations were nearly constant in spite of the Re_{c,D_h} increase. While this should result in increased $\bar{\phi}$, the discussion of Section 4.2.2.2 dealing with T_c measurement accuracy dependence on coolant flow rate helps explain how $\bar{\phi}$ could decrease in spite of increased h_i and Re_{c,D_h} values.

4.2.3. External Heat Transfer Coefficient and Adiabatic Wall Temperature Estimation

To better understand the overall film cooling flow field, T_{aw} and h_f must be quantified. Knowledge of these two parameters would provide insight on the relative effects of internal and external cooling. Further, as demonstrated in Eq. (2-44), knowledge of both T_{aw} and h_f is critical to ensure near ambient and engine representative conditions are correctly matched. The

extrapolation method of Greiner et al. [30] described in Sections 2.5.1 and 3.7.2 is used to estimate T_{aw} and h_f values.

As stated in Section 2.5.1, correct use of the T_{aw} extrapolation method of Greiner et al. [30] demands that all experimental measurements used for an extrapolation be taken with the same external flow field. This requires matching blowing ratio and density ratio while forcing the surface temperature to vary. Surface temperature variations in this study were achieved by varying internal coolant flow rate. However, the strong dependence of T_c on $\dot{m}_c$ creates significant difficulty in matching density ratio. This section highlights how difficulty matching density ratio and a large uncertainty in heat flux posed a serious challenge to obtaining useful T_{aw} and h_f values.

A reproduction of the representative T_{aw} extrapolation from Section 3.7.2 is presented with particular attention drawn to the experimental conditions selected for the extrapolation. Table 4-8 displays experimental conditions selected to estimate T_{aw} at $M = 1.05$ and $DR \approx 1.55$. The test plate surface temperature was forced to vary by increasing the internal coolant flow rate from zero to one row excess coolant flow, resulting in a 35 K decrease in T_c and 0.05 increase in DR . Only these two points are used to maintain a constant DR value.

Table 4-8: T_{aw} estimate experimental conditions: $M = 1.05$, $DR \approx 1.55$

#	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_\infty$ (10^{-3} kg/min)	$\dot{m}_{c,in}$ (10^{-3} kg/min)	$\dot{m}_{c,out}$ (10^{-3} kg/min)	Extra Coolant (rows)	M (-)	I (-)	Re_c (-)	Re_∞ (-)
16	1010	1502	1.52	776	1.76	0.00	0	1.05	0.73	291	203
17	975	1499	1.57	776	6.14	4.38	1	1.05	0.71	297	204

While the T_c and DR variations in Table 4-8 are not ideal, Figure 4-13 shows that the T_w and q'' data collected provided a reasonable trend for T_{aw} estimation at three thermocouple locations: TC 1, TC 2, and TC 4. The estimated T_{aw} value for each thermocouple is determined

as the T_w value for which q'' is zero, indicated in Figure 4-13 as the x-axis intercept of the trend lines generated by the two data points for each thermocouple. A list of estimated T_{aw} values is given in Table 4-9. The fitted curves for thermocouples 3 and 6 (dashed lines in Figure 4-13) rely on insignificant differences in T_w and q'' and should be disregarded. A larger spread in T_w will generate a larger spread in q'' and should generate a usable T_{aw} estimate. Alternatively, including more measurements in the data set used to generate the trend lines could improve T_{aw} estimate accuracy.

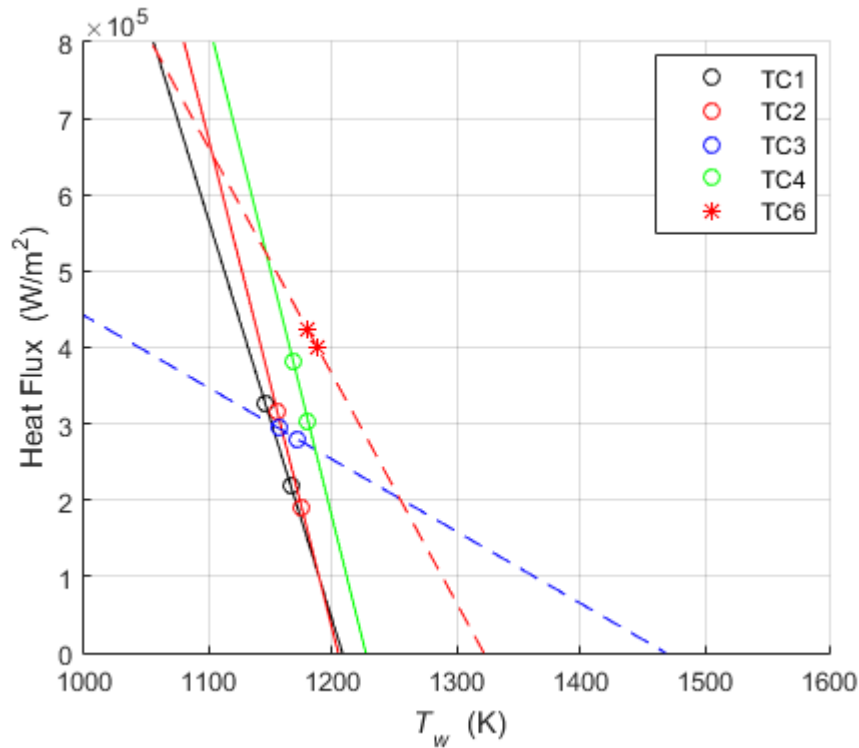


Figure 4-13: Adiabatic wall temperature extrapolation for $M = 1.05$, $DR \approx 1.55$ high temperature condition

Table 4-9 presents the T_w and q'' values used for the extrapolation depicted in Figure 4-13 as well as the T_{aw} and h_f values calculated from the extrapolation. As discussed in Section 3.7.2, the *app* and *CP* subscripts display corrected values. The correction for T_{aw} accounts for curvature in the T_w vs. q'' function while the h correction gives a value if all properties are

evaluated at T_∞ instead of T_{aw} . Considering the $\overline{T_w}$ values near 1170 K and the very low coolant flow for the cases of Figure 4-13, it seems unlikely that T_w and T_{aw} would differ by a large margin. Thus, the T_{aw} estimates from TC 1, TC 2, and TC 4 seem most reasonable.

Table 4-9: Apparent and corrected adiabatic wall temperature and heat transfer coefficient for $M = 1.05$, $DR \approx 1.55$ high temperature condition

TC #	x/d	y/d	$\overline{T_w}$ (K)	$\overline{q''}$ (kW/m ²)	$T_{aw,app}$ (K)	T_{aw} (K)	h_{app} (W/m ² ·K)	h_{CP} (W/m ² ·K)
1	5	-14	1156	273	1208	1209	5258	5137
2	5	14	1165	254	1205	1205	6440	6327
3	10	-10	1164	287	1469	1491	943	823
4	10	10	1175	342	1227	1228	6526	6375
6	20	10	1184	412	1322	1326	2985	2808

Comparing the values of Table 4-9 to those of Greiner et al. [30] shown in Table 4-10 is complicated by significant differences between the two experiments. Both the computational [30] and experimental studies [60] by Greiner et al. do not incorporate film cooling, relying instead on a flat plate with a constant freestream temperature. Also, the T_w and q'' values considered differ greatly from those encountered during film cooling experimentation for this study. Thus, the comparison to the data of Greiner et al. [30] is presented primarily in terms of trends. As expected, the corrected T_{aw} values were greater than the apparent T_{aw} values for this study. The difference between corrected and apparent values ($T_{aw} - T_{aw,app}$) is much greater for Greiner et al. [30], but this stems from a much larger difference between T_w and T_∞ than encountered in this study. Finally, the h_{CP} value is less than the h_{app} value, consistent with the findings of Greiner et al. Again, the difference between the apparent and corrected values is much smaller for the data of this study, consistent with the smaller difference between T_w and T_∞ as compared to the data of Greiner et al. [30].

Table 4-10: Apparent and corrected adiabatic wall temperature and heat transfer coefficient from Greiner et al. [30]

T_w (K)	q'' (kW/m ²)	T_∞ (K)	$T_{aw,app}$ (K)	T_{aw} (K)	h_{app} (W/m ² ·K)	h_{CP} (W/m ² ·K)
700	51	1600	1471	1592	76.7	56.1
750	53					
800	59					

Assuming a constant T_∞ , changing T_c will change DR and affect the mixing of film coolant and freestream gas. However, changing T_c also changes T_w and q'' , providing a larger range of T_w and q'' values and potentially enhancing the accuracy of T_{aw} estimates. Table 4-11 displays a list of experimental conditions used to estimate T_{aw} for $M \approx 1.05$. A relatively large range of T_c and DR values are included, featuring $830 \text{ K} \leq T_c \leq 1010 \text{ K}$ and $1.52 \leq DR \leq 1.83$. The changes in T_c were effected by changing the internal coolant flow rate, ranging from zero to five rows excess coolant. Also, data points acquired at the same conditions were averaged to reduce scatter within the T_w and q'' data. This included points with 2 and 5 rows excess coolant. The 2 row excess average point incorporated Point #'s 9, 11, 13, 14, 15, 18, 19, 20, and 21 while the 5 row excess average point incorporated Point #'s 7, 12, and 22.

Table 4-11: T_{aw} estimate experimental conditions: $M \approx 1.05$, $1.52 \leq DR \leq 1.83$

#	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_\infty$ (10 ⁻³ kg/min)	$\dot{m}_{c,in}$ (10 ⁻³ kg/min)	$\dot{m}_{c,out}$ (10 ⁻³ kg/min)	Extra Coolant (rows)	M (-)	I (-)	Re_c (-)	Re_∞ (-)
16	1010	1502	1.52	776	1.76	0.00	0	1.05	0.73	291	203
17	975	1499	1.57	776	6.14	4.38	1	1.05	0.71	297	204
Avg	912	1480	1.66	776	10.82	9.06	2	1.05	0.67	310	205
8	878	1504	1.75	780	15.20	13.45	3	1.04	0.62	315	204
6	839	1502	1.83	788	19.88	18.12	4	1.04	0.59	326	206
Avg	835	1472	1.80	782	24.27	22.48	5	1.06	0.63	333	207
10	774	1464	1.93	781	24.27	22.51	5	1.04	0.56	343	209

The graphical results of the T_{aw} estimate process are shown in Figure 4-14. Trends of decreasing q'' with increasing T_w are present at each thermocouple location, providing physically realistic T_{aw} estimates. Assuming the results of Figure 4-14 present the correct trends in T_{aw} based upon spatial position, the upstream thermocouples (TC 1 and TC 2) both present much lower T_{aw} values than the downstream thermocouples (TC 3, TC 4, and TC 6). However, because the data points of Table 4-11 were collected on three separate days, the observed trends of Figure 4-14 may result from time-dependent FCR performance effects. For this reason, it is again suggested that all data points needed for a comparison be collected on a single day over as small a time span as possible. Further, the lack of confidence in the estimated T_{aw} and h_f values implies that any calculated η and Bi values will lack accuracy.

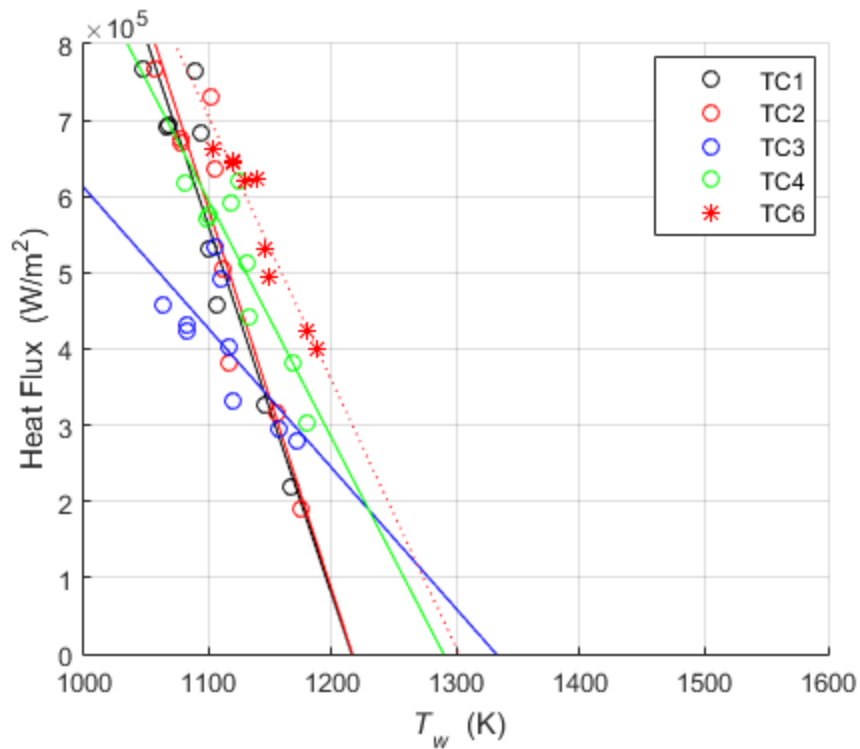


Figure 4-14: Adiabatic wall temperature extrapolation for $M = 1.05$, $1.52 \leq DR \leq 1.83$ high temperature condition

Using averaged T_w and q'' values for the 2 and 5 row excess internal coolant flow, $M \approx 1.05$ cases greatly reduced data scatter, providing at minimum an appearance of improved accuracy for T_{aw} and h_f values. Values for apparent and corrected T_{aw} and h_f are shown in Table 4-12 as averages of the values calculated for each individual case. Because correcting from apparent values incorporates case-specific temperature measurements (see Section 3.7.2 and Eqs. (2-17) and (2-18)), the corrected T_{aw} and h_{CP} values differ for each case. Thus, the values of Table 4-12 represent the average across all conditions considered. The higher h_{CP} values at TC1 and TC2 than at TC3, TC4, and TC6 most likely result from an absence of coolant coverage over these thermocouples.

Table 4-12: Apparent and corrected adiabatic wall temperature and heat transfer coefficient for $M = 1.05, 1.52 \leq DR \leq 1.83$

TC #	x/d	y/d	$\overline{T_w}$ (K)	$\overline{q''}$ (kW/m ²)	$T_{aw,app}$ (K)	$\overline{T_{aw}}$ (K)	h_{app} (W/m ² ·K)	$\overline{h_{CP}}$ (W/m ² ·K)
1	5	-14	1156	273	1212	1215	4951	4698
2	5	14	1165	254	1213	1216	5095	4874
3	10	-10	1164	287	1314	1323	1990	1818
4	10	10	1175	342	1283	1289	3270	3046
6	20	10	1184	412	1306	1312	3394	3158

From the h_{CP} data of Table 4-12, spatially discrete Bi values were calculated, as shown in Table 4-13. The h_f values used to calculate Bi were taken as the lowest and highest of the values used to calculate $\overline{h_{CP}}$ as shown in Table 4-12. The lowest and highest Bi cases were selected to display the impact of T_w and q'' on h_{CP} using the method of Greiner et al. [30]. These T_w and q'' variations caused the shown variation in Bi of up to 0.09 for a single blowing ratio. Calculated Bi magnitudes indicate considerably less thermal resistance to conduction through the test plate material as compared to convection through the external boundary layer flow. Given the small thickness and high conductivity of the test plate, this is not surprising.

Table 4-13: Biot number values for $M = 1.05$, $1.52 \leq DR \leq 1.83$

TC #	x/d	y/d	k (W/m·K)		h_{CP} (W/m ² ·K)		Bi (-)	
			Low	High	Low	High	Low	High
1	5	-14	25.48	25.22	4153	5459	0.207	0.274
2	5	14	25.66	25.44	3771	5589	0.187	0.279
3	10	-10	25.72	25.54	1562	2201	0.077	0.109
4	10	10	25.98	25.84	2744	3665	0.134	0.180
6	20	10	26.29	26.10	2933	3465	0.142	0.169

4.2.4. Internal Coolant Flow Sweep

By holding blowing ratio constant, the combined effects of varied internal coolant flow rate and coolant temperature may be observed. Table 4-14 includes experimental conditions for a sweep of internal coolant flow rate conducted at $M = 1.05$. Points are listed in order of excess coolant flow. It is worthwhile to note that T_c varies by 170 K over the range of internal coolant flow rate considered, yielding a 0.25 range in DR . Any differences in results must thus be considered a combined effect of varied coolant temperature and internal coolant flow rate.

Table 4-14: Internal coolant flow sweep high temperature experimental conditions

#	T_c (K)	T_∞ (K)	DR (-)	$\dot{m}_{c,in}$ (10 ⁻³ kg/min)	$\dot{m}_{c,out}$ (10 ⁻³ kg/min)	Extra Coolant (rows)	M (-)	I (-)	Re_c (-)	Re_∞ (-)	Re_{c,D_h} (-)
16	1010	1502	1.52	1.76	0.00	0	1.05	0.73	291	203	33
17	975	1499	1.57	6.14	4.38	1	1.05	0.71	297	204	119
14	917	1498	1.67	10.82	9.06	2	1.05	0.66	309	204	219
8	878	1504	1.75	15.20	13.45	3	1.04	0.62	315	204	316
6	839	1502	1.83	19.88	18.12	4	1.04	0.59	326	206	425
12	840	1479	1.80	24.27	22.51	5	1.05	0.61	326	206	518

4.2.4.1. Surface Temperature

Figure 4-15 displays the $\overline{T_w}$ values accompanying the data of Table 4-14. As expected, the combination of increased $\dot{m}_c$, and thus internal heat transfer coefficient, and decreased T_c

results in lower $\overline{T_w}$ values for cases with higher internal coolant flow. Cases with 3 and 4 rows of excess coolant agree fairly well until $x/d \approx 12$, at which point the 4 row case exhibits lower $\overline{T_w}$ values as expected. Compared to the 3 and 4 row cases, the 5 row case features slightly greater $\overline{T_w}$ values for $x/d < 5$ and lower $\overline{T_w}$ values for $x/d > 10$. These seemingly errant $\overline{T_w}$ values are attributed to the order of data collection. Because the case with 5 rows of excess coolant was collected on a different day than the other points of Figure 4-15, the repeatability analysis of Section 3.8.3.1 indicates that the comparison may be compromised by variations in FCR performance.

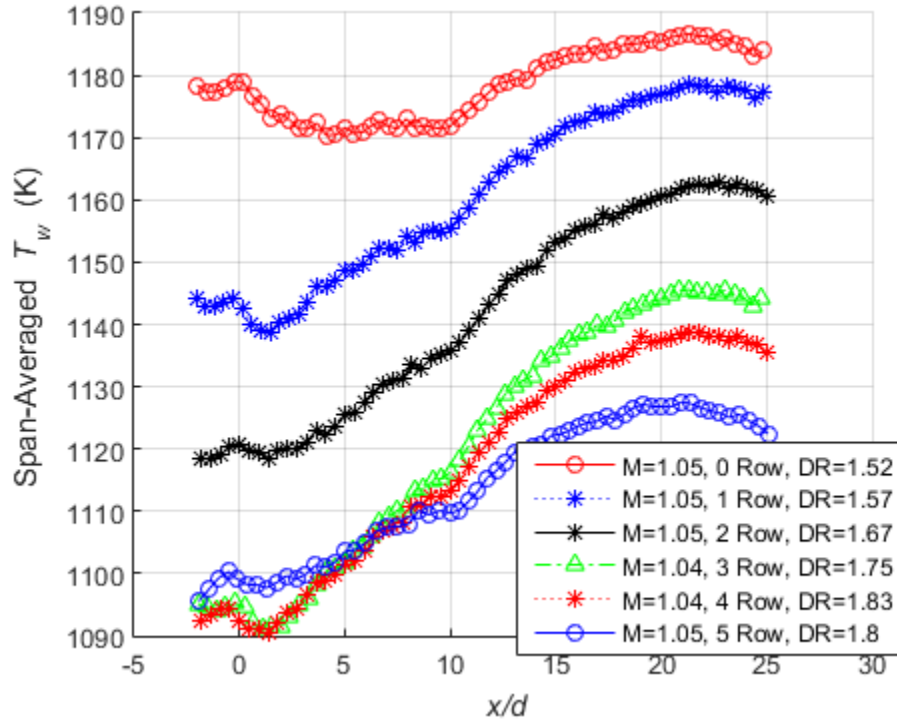


Figure 4-15: High temperature internal coolant flow sweep, $M \approx 1.05$: span-averaged surface temperature

Quantitatively, T_c decreased from 1010 K to 840 K as excess coolant increased from 0 to 5 rows (Re_{c,D_h} increase from 33 to 518). This is as expected. $\overline{T_w}$ decreased by a maximum of 85 K at any given downstream position ($x/d = 2$). Assuming constant T_∞ of 1500 K, this should

correspond to a 0.04 decrease in ϕ . The exact reason for this larger decrease in T_c than T_w is not understood, but it is most likely a rig- or cooling scheme-dependent phenomenon.

4.2.4.2. Internal Heat Transfer Coefficient and Heat Flux

Heat flux measurements for the data points in Table 4-14 are displayed in Figure 4-16. Calculation of heat flux is described in Section 3.7.2 and refers to the heat flux through the test plate. As internal coolant flow increases, heat flux values rise for all thermocouple positions. This rise in q'' is consistent with increased internal coolant Reynolds number from increased internal coolant flow as well as the accompanying increase in energy difference between the freestream and coolant gas flows caused by decreased coolant temperature. The range of Re_{D_h} for these cases spans from approximately 30 to 520. The upstream thermocouple locations demonstrate the greatest increase in heat flux, rising from the lowest heat flux values to the greatest. A combination of decreased coolant temperature, increased h_i values due to internal coolant impingement, and proximity to the external impingement region on the test plate leading edge likely cause this result. Proximity to the leading edge is considered a contributing factor because the high conductivity of the test plate could permit axial conduction from the leading edge to the upstream thermocouple locations.

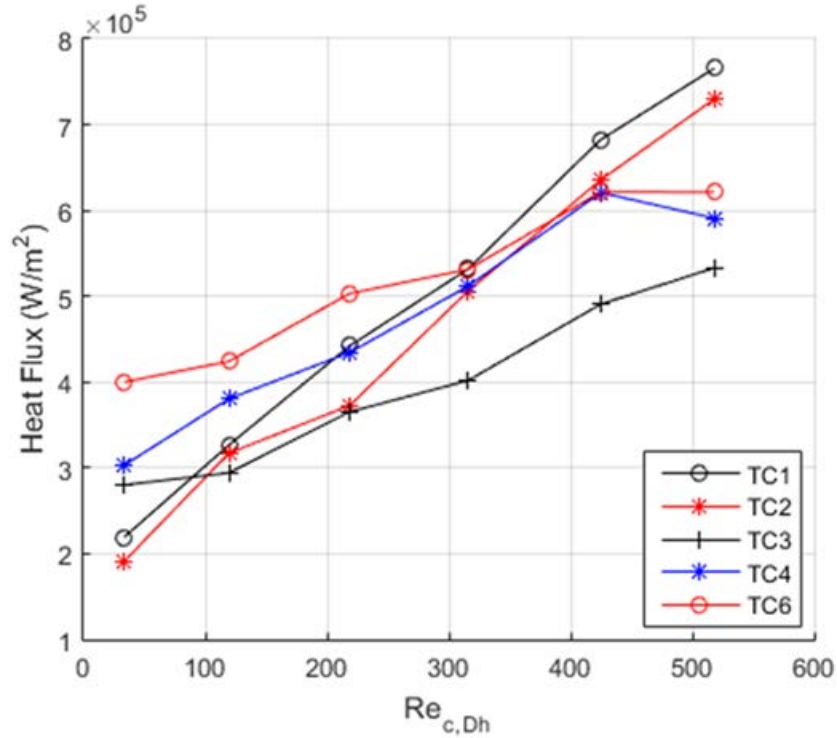


Figure 4-16: High temperature internal coolant flow sweep, $M \approx 1.05$: heat flux

Further, a span-wise difference in q'' values is noted in Figure 4-16. TC 1 reads a greater q'' value than TC 2 for all internal coolant flow rates but TC 3 reads a lesser q'' value than TC 4. Because these q'' trends are consistent with the h_i trends of Figure 4-17, span-wise variation in q'' is accepted as a result of h_i variations due to internal coolant flow effects. To better evaluate trends in q'' , a correlation of internal coolant Reynolds number to Nusselt number follows the h_i discussion.

Figure 4-16 reveals that downstream q'' values are greater than or comparable to upstream q'' values over much of the $Re_{c,Dh}$ range considered, but the upstream q'' values exceed the downstream values for $Re_{c,Dh} > 400$. The cooling scheme examined only ejected film coolant from three holes on either side of the test plate centerline, meaning that TC1 and TC2 were outside the film-cooled region (see Figure 3-18). Because the matched blowing ratio for all cases should result in a nearly identical external flow field, h_f and T_{aw} values should be fairly

constant for all Re_{c,D_h} values at a given thermocouple location. Table 4-11 presents estimated values for h_f , demonstrating that upstream h_f values are approximately 50% higher than at downstream positions. Assuming the change in surface temperature is spatially uniform as Re_{c,D_h} increases, the lower downstream h_f values would cause downstream q'' values to increase more slowly than upstream q'' values. Eventually, the upstream q'' values would exceed those measured at downstream positions. However, the large q'' uncertainty of 97 % implies that none of the results of Figure 4-16 can be said to conclusively differ. Thus, any observed trends may or may not correspond to real physical effects.

This difference in coolant coverage should reduce the potential for heat transfer at the downstream thermocouple locations, resulting in a less dramatic increase in heat flux as compared to the upstream locations. However, for higher coolant temperatures (lower coolant in-flows), the downstream locations experienced higher heat flux than the upstream locations. These higher coolant temperatures would provide less cooling effect while the coolant jets would still increase the heat transfer coefficient downstream of the cooling holes. Such an increase in heat transfer coefficient could potentially be overcome by a reduction in film temperature afforded by the greater cooling capacity of lower temperature coolant. To better understand the nature of this dynamic system, external heat transfer coefficients with film cooling present should be examined.

Figure 4-17 displays internal heat transfer coefficient values for the cases of Table 4-14 calculated as described in Section 3.7.2. As coolant in-flow was increased, h_i increased by over 140 % at the upstream thermocouple locations (TC1 and TC2). Downstream thermocouples (TC3, TC4, and TC6) experienced much less variation across the range of coolant in-flow (up to 23 %), suggesting that these locations may experience fully developed internal flow.

Alternatively, these observed h_i trends could result from Nusselt number variations due to increased internal coolant flow. Additionally, while downstream h_i values are spread over a large range at low internal coolant flow rates, this spread dissipates for higher internal coolant flow rates, indicating better flow uniformity across the internal channel span and along the internal channel axial dimension with increased internal coolant flow.

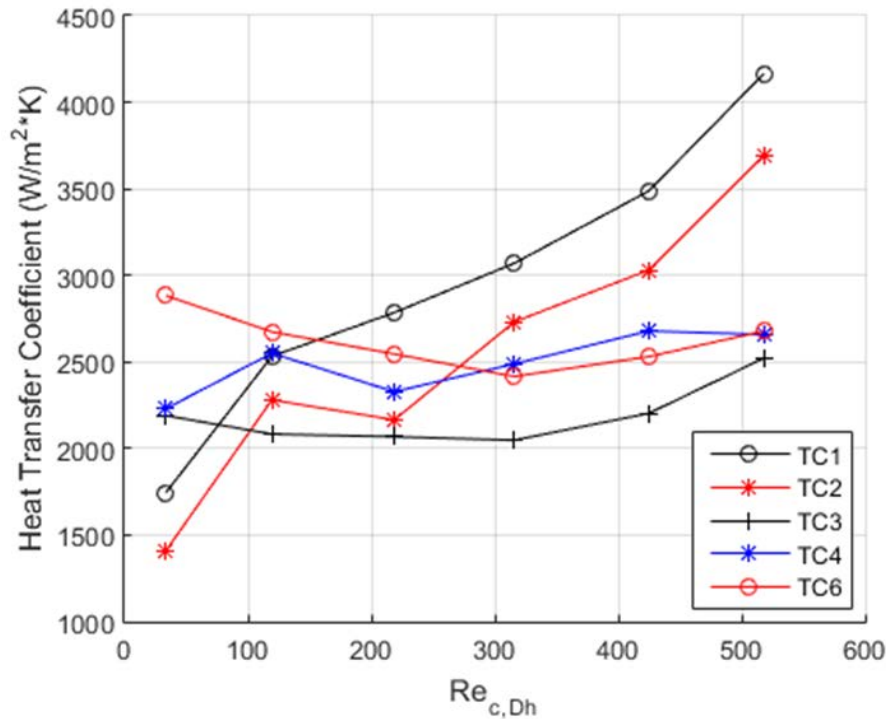


Figure 4-17: High temperature internal coolant flow sweep, $M \approx 1.05$: internal heat transfer coefficient

The different trends in heat transfer coefficient for thermocouple locations 1 and 2 as compared to locations 3-6 in Figure 4-17 are likely due to a combination of thermocouple placement and the sharp turn of the internal coolant flow path, shown in Figure 3-19. This geometry forces the coolant to impinge on the internal surface of the test plate (see Figure 4-3), producing a region of great turbulence. While this turbulence would almost certainly not dissipate before reaching the downstream thermocouple locations, the flow would likely achieve better uniformity at the downstream locations due to increased time for mixing.

Further, different span-wise trends in h_i values are observed at the upstream thermocouple locations as compared to the downstream locations. At the upstream location, TC 1 displays greater h_i values than TC 2 for the entire range of $\dot{m}_c$ considered. While TC 3 yields lower h_i values than TC 4, downstream h_i values appear to be converging at all thermocouple locations for the highest $\dot{m}_c$ values. This change of span-wise h_i trend with axial position suggests a change in the span-wise distribution of the internal coolant flow between the upstream and downstream thermocouple locations. An asymmetric internal flow restriction or asymmetric distribution of coolant exiting the coolant baffle plate could reduce coolant flow to one side of the internal channel, causing the observed upstream span-wise h_i variation. Also, the h_i trend at downstream locations exhibits $\dot{m}_c$ dependence. It is possible that lower internal coolant flow rates fail to distribute uniformly across the entire internal channel span, resulting in the observed span-wise h_i difference at downstream locations that dissipates with increased $\dot{m}_c$. Adding centerline instrumentation would provide additional data helpful to explain these position and coolant flow rate dependent span-wise variations in h_i .

Table 4-15 displays Nusselt number values calculated from the h_i values of Figure 4-17 as well as values calculated using a correlation for fully developed turbulent internal flow with a constant heat rate, shown in Eq. (4-1) [6].

$$Nu_{D_h} = 0.022Pr^{1/2}Re_{D_h}^{4/5} \quad (4-1)$$

It should be noted that the very low Re_{c,D_h} values present for these conditions would generally exclude the possibility of turbulent flow. Mathematically speaking, this explains the enormous difference between calculated and estimated Nu_{c,D_h} values. However, it fails to account for the geometry of the internal coolant channel. The combination of the discrete jets produced by the coolant baffle plate and the 90 degree flow direction turn beneath the test plate almost certainly

results in highly turbulent flow, causing the large calculated Nu_{c,D_h} values in Table 4-15. For a given thermocouple location, Nu_{c,D_h} increases with Re_{c,D_h} as expected. Also, while the 0, 1, and 2 row cases lack a definite trend for Nu_{c,D_h} with axial position, cases with 3 or more rows of excess internal coolant flow exhibit decreasing Nu_{c,D_h} values with axial position indicative of entry region flow. This behavior is consistent with hydrodynamic boundary layer development following the 90 degree flow turn beneath the test plate.

Table 4-15: Calculated and correlated Nusselt number values for high temperature internal coolant flow sweep

#	Extra Coolant (rows)	Re_{c,D_h}	Nu_{c,D_h}					$Nu_{c,est}$
			TC1	TC2	TC3	TC4	TC6	
16	0	33	141	115	178	182	236	0.61
17	1	119	202	185	169	209	221	1.70
14	2	219	244	192	184	208	229	2.75
8	3	316	284	254	192	234	229	3.69
6	4	425	337	295	216	263	250	4.68
12	5	518	394	352	242	257	260	5.48

The disagreement between Nu_{c,D_h} and $Nu_{c,est}$ values in Table 4-15 warrants further investigation to assess the trends in Nu_c . A correlation of $Re_{c,int}$ to Nu_c is desired. The form $Nu = CRe^mPr^n$, where C , m , and n are constants determined by a curve fit, was used to generate the correlation. It is hypothesized that the coolant flow is heavily disturbed as the coolant flow turns beneath the test plate. As such, the hydrodynamic and thermal boundary layers must redevelop, meaning the coolant flow beneath the flat section of the test plate is a turbulent developing flow and downstream position (x) will impact Nu values. Thus, the origin selected to define the length scale necessary for $Re_{c,x}$ and $Nu_{c,x}$ was the downstream edge of the coolant hole inlet. The curve fit results are shown graphically in Figure 4-18. Because of the separation between data points from different thermocouples, three different curve fits were

generated. The first utilized all available thermocouples (TC1, TC2, TC3, TC4, and TC6), the second relied on TC1-TC4, and the last used only TC6. These results do not exhibit the expected trend of decreasing $Nu_{c,x}$ with downstream position, but the positions considered span a fairly small range ($4.7 \text{ mm} < x < 12.4 \text{ mm}$, $0.74 < x/D_h < 1.93$). It is possible that the expected decrease in $Nu_{c,x}$ would not occur until greater x values. Additionally, the decrease in internal channel flow area at $x/d = 30$ would increase flow velocity, thus leading to an increase in both $Re_{c,x}$ and $Nu_{c,x}$. It is also noted that the overall range in $Nu_{c,x}$ decreased with x/d . It is hypothesized that decaying turbulence levels and boundary layer development resulted in decreased dependence on $Re_{c,x}$ for downstream thermocouple locations. Both these phenomena would serve to reduce mixing and fluid velocity adjacent to the internal test plate surface, resulting in decreased convective heat transfer at the downstream locations.

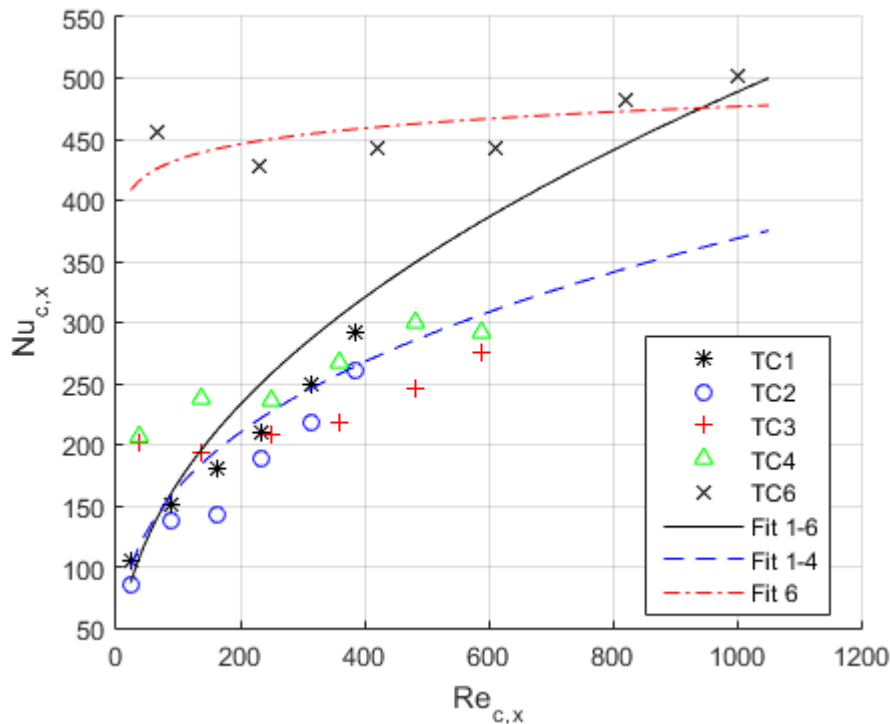


Figure 4-18: Internal heat transfer correlation: high temperature internal coolant flow sweep, $M \approx 1.05$, cooling hole inlet origin

Because the data of Figure 4-18 provided a poor correlation when using data from all thermocouple locations, an alternative origin location was selected at the internal upstream face of the test plate located beneath the leading edge quarter cylinder. This origin provided larger x values ($9 \text{ mm} < x < 14 \text{ mm}$, $1.4 < x/D_h < 2.2$) with a smaller percent difference between locations, resulting in a tighter grouping of data points. With this tighter grouping, the curve fit relying on data from all thermocouples appears to fit the data better, as seen in Figure 4-19. As expected, increasing $Re_{c,x}$ resulted in an increase in $Nu_{c,x}$. This is true when considering $Re_{c,x}$ increases due to increased mass flow or increased x . The increase in $Nu_{c,x}$ due to mass flow related $Re_{c,x}$ increases is expected, but the positional $Nu_{c,x}$ increases are somewhat unexpected as $Nu_{c,x}$ should decrease with x . It is possible that the range of $Re_{c,x}$ considered is too small to observe this effect. Measurement accuracy also likely has a significant impact on the observed trends, especially for the low $Re_{c,x}$ values presented. A larger range of $Re_{c,x}$ would most likely reveal the expected $Nu_{c,x}$ trends with position.

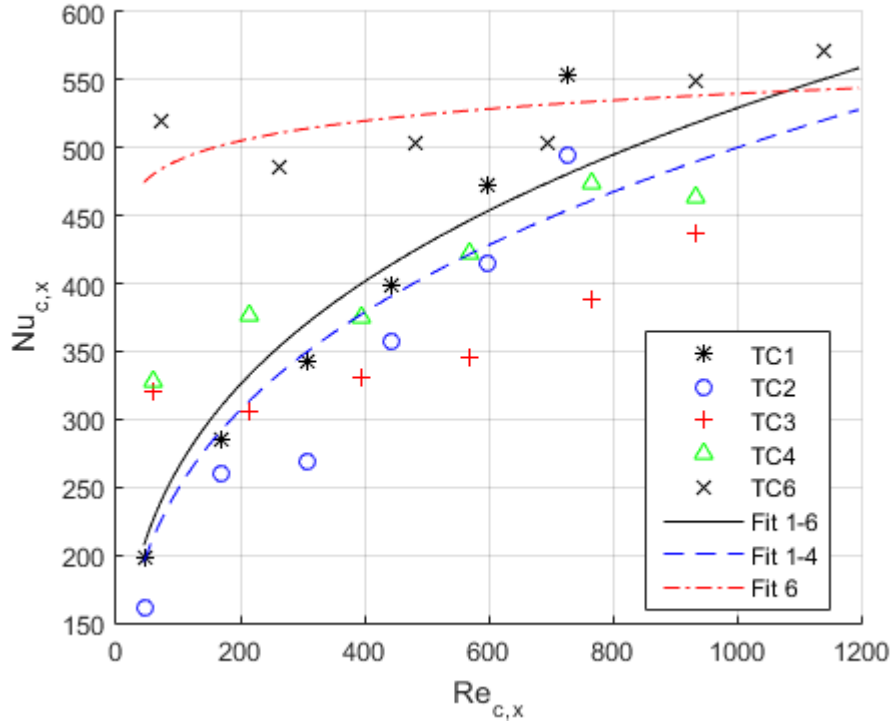


Figure 4-19: Internal heat transfer correlation: high temperature internal coolant flow sweep, $M \approx 1.05$, upstream test plate edge origin

4.2.4.3. Overall Effectiveness

Figure 4-20 displays $\bar{\phi}$ profiles for the data points presented in Table 4-14. It was expected that the plenum-fed case would provide the lowest $\bar{\phi}$ values due to reduced internal convective cooling while increased internal coolant flow should provide increased $\bar{\phi}$ values due to increased internal convective cooling. However, the test case with a single row of excess coolant flow yielded the greatest peak effectiveness value, and the plenum-fed case clearly provided the greatest effectiveness value when considered over the full x/d range presented. This occurred because $\bar{\phi}$ decayed less rapidly with downstream distance for the plenum-fed case, which is consistent with the $\bar{\phi}$ profile shape discussion of Section 4.2.2.2. Additionally, the limited region of elevated $\bar{\phi}$ values for the plenum-fed case ($0 < x/d < 10$) clearly indicated cooling almost exclusively via film cooling as expected. The differences in $\bar{\phi}$ values observed in

Figure 4-20 are great enough to exceed the uncertainty of $\delta\phi = 0.032$ and thus must be investigated further. The impact of observed time dependent effects on $\overline{\phi}$ will be considered. Possible causes for $\overline{\phi}$ decrease with increased coolant flow include T_c measurement inaccuracy due to unmeasured T_c increase between the point of measure and the coolant hole inlet (see previous discussion in Section 4.2.2.2) as well as potential measurement error due to not correcting for radiative and conductive heat transfer into the coolant thermocouple. Both hypotheses provide a reasonable explanation of observed $\overline{\phi}$ trends and are considered further. Additionally, time dependent effects likely play a role in the apparent lack of trend between $\overline{\phi}$ and $\dot{m}_c$.

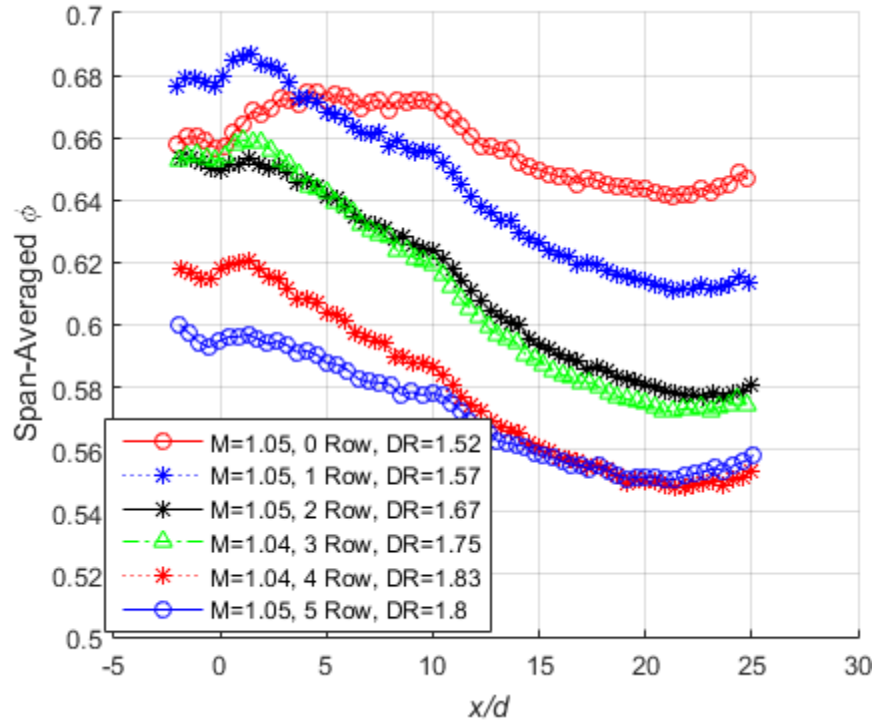


Figure 4-20: High temperature internal coolant flow sweep, $M \approx 1.05$: span-averaged overall effectiveness

The shape of the $\overline{\phi}$ and $\overline{T_w}$ profiles in Figure 4-15 and Figure 4-20 is of some interest. The plenum-fed case features a relatively flat profile with the bulk of cooling effect occurring in

the near-hole region ($0 < x/d < 12$). The cases with excess coolant supplied feature a much stronger downward taper in $\bar{\phi}$ in both the near-hole region and downstream. This is consistent with the larger difference between upstream and downstream h_i values for the higher coolant flow rates, as described in Sections 4.2.2.2 and 4.2.4.2. Such a difference in h_i while externally driven parameters such as η , h_f , and Bi remain constant would result in greater $\bar{\phi}$ dependence on x/d .

Because impingement cooling is generally more effective at reducing surface temperature than film cooling, it seems likely that the low T_w values at small x/d values are a result of the internal coolant flow impinging on the upstream internal surface of the test plate (see Figure 4-3). The channel flow at downstream locations coupled with rising internal coolant temperatures would provide less cooling effect for the test plate, resulting in the large rise in $\bar{T}_w$ and corresponding decay in $\bar{\phi}$. Plenum-fed cases featured much higher T_c values and much lower coolant mass flow rates, resulting in much smaller spreads in $\bar{T}_w$ values and thus produced much flatter $\bar{T}_w$ and $\bar{\phi}$ profiles. Film cooling effect is still discernible at $x/d = 0$, but the overall shape of the $\bar{T}_w$ profiles reflects the relatively greater contribution of the internal impingement cooling.

If considering the order of data collection for the $\bar{\phi}$ profiles in Figure 4-20, the first 5 points (0-4 rows excess coolant) were all collected on one day in the order of 2-0-1-3-4 rows of excess coolant flow. These points were numbers 1, 2, 7, 10, and 13 out of 16 points collected that day. The point with 5 rows excess coolant was the second of six points collected the following day. This order is significant because the repeatability analysis of Section 3.8.3.1 revealed a trend for increased $\bar{\phi}$ values within a given day and from day to day, indicating that

the trend of decreasing $\bar{\phi}$ with increased internal coolant flow is probably a real phenomenon. It is recommended that this coolant flow sweep be revisited with data collection performed with two sequences: once with increasing coolant flow and once with decreasing coolant flow.

Another possible explanation for the $\bar{\phi}$ values shown in Figure 4-20 lies in the design of the FCR. Considering the heat transfer environment at the coolant measurement thermocouples, there is clearly convective heat transfer out of the thermocouples into the coolant gas, conductive heat transfer into the thermocouple from adjacent metal along the Inconel thermocouple sheath, and radiative heat transfer into the thermocouple from surrounding material. As coolant flow rate is increased, the percent contribution to the thermocouple energy balance from internal coolant convection would increase relative to conductive and radiative heat transfer. This would result in measured T_c values greater than the actual T_c value for all conditions, but this difference between measurement and reality would be greatest for low coolant flow rates with minimal convective heat transfer. Additional flow routing issues addressed in Section 4.2.2.2 would serve to further reduce the convective heat transfer away from the coolant thermocouples at low coolant flow rates, further increasing the magnitude of T_c error for low coolant flow rates. If T_c measurements exceed the actual T_c values, resulting $\bar{\phi}$ values would be inflated. While the magnitude of the T_c measurement error is presently unknown, two solutions are recommended to reduce the error.

The extent of coolant mass flow increase required to minimize this error to an acceptable level is presently unknown so future FCR studies should examine this issue in greater detail. Temperature measurements of metal parts surrounding the internal channel thermocouples could be made to facilitate radiation and conduction corrections for T_c measurements. Additionally, the adjacent positioning of the boundary layer bleed slot and main cooling block (see Figure

3-13) creates a counterflow heat exchanger between the boundary layer bleed and coolant gas flows. Insulating the main cooling block from the boundary layer bleed slot could help reduce unwanted heat transfer to both coolant gas and coolant gas thermocouples.

In order to more accurately determine T_c , a solution is proposed to account for radiative and conductive heat transfer effects at the thermocouple tip. Figure 4-21 shows the relevant temperatures influencing the internal channel thermocouple measurement. Metal temperatures are denoted as T_{M1} and T_{M2} , the indicated temperature at the thermocouple weld bead is T_{TC} , and the actual coolant temperature is shown as T_c . Assuming steady state, the temperature of the thermocouple is dependent on radiative heat transfer from surroundings (T_{M1} and T_{M2}), conductive heat transfer through the thermocouple sheath, and convective heat transfer with the internal coolant flow. As such, an energy balance, shown in Eq. (4-2), may be used to assess the actual coolant temperature. Equations (4-3)-(4-5) provide appropriate definitions for the various heat flux terms.

$$q''_{rad} + q''_{conv} + q''_{cond} = 0 \quad (4-2)$$

$$q''_{rad} = \varepsilon\sigma[(T_{M1}^4 - T_{TC}^4) + (T_{M2}^4 - T_{TC}^4)] \quad (4-3)$$

$$q''_{conv} = h(T_c - T_{TC}) \quad (4-4)$$

$$q''_{cond} = \frac{k}{\Delta x}(T_{M2} - T_{TC}) \quad (4-5)$$

Substituting Eqs. (4-3)-(4-5) into Eq. (4-2), the energy balance may be solved for the actual coolant temperature value. It is suggested that thermocouples be placed in the internal coolant channel to measure metal temperatures and h be taken from the furthest upstream test plate thermocouple locations. These metal temperature thermocouples should be anchored by spot welding a Ni-Cr ribbon over the thermocouple tip or exposing the leads and spot welding them directly to the metal walls in close proximity to one another. While these corrections would be a

complex addition to testing, a brief investigation would serve to assess the actual need for radiation and conduction corrections in the future. If corrections are unnecessary, the instrumentation may be removed and the relevant data processing omitted.

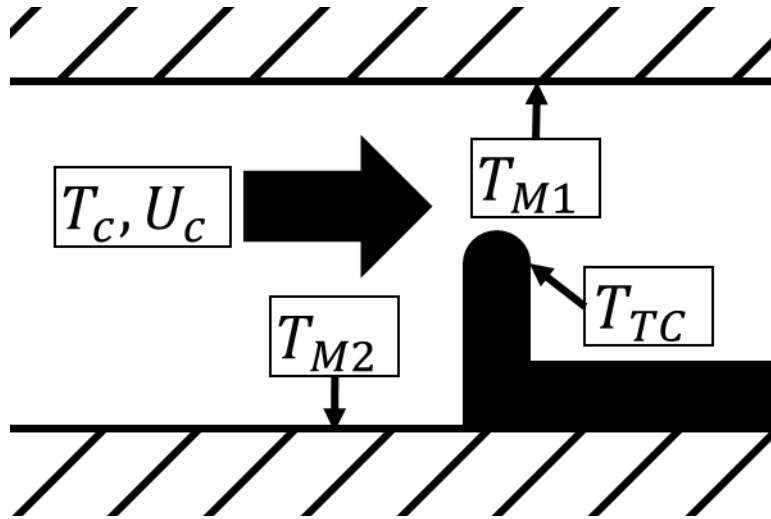


Figure 4-21: Internal channel thermocouple measurement schematic

4.3. Near Ambient Conditions

Similar to the treatment of engine representative conditions in Section 4.2, this section examines data collected at near ambient conditions. The baseline near ambient conditions are analyzed at length followed by a brief consideration of trends resulting from varied internal coolant flow rates and blowing ratios. Because near ambient data collection was structured strictly to support matching engine representative conditions, a limited number of useful comparisons can be made.

4.3.1. Baseline Near Ambient Results

Near ambient conditions included cases with both carbon dioxide and argon coolant ejection. Because the near ambient experimental regime has not previously been explored with the AFIT FCR, a brief analysis of representative results for each cooling gas species is included.

These baseline cases are examined to investigate spatial effects not readily apparent in span averaged data. Emphasis is placed on identifying significant flow features and trends and assessing image quality

4.3.1.1. Carbon Dioxide Baseline Result

The near ambient repeatability condition was considered as the baseline near ambient condition with carbon dioxide film coolant. Figure 4-22 displays T_w contours for one of the near ambient repeatability points, referenced as Point 54 in Table 4-3. Experimental conditions included $M = 1.08$, $DR = 1.66$, $I = 0.70$, and $Re_{\infty,D} = 3760$ with two rows excess internal coolant flow. This condition is the near ambient equivalent of the high temperature repeatability condition, and it was selected as a representative near ambient condition for in-depth analysis of thermal features similar to that of Section 4.2.1. As such the temperature scale was chosen to maximize contrast within the T_w distribution.

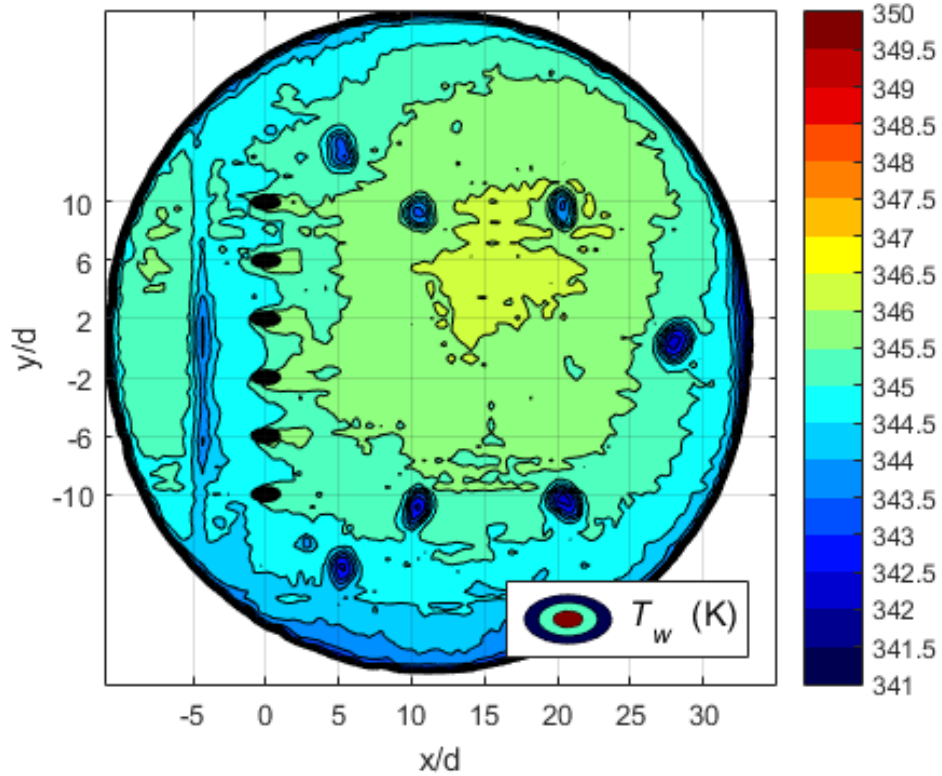


Figure 4-22: Baseline carbon dioxide near ambient condition (#54) surface temperature contours – $M = 1.06$, $DR = 1.66$, $Re_{\infty,D} = 3760$

A very slight region of film cooling effect is apparent immediately downstream of the cooling holes. The magnitude of film cooling effect is very small, resulting in a T_w decrease of no more than 1 K. If the coolant absorbed a significant amount of energy between the point of T_c measurement and the cooling hole entrance, cooling capacity would be reduced compared to expectations. Additionally, convective heat transfer may not significantly outweigh conductive heat transfer at the point of T_c measure, resulting in inflated T_c values (see Section 4.2.4.3). Further, the use of a foreign gas to artificially augment DR eliminates the large energy difference that would be present for an equivalent DR using air coolant. Both the hypothesized T_c measurement errors and foreign gas injection effects could result in the very small cooling effect observed in Figure 4-22. Elevated T_w values for $x/d < -5$ are consistent with freestream gas impingement on the curved leading edge of the test plate. Surface temperature peaks on the flat

after body near $x/d = 15$ as a result of minimal internal and external cooling effect. Higher T_w values are skewed towards $y/d > 0$ consistent with the discussion of Section 4.2.1.

The surface thermocouple weld beads still appear to behave differently than the rest of the surface in spite of uniform flat black paint application to the test plate, suggesting the weld beads experience different flow characteristics than the surrounding test plate. This is consistent with the slight weld bead protrusion above the surrounding surface. However, as suggested in Section 3.3.1.1 with the IR calibration discussion, the high thermal conductivity of the test plate should minimize spatial temperature differences. A thin region of decreased T_w oriented vertical to the viewer at $x/d = -5$ is attributed to poor paint coverage at the transition from curved leading edge to flat after body. Overall, the T_w contours displayed in Figure 4-22 exhibit trends consistent with the baseline high temperature condition of Figure 4-1 and the accompanying discussion of Section 4.2.1.

The pattern of decreasing T_w contours emanating from the region of peak T_w centered at $x/d = 16$, $y/d = 6$ indicates possible influence from IR reflections or optical distortion near the sapphire window edge. Background IR radiation could also play a role in generating the puzzling features of Figure 4-22. Four solutions are recommended to correct for or minimize these effects.

First, the sapphire window used did not feature any form of anti-reflective coating. A dedicated window for near ambient testing with an anti-reflective coating would drastically reduce IR radiation reflected off the window from surroundings and increase contrast within the test plate field of view. Creating some form of enclosure for the FCR and camera to shield them from surroundings would serve to minimize influence of background radiation. Additionally, an IR image could be collected before testing and subtracted from test images to remove the

influence of background radiation. Optical distortion effects should be accounted for by viewing an object of a uniform, known temperature positioned where the test plate would reside during testing. A blackbody would be ideal for this correction. A blackbody could also be used to track IR window transmissivity in time. However, as long as a dedicated window is utilized for near ambient tests, this should be of minimal concern as significant window fouling has only been observed during engine representative testing.

Overall effectiveness contours for the data of Figure 4-22 are shown in Figure 4-23. Because T_c is taken as a fixed value, any variation in T_w is manifested directly in ϕ . Thus, Figure 4-23 displays all the same trends as Figure 4-22. Some features are slightly more difficult to discern because of scale, but the scale used provides consistency across all ϕ contour plots.

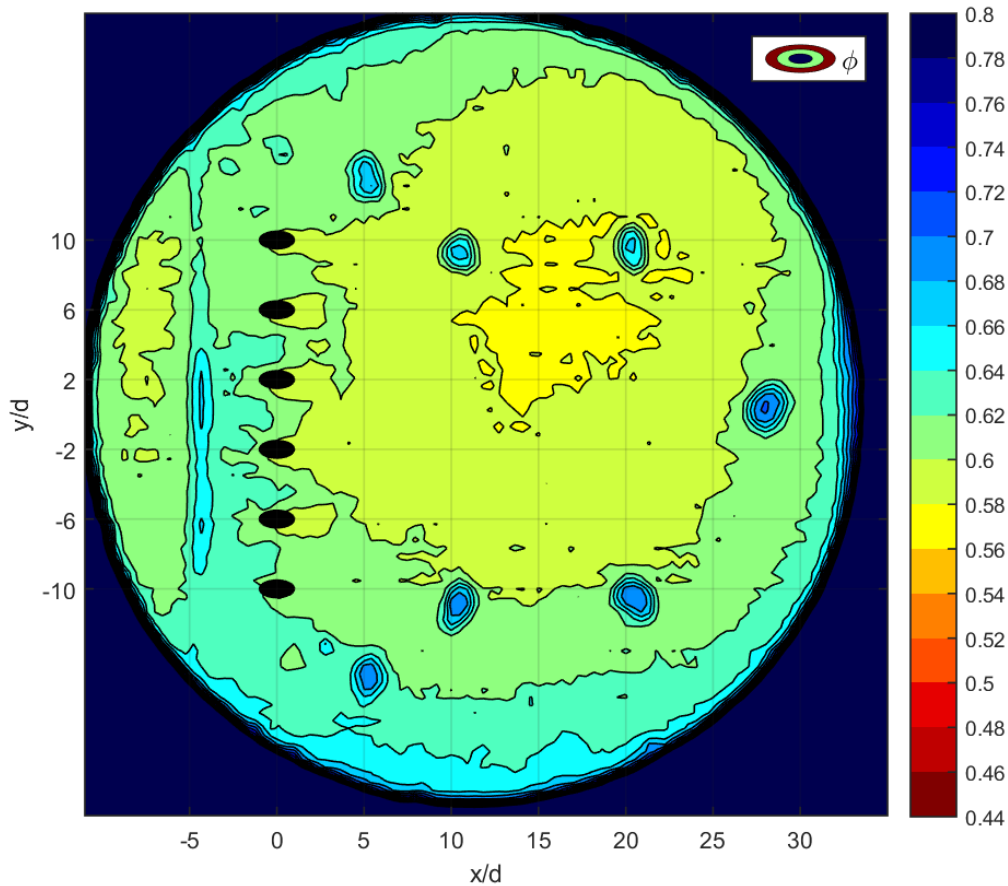


Figure 4-23: Baseline carbon dioxide near ambient condition (#54) overall effectiveness contours – $M = 1.06$, $DR = 1.66$, $Re_{\infty,D} = 3760$

4.3.1.2. Argon Baseline Result

A sample T_w contour plot for the argon cooled near ambient conditions (Point #7 in Table 4-2) is presented in Figure 4-24. Experimental conditions for Figure 4-24 include $M = 1.01$, $DR = 1.52$, and $I = 0.67$, and $Re_{\infty,D} = 5540$ with an argon cooled, plenum-fed cooling scheme. This condition was selected because no equivalent to the carbon dioxide cooled near ambient or air cooled high temperature repeatability conditions could be attained using argon coolant due to rig limitations. Several of the noted features from Figure 4-1 and Figure 4-22 are also present in Figure 4-24, including: elevated T_w values along the test plate leading edge and for $y/d > 0$; reduced temperature on the surface thermocouple weld beads as well as upstream of and between film cooling holes; and decreasing temperature with axial position. All of these features are consistent with the analysis of baseline results in Sections 4.2.1 and 4.3.1.1. As with the discussion of Section 4.3.1.1, this point is intended to identify prominent features in the T_w distribution for near ambient, argon cooled conditions. The temperature scale was thus chosen to maximize contrast within the T_w distribution.

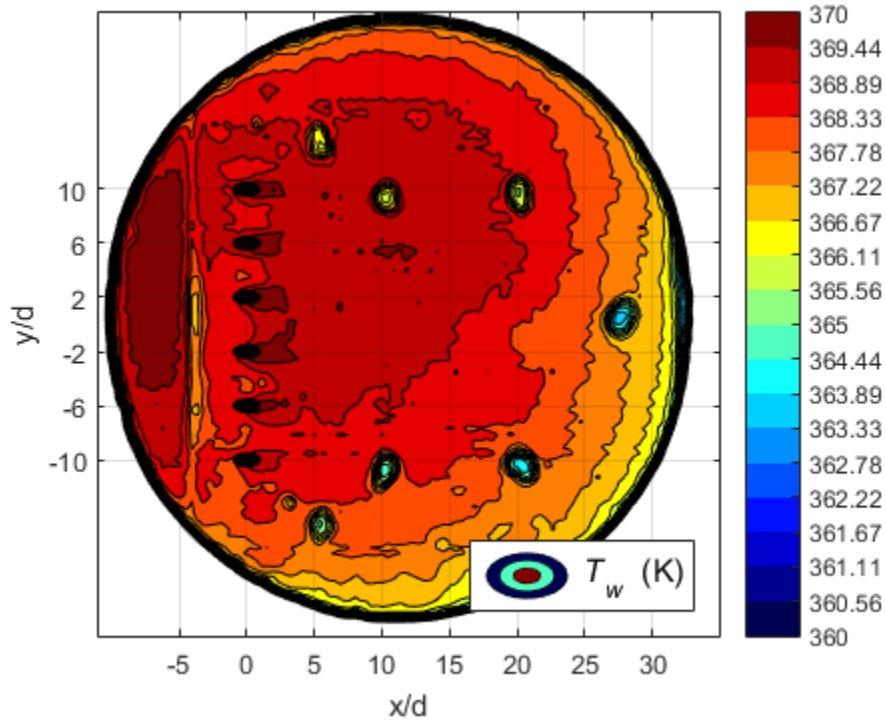


Figure 4-24: Sample argon near ambient condition (#7) surface temperature contours – $M = 1.01$, $DR = 1.52$, $Re_{\infty,D} = 5540$

Similar to the carbon dioxide cooled case of Section 4.3.1.1, a region of reduced T_w immediately downstream of the film cooling holes indicates little external cooling effect. The magnitude of T_w reduction is no more than 1 K. A bulk of the cooling effect appears to be concentrated upstream of and between the film cooling holes ($x/d < 2$) as internal cooling despite the plenum-fed scheme utilized. However, Figure 4-2 and Figure 4-3 indicate that internal coolant impingement likely only occurs for $x/d < -2$. High thermal conductivity values, increased heat transfer along the cooling hole passages, and internal coolant impingement could combine to create the regions of internal cooling effect upstream of and between the cooling holes ($x/d < 2$). As stated in Section 4.3.1.1, the use of a spatially constant T_c value indicates that ϕ contours will reveal all the same trends as T_w contours. For this reason, a ϕ contour plot for this case is omitted.

In order to increase the magnitude of temperature differences in the near ambient flow field, two solutions are apparent. First, replacing the metallic test plate with an identical test plate constructed of a less conductive material would cause the difference between internal and external test plate temperature to increase. However, this change in k without a corresponding test plate thickness change would cause Bi to change as well. Because Bi relates convective heat transfer into the test plate to conductive heat transfer through the test plate, having fixed values for Bi , T_∞ , and T_c ultimately determines the temperature difference between the external and internal test plate surfaces. Thus, changing test plate material is not a good option to increase temperature differences. Second, replacing the foreign gas species with air at an appropriate temperature to maintain DR similitude would increase the energy difference between coolant and freestream flows, resulting in greater temperature differences. Greater temperature differences would be more readily discerned with the instrumentation currently available, providing a greater signal relative to the uncertainty. Both of these courses of action should help increase temperature differences in the flow field to a more readily observed level.

Some analysis was performed to assess the impact of changing the test plate material on the Biot number. Assuming the quarter cylinder at the test plate leading edge may be approximated as a cylinder in crossflow, the correlation shown in Eq. (4-6) [3] was used to estimate the percent change in $\overline{h_f}$ from near ambient to engine representative temperatures.

$$\overline{Nu_D} = 0.3 + \frac{0.62Re_D^{1/2}Pr^{1/3}}{\left[1 + \left(\frac{0.4}{Pr}\right)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re_D}{282000}\right)^{5/8}\right]^{4/5} \quad (4-6)$$

Using thermal conductivity values for Hastelloy X assessed via Eq. (3-11), the estimated $\overline{h_f}$ values from Eq. (4-6), and the test plate thickness, Bi was calculated at near ambient and engine

representative temperatures. Values considered included $Re_D = 3900$; $T_\infty = 1500$ K and $T_w = 1100$ K for engine representative conditions; and $T_\infty = 380$ K and $T_w = 360$ K for near ambient conditions.

It was found that the ratio of high-to-low temperature Bi was 1.28 for the conditions assessed. This Bi ratio was arrived at by dividing high-to-low temperature ratios of h_f and k for Hastelloy X of 2.96 and 2.31, respectively. While the ratio of Bi values is not a perfect match with a value of 1.28, it is a reasonably good match and indicates that h_f decreases nearly threefold and k decreases by nearly the same factor from high to low temperature. Considering the preferred material for matching Bi in near ambient tests, Corian, the conductivity decreases by approximately a factor of 10 in comparison to Hastelloy X ($k \approx 1.06 \frac{W}{m \cdot K}$ for Corian and $k \approx 11 \frac{W}{m \cdot K}$ for Hastelloy X). Thus, a Corian test plate would need to be no more than one-tenth the thickness to ensure Bi similitude. While a lower conductivity test plate material is still desired to increase temperature differences for near ambient tests, it would not be practical due to model size limitations. Initial research indicates that thermally conductive polymers may be able to provide thermal conductivity in the range $5 \frac{W}{m \cdot K} < k < 8 \frac{W}{m \cdot K}$, only requiring a 1.5 to 2 fold thickness increase. Further research to find engineering materials with a suitable thermal conductivity is recommended.

4.4. Matched Engine Representative and Near Ambient Conditions

The following comparisons support the primary objective of matching film cooling performance at near ambient conditions to parametrically similar engine representative conditions. Data collection at near ambient conditions was conducted to match as many parameters as possible to engine representative temperature data points. A summary of

conditions for the engine representative data points and the attempted matches at near ambient conditions is presented in Table 4-16. Each high temperature point, displayed on the right, is accompanied by at least two points at near ambient conditions: one for matched Re_{∞} and one for matched Re_c . Cases with more than two near ambient points failed to accurately match the high temperature conditions with a single test point. Thus, the two near ambient points presented are utilized to determine film cooling performance at the high temperature condition for either the Re_c or Re_{∞} matched case. The numbers listed in the column labeled “#” enable cross-referencing with the experimental conditions tables (Table 4-1, Table 4-2, and Table 4-3 as appropriate) to provide greater detail on experimental conditions.

Table 4-16: Summary of experimental conditions for engine representative and near ambient matched conditions tests

#	Gas	DR (-)	M (-)	I (-)	ACR (-)	#	Gas	DR (-)	M (-)	I (-)	ACR (-)	Match
1	Air	1.56	0.52	0.17	0.43	3	Ar	1.49	0.53	0.19	0.27	Re_{∞}
						2	Ar	1.52	0.52	0.18	0.24	Re_c
2	Air	1.67	0.62	0.23	0.51	41	CO2	1.67	0.59	0.21	0.51	Re_{∞}
						43	CO2	1.67	0.65	0.26	0.57	Re_{∞}
						42	CO2	1.67	0.63	0.24	0.56	Re_c
17	Air	1.57	1.05	0.71	0.88	11	Ar	1.49	1.08	0.79	0.56	Re_{∞}
						8	Ar	1.52	1.02	0.68	0.52	Re_c
						10	Ar	1.52	1.08	0.76	0.55	Re_c
16	Air	1.52	1.05	0.73	0.88	9	Ar	1.49	1.06	0.76	0.52	Re_{∞}
						7	Ar	1.52	1.01	0.67	0.55	Re_c
						13	Ar	1.52	1.09	0.79	0.56	Re_c
27	Air	1.53	1.57	1.62	1.31	20	Ar	1.49	1.59	1.69	0.82	Re_{∞}
						21	Ar	1.52	1.59	1.66	0.82	Re_c
30	Air	1.54	2.08	2.82	1.74	26	Ar	1.50	2.06	2.82	1.06	Re_{∞}
						27	Ar	1.53	2.07	2.81	1.06	Re_c
32	Air	1.55	2.63	4.48	2.20	33	Ar	1.49	2.63	4.65	1.36	Re_{∞}
						34	Ar	1.52	2.64	4.59	1.36	Re_c
11	Air	1.65	1.04	0.66	0.87	56	CO2	1.66	1.09	0.71	0.95	Re_{∞}
18	Air	1.68	1.06	0.67	0.87	53	CO2	1.67	1.05	0.66	0.92	Re_c

4.4.1. Carbon Dioxide Match with Excess Internal Coolant Flow

Of the two gases used for near ambient matching tests, carbon dioxide provided the best agreement with high temperature results in terms of effectiveness magnitude. Table 4-17 presents experimental conditions for the near ambient match of engine representative Point 2 in Table 4-16. Good agreement among all cases is observed for all film cooling parameters presented except ACR . While Greiner et al. [43] computationally demonstrated that ACR may have a significant impact on η , ACR impact on ϕ was not discussed. For each of the Reynolds number matching cases, the specified Re to be matched agrees reasonably well with the engine representative condition. The Reynolds number based on internal channel hydraulic diameter, Re_{D_h} , is also included to compare flow conditions in the internal channel.

Table 4-17: Comparison of experimental conditions for $M = 0.62$ with 3.5 rows excess coolant flow

Condition	Match	Coolant Gas	DR	M	I	ACR	Re_c	Re_∞	Re_{c,D_h}
Combustion	N/A	Air	1.67	0.62	0.23	0.88	181	204	219
Ambient	Re_∞	CO_2	1.67	0.59	0.21	0.51	156	206	191
Ambient	Re_∞	CO_2	1.67	0.65	0.26	0.57	173	205	191
Ambient	Re_∞ avg	CO_2	1.67	0.62	0.23	0.54	164.5	205.5	191
Ambient	Re_c	CO_2	1.67	0.63	0.24	0.56	196	239	224

Per the discussion of Albert and Bogard [7] found in Section 2.6.2, matching Re_∞ and Re_{D_h} is critical to ensure similitude in the h_f/h_i ratio. The data of Table 4-17 clearly demonstrate that none of the near ambient matching cases simultaneously match both Re_∞ and Re_{D_h} . Thus, it is asserted that the ratio of external to internal convective heat transfer is not correctly matched for any of the cases of Table 4-17. Additionally, the Bi term in Eq. (2-44) underscores the importance of correctly matching the ratio of external convective heat transfer to conductive heat transfer through the test plate. Because T_{aw} and h_f are not known for the near

ambient conditions, Bi similitude cannot be directly assessed. However, correctly matching Re_∞ should provide the best possible match without direct calculation. Further, the analysis and discussion at the end of Section 4.3.1.2 indicate that Bi should be matched within approximately 30% as long as Re_∞ and η are matched between the two cases. While Re_∞ and η cannot be said to have been perfectly matched, conditions were matched as best as possible, indicating that the Bi values for the high and low temperature conditions should not differ greatly.

Keeping the mild disagreement in various Re values from Table 4-17 in mind, ϕ contours for the data points of Table 4-17 are presented in Figure 4-25. Near ambient results with carbon dioxide coolant include both Re_∞ cases (upper right and lower left) and the Re_c match (lower right). The near ambient results match engine representative results better than anticipated. Effectiveness magnitude is better matched at near ambient conditions for the Re_∞ match than the Re_c match, but neither matches the trends of the high temperature result very well. The two Re_∞ matching cases agree well because little was changed between the two points and they were collected consecutively on a single day. However, both near ambient Re_∞ matching cases are displayed, highlighting the minimal impact caused by changing M by 0.06. Further analysis using $\bar{\phi}$ averaged over the middle two pitches follows to provide better insight on the magnitude of error.

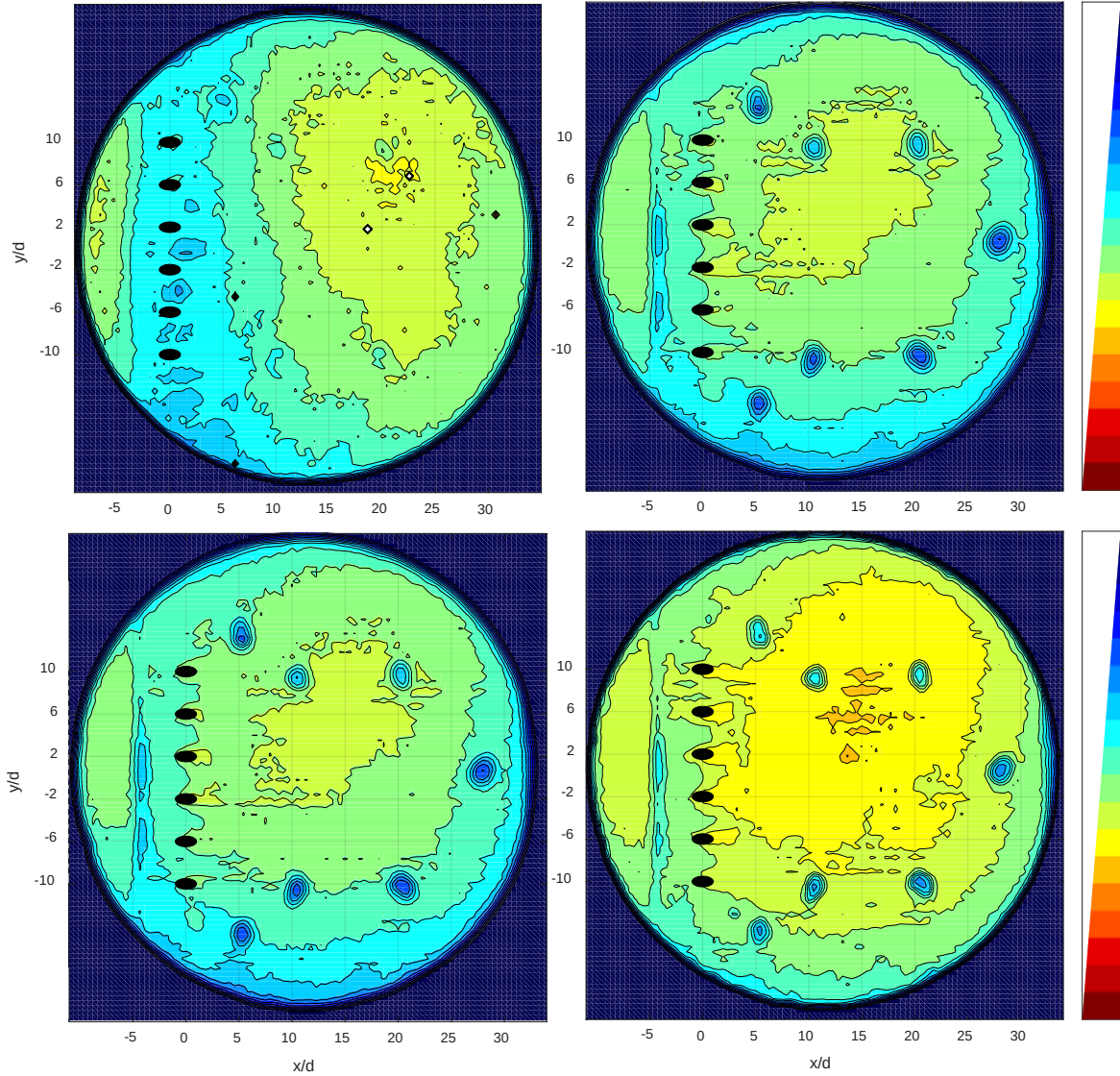


Figure 4-25: Overall effectiveness contours for $M = 0.62$, $DR = 1.67$ with 3.5 rows excess coolant flow: combustion (top left) and near ambient conditions, Re_{∞} match (top right, lower left) and Re_c match (lower right)

While the difference in $\bar{\phi}$ is only 0.05 at a maximum, Figure 4-26 shows that the trends in $\bar{\phi}$ for the near ambient conditions are still far from matching the high temperature result. Oddly, the near ambient data presented in Figure 4-26 would likely be judged more correct or accurate according to expected trends if the curves were inverted. However, as demonstrated in Figure 4-22, Figure 4-24, and the accompanying discussion of Section 4.3.1, the measured surface temperature exhibits the same trends shown in both ϕ plots presented. Thus, there is

most likely an error in T_w due to radiation reflected off the sapphire window causing elevated measured emission and thus apparent temperature in the middle of the window. This artificially increased T_w would cause a decrease in ϕ that could dissipate towards the edges of the viewing window, resulting in the observed increase in ϕ with x/d . Additionally, the Re_∞ matched near ambient cases better match the high temperature magnitude than the Re_c matched case.

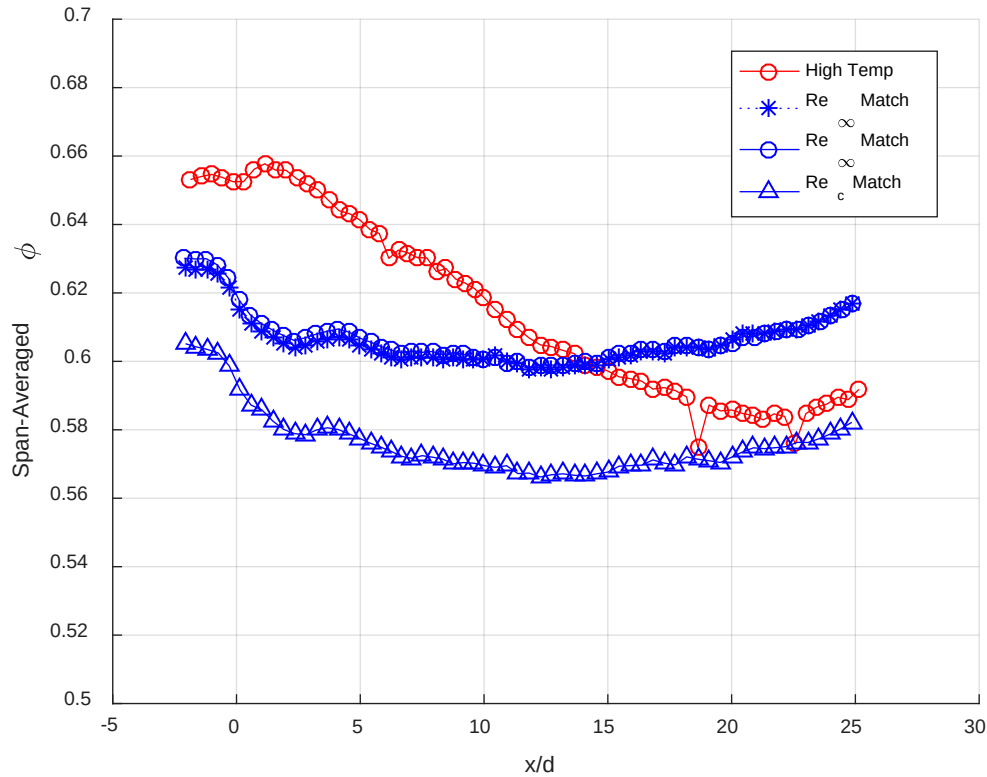


Figure 4-26: Span-averaged overall effectiveness for $M = 0.62$, $DR = 1.67$ with 3.5 rows excess coolant flow

The repeatability analysis of Section 3.8.3.2 indicates a significant difference (potentially up to $\Delta\phi = 0.02$) could exist between the Re_∞ and Re_c matching cases because those data points were collected on separate days. Additionally, the previously mentioned differences in various Reynolds numbers may play a significant role in the results of Figure 4-25 and Figure 4-26. The Re_c matching case featured a 17% increase in Re_∞ in order to maintain M and I similitude while matching Re_c . This Re_∞ increase would serve to increase h_f and thus heat transfer into

the test plate, resulting in the observed reduction in ϕ . Similarly, the Re_∞ matching case featured a 10% decrease in Re_c and a 13% decrease in Re_{c,D_h} to maintain M and I similitude while matching Re_∞ . A reduction in Re_c has implications for the external flow field, likely resulting in reduced η , while the reduced Re_{c,D_h} value would reduce h_i . Both of these effects would serve to reduce ϕ , potentially causing the observed disagreement between near ambient and engine representative data.

4.4.2. Argon Match with Plenum-Fed Scheme

Table 4-18 displays experimental conditions for high temperature point #16 and the accompanying near ambient matched conditions ($M = 1.05$, $DR = 1.52$ with a plenum-fed cooling scheme). The data shown indicate good agreement for film cooling parameters (DR , M , and I) and good agreement for the intended Reynolds number to matched. Cases matching Re_c provided a much better Re_{D_h} match than for the Re_∞ matched case. Due to argon's nature as a monatomic gas, ACR was not well matched between the high temperature and near ambient conditions. This is to be expected for all argon cooled cases if ACR is not specifically selected as a parameter of interest.

Table 4-18: Comparison of experimental conditions for $M = 1.05$ plenum-fed cooling

Condition	Match	Coolant Gas	DR	M	I	ACR	Re_c	Re_∞	Re_{D_h}
Combustion	N/A	Air	1.52	1.05	0.73	0.88	291	203	33
Ambient	Re_∞	Argon	1.49	1.06	0.76	0.55	184	202	21
Ambient	Re_c	Argon	1.52	1.01	0.67	0.52	259	296	30
Ambient	Re_c	Argon	1.52	1.09	0.79	0.56	273	288	31
Ambient	Re_c avg	Argon	1.52	1.05	0.73	0.54	266	292	30.5

Overall effectiveness contour plots for the experimental conditions of Table 4-18 are displayed in Figure 4-27. The high temperature result is displayed upper left while the near

ambient Re_∞ matched result is upper right and the two results for matched Re_c occupy the lower row. It is immediately apparent that none of the argon-cooled near ambient cases match the high temperature case in terms of magnitude or profile. One possible cause for this mismatch could be incorrectly matched heat transfer parameters such as heat transfer coefficients or the Biot number. While a lack of information on h_f for near ambient conditions prevents calculation of Bi , the analysis and discussion of Section 4.3.1.2 indicate that Bi values should be reasonably well matched by matching Re_∞ as done in this case. The ratio of external to internal heat transfer coefficients, h_f/h_i , must also be matched, but a lack of h_f values for near ambient conditions precludes direct assessment of this ratio. Additionally, film cooling parameters such as DR and I may have been incorrectly matched due to inaccurate coolant temperature measurement. Such an inaccuracy could arise from a change in T_c between the point of measurement and the cooling hole inlet. The very low coolant flow rate for the near ambient conditions (≈ 0.5 SLPM) would certainly allow sufficient coolant dwell time to allow T_c to increase, although the magnitude of that increase cannot be commented on with any level of certainty.

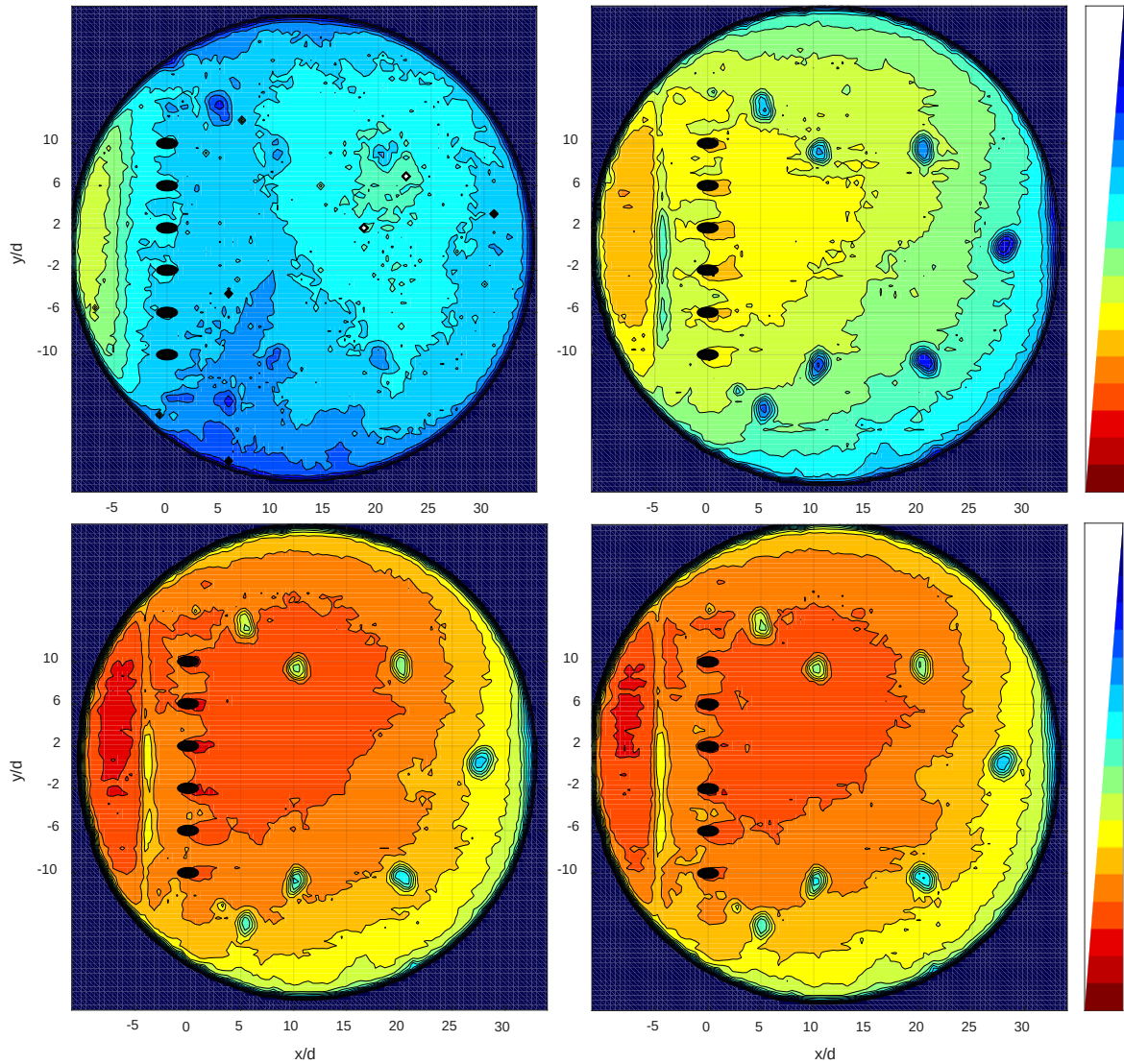


Figure 4-27: Overall effectiveness contours for $M = 1.05$, $DR = 1.52$ with plenum-fed cooling scheme: high temperature (#16, top left) and near ambient conditions, Re_∞ match (#9, top right) and Re_c match (#7, lower left; #13, lower right)

The results of the Re_c match (bottom row of Figure 4-27) agree extremely well. This is because they only differ by a very slight change in film coolant mass flow, and the data points were collected consecutively on a single day. The Re_∞ matching case was collected on a different day, suggesting that repeatability issues may play a significant role in the observed ϕ variations. The best available explanation for the difference between the Re_∞ and Re_c cases lies in the various Reynolds numbers presented in Table 4-18. While Re_∞ is greater for the Re_c

matching cases, Re_{D_h} is also greater. The simultaneous increase in these two critical Reynolds number indicates that the Re_c cases should experience greater heat flux into the test plate. Increased heat flux could increase the difference between internal and external surface temperatures, resulting in decreased effectiveness.

This poor match is further reinforced in the span-averaged results of Figure 4-28. The Re_∞ matched near ambient case matches the high temperature result magnitude better than the Re_c matched cases, but neither correctly matches the overall profile of $\bar{\phi}$ with increasing x/d . The trends of increasing effectiveness with increased x/d and minimal cooling effect are also very apparent. However, unlike the trends observed in the T_w contour plot of Figure 4-24, it is apparent that there is a slight rise in effectiveness shortly downstream of the cooling holes ($x/d \approx 2.5$). For the Re_c matching cases (circles and triangles), effectiveness shows a slight decay following this rise consistent with expected trends. These two points were collected in immediate succession with only a 0.08 change in M , so the ϕ values are expected to differ very little. The Re_∞ matched near ambient case still matches the high temperature ϕ magnitude than the Re_c matched cases. This appears to be true even when changing coolant gas species.

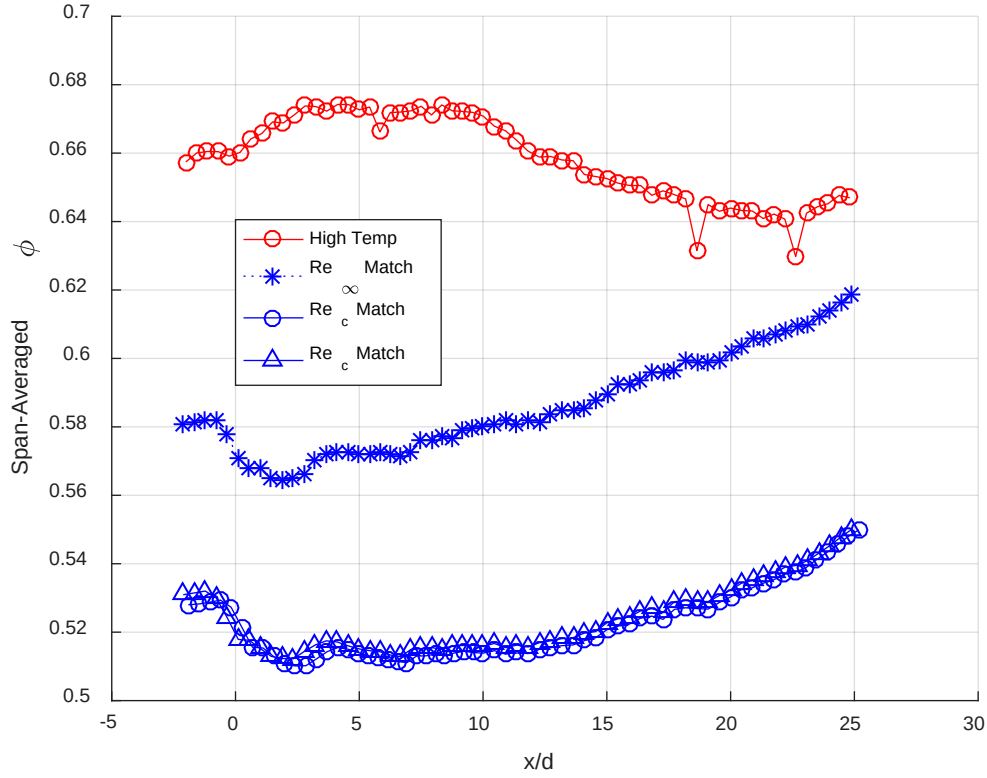


Figure 4-28: Span-averaged overall effectiveness for $M = 1.05$, $DR = 1.52$ with plenum-fed cooling scheme

The most likely cause for the disagreement between the Re_∞ and Re_c cases lies in the Re_∞ values of the two cases. A 30 % increase in Re_∞ was required in order to maintain M and I similitude while matching Re_c . As discussed with the CO_2 matching case of Section 4.4.1, this would serve to increase h_f and thus heat transfer into the test plate. On the other hand, because Re_{c,D_h} was fairly constant for all the near ambient cases considered, h_i should also be relatively constant for all cases. Such an increase in h_f while maintaining a constant h_i value would result in decreased ϕ values as observed. While all relevant Re values cannot be simultaneously matched, it is possible to match the ratio of external to internal heat transfer coefficients by providing excess coolant flow for the higher Re_∞ cases. This approach is recommended for future testing to ensure similitude in the ratio of external to internal convective heat transfer, thus ensuring similitude in ϕ .

While preliminary analysis was performed for additional matched near ambient and engine representative conditions presented in Table 4-16, the results presented in Section 4.4 accurately represent the entire range of cases in Table 4-16. Additionally, matched near ambient and engine representative conditions featured a myriad of issues. Chief among these issues was the observed trend of increasing ϕ values with x/d for near ambient conditions. This troubling trend most likely stems from potential background radiation, optical distortion, and IR window reflection issues discussed in Section 4.3.1.1. Retroactive corrections cannot be accomplished to account for these particular phenomena. Thus, considering the questionable data quality for near ambient results and the representative nature of results presented in Section 4.4, additional cases from Table 4-16 are omitted.

4.5. PSP Results

Development of the PSP method for use in the FCR is critical to achieve the objectives of investigating physical model size in scaling η and making ϕ and η predictions. In addition to investigating model size effects, comparison with data acquired on a large scale model in the Leading Edge Film Cooling Tunnel serves to assess FCR flow quality. As discussed in Section 3.5, the LEFCT model used by Wiese et al. [39] and the one row showerhead test plate are nearly perfect scale models, differing only in cooling hole length-to-diameter ratio. Thus, the one row showerhead test plate was utilized for PSP investigations.

As employment of the PSP method was developed for the FCR, several data sets were collected using a variety of coolant gas species and varied experimental conditions. A representative sampling of those conditions is presented in Table 4-19. Data collected in the LEFCT with the model described in Section 3.5 (identical to that used by Wiese et al. [39]) are listed at the bottom of the table and will be used for comparison. Because the cooling uniformity

issue described in Section 3.4.1 was not resolved until after PSP testing was completed, the results obtained presented uneven coolant distributions across the test plate span. To help mitigate this uneven distribution, all but the three center holes on the one row showerhead test plate were blocked. As such, conditions presented will focus on the three cooling holes with inlets nearest the test plate centerline.

Table 4-19: Summary of PSP experimental conditions

FCR Data								
Point	Coolant Gas	DR	M	I	ACR	Re_c	$Re_{\infty,d}$	$Re_{\infty,D}$
1	CO ₂	1.55	0.94	0.57	0.77	585	502	9385
2	CO ₂	1.55	1.57	1.60	1.34	993	507	9485
3	Argon	1.39	0.90	0.58	0.47	374	506	9454
4	Argon	1.39	1.49	1.59	0.77	624	511	9557
5	N ₂	0.97	0.73	0.55	0.76	384	506	9453
6	N ₂	0.97	1.26	1.62	1.30	663	509	9516
7	He	0.14	0.28	0.55	0.001	133	515	9630
8	He	0.14	0.49	1.69	0.003	231	509	9523
LEFCT Data								
9	CO ₂	1.56	0.87	0.49	0.73	585	530	9911
10	CO ₂	1.57	1.51	1.45	1.26	1011	530	9902
11	Argon	1.42	0.86	0.52	0.45	383	532	9953
12	Argon	1.42	1.44	1.46	0.75	638	532	9941
13	N ₂	0.99	0.70	0.49	0.72	399	540	10106
14	N ₂	0.99	1.22	1.50	1.26	696	540	10098
15	He	0.14	0.27	0.50	1.38	134	529	9888
16	He	0.14	0.46	1.50	2.39	233	528	9883

Figure 4-29 displays η contours for PSP conditions measured with CO₂ coolant, $I = 0.57$ (#1) and 1.60 (#2), $Re_{\infty,D} \approx 9400$, and two rows excess internal coolant flow. The two rows excess coolant flow were based on supplying two more rows of three holes each with coolant for the appropriate momentum flux ratio. A dramatic increase in coolant coverage is noted with increased I downstream of $x/d = 10$. Peak η magnitudes do not seem to vary greatly between the two cases. This is not surprising because η should vary from a minimum value of 0 in the

freestream up to a maximum value of 1 at the cooling hole exit. Because PSP functions as an oxygen sensor, a region of η values approaching the maximum possible value of 1 should be visible somewhere in the coolant hole or along the test plate regardless of the coolant flow rate. The $I = 1.60$ case also appears to have stronger jet penetration in the negative y/d direction. This is consistent with the greater amount of momentum in the coolant jets, penetrating further into the freestream flow before being turned along the test plate.

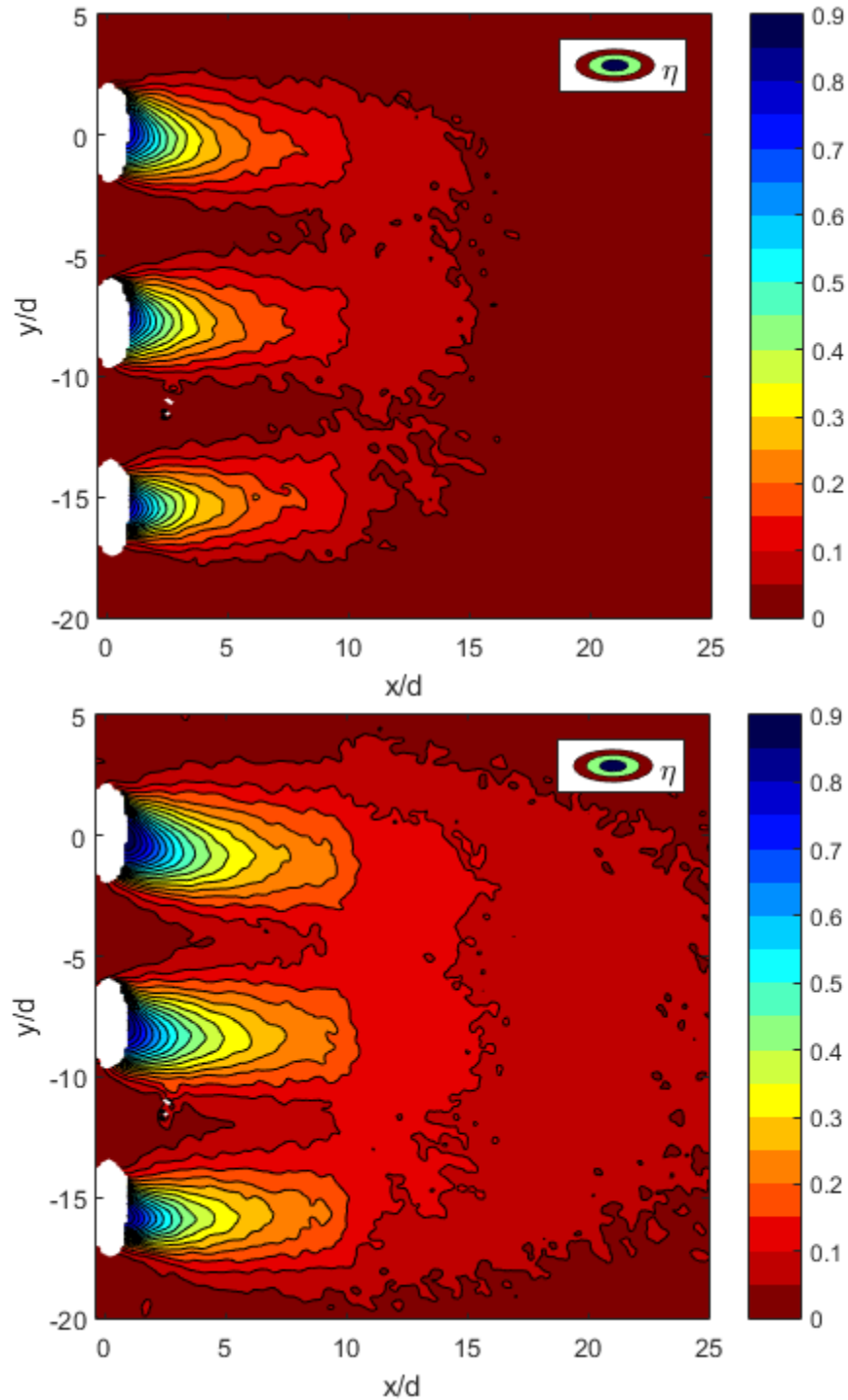


Figure 4-29: FCR adiabatic effectiveness contours for CO₂, $I = 0.57$ (#1), 1.60 (#2)

Contours for η data collected with CO₂ coolant at matched conditions in the LEFCT are presented in Figure 4-30. The point of termination for cooling effect is not visible within the acquired field of view, but η does appear to decrease to near 0 at $x/d = 9$. This is consistent

with the FCR data of Figure 4-29 for which η decreases to $\eta < 0.1$ for $x/d > 10$. However, the regions of peak η differ in both size and location. The LEFCT data display increased coolant jet penetration with much narrower coolant traces compared to the FCR data. Increased coolant jet penetration resulted in coolant traces located at smaller y/d values. Increased width for FCR coolant traces may arise from higher freestream turbulence enhancing coolant mixing with freestream flow. However, no data has been collected to examine FCR turbulence levels, so no quantitative comparison can be made between the two facilities. Additionally, near-hole peak η values appear much larger for the LEFCT data than the FCR data. This may also be attributed to elevated turbulence levels in the FCR as compared to the LEFCT.

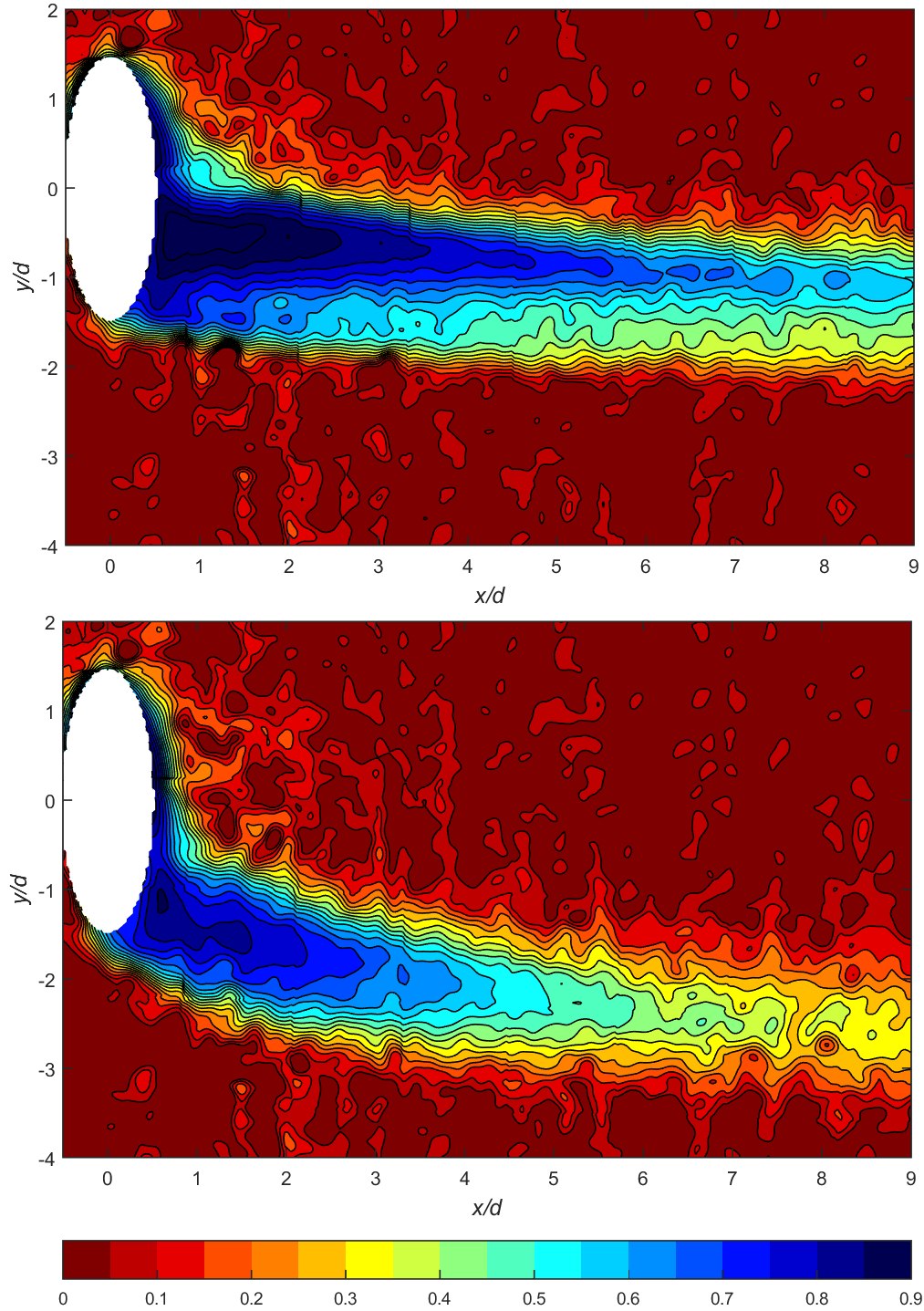


Figure 4-30: LEFCT adiabatic effectiveness contours for CO_2 , $I = 0.49$ (#9), 1.45 (#10)

Adiabatic effectiveness contours for representative argon cooled conditions are shown in Figure 4-31. Experimental conditions featured argon coolant, $I = 0.58$ (#3) and 1.58 (#4), $Re_{\infty,D} \approx 9500$, and two rows excess internal coolant flow. The definition of excess coolant flow

is consistent with that used for the CO_2 cases above. The trends described previously for the CO_2 cases of Figure 4-29 are all still present, but a noticeable skew in η towards holes at greater y/d values is also present. This is consistent with the preferred coolant flow direction observed prior to coolant baffle plate installation. Comparing the argon cooled cases of Figure 4-31 to the CO_2 cooled cases of Figure 4-29, the argon cases appear to achieve both better coolant coverage and greater peak η values. This runs counter to the expectation that η should increase with increased DR or increased M as experienced when changing from carbon dioxide to argon injection. However, many other factors could affect coolant spreading such as turbulence intensity and diffusion coefficients. Further, effects not yet considered such as repeatability error or coolant jet dynamic pressure could have a significant impact on calculated η values.

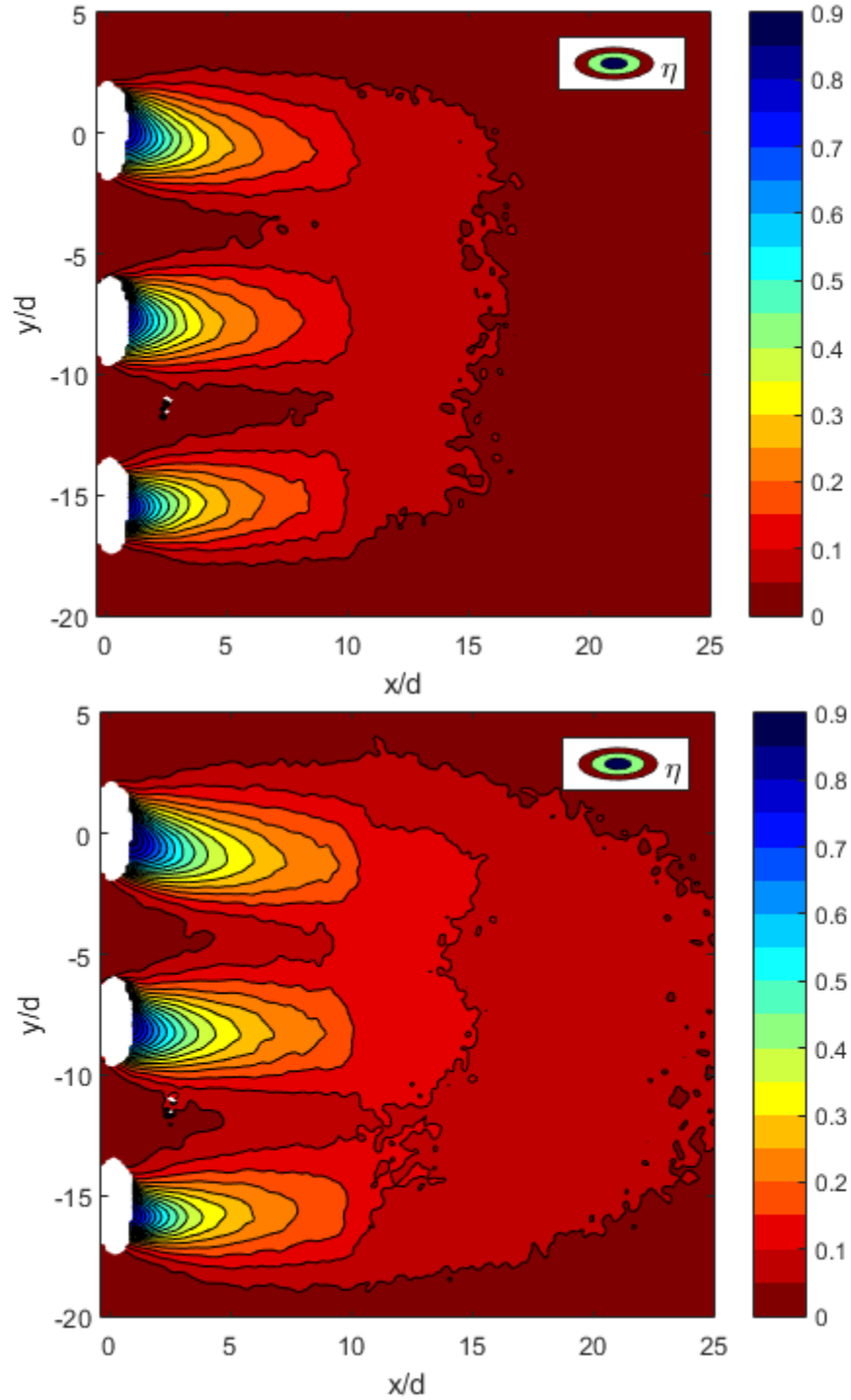


Figure 4-31: FCR adiabatic effectiveness contours for argon, $I = 0.58$ (#3), 1.58 (#4)

Argon cooled η contours from the LEFCT matching the conditions of Figure 4-31 are presented in Figure 4-32. Compared to the FCR data of Figure 4-31, the $I = 0.52$ case features much greater η values over $0 < x/d < 9$, but the $I = 1.46$ case matches the rate of downstream η

decay reasonably well. Elevated freestream turbulence could be responsible for the more rapid decay in η for the FCR $I \approx 0.5$ data and is most likely responsible for the broad coolant traces present in the FCR data. The top cooling hole for the FCR $I = 1.58$ condition features coolant jet penetration comparable to that for the LEFCT $I = 1.46$ condition, but the other two holes for the FCR data appear to lack the jet momentum necessary to penetrate towards negative y/d . The LEFCT $I \approx 1.5$ case also differs from the FCR data in that it clearly presents coolant jet separation. Jet reattachment is not apparent in the data presented.

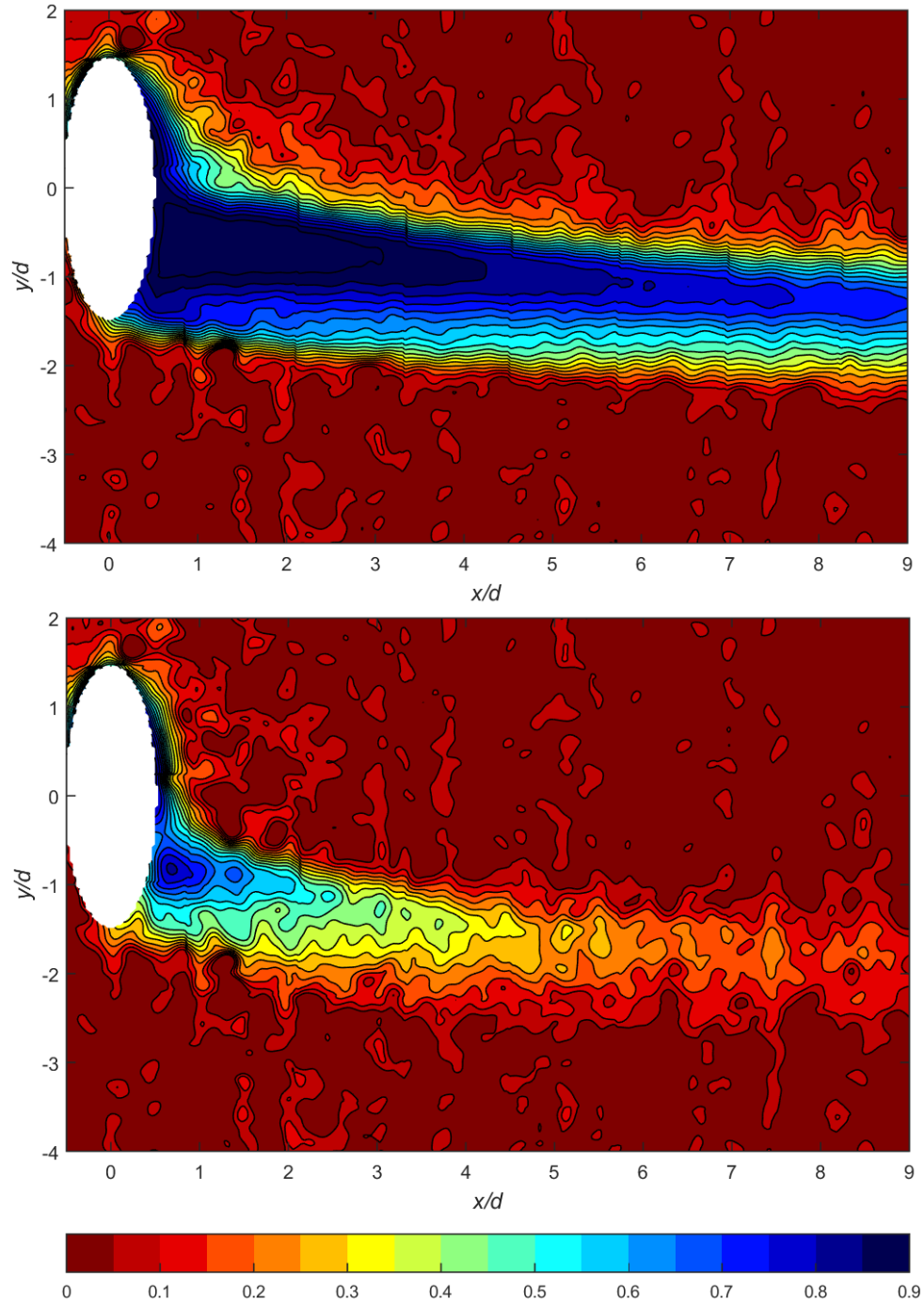


Figure 4-32: LEFCT adiabatic effectiveness contours for argon, $I = 0.52$ (#11), 1.46 (#12)

Nitrogen cooled η contours are displayed in Figure 4-33 for experimental conditions of $I = 0.56$ (#5) and 1.62 (#6), $Re_{\infty,D} \approx 9500$, and two rows excess internal coolant flow. The data reveal a trend of decreasing η with decreasing DR , consistent with published results.

Additionally, the nitrogen cooled cases provide much less cooling coverage, particularly for the $I \approx 0.6$ case. Instead of reaching out to $x/d = 15$, the cooling effect is limited to $x/d \leq 10$. Peak η values also decrease compared to the CO_2 and argon cases. A slight coolant jet skew towards positive y/d is also present.

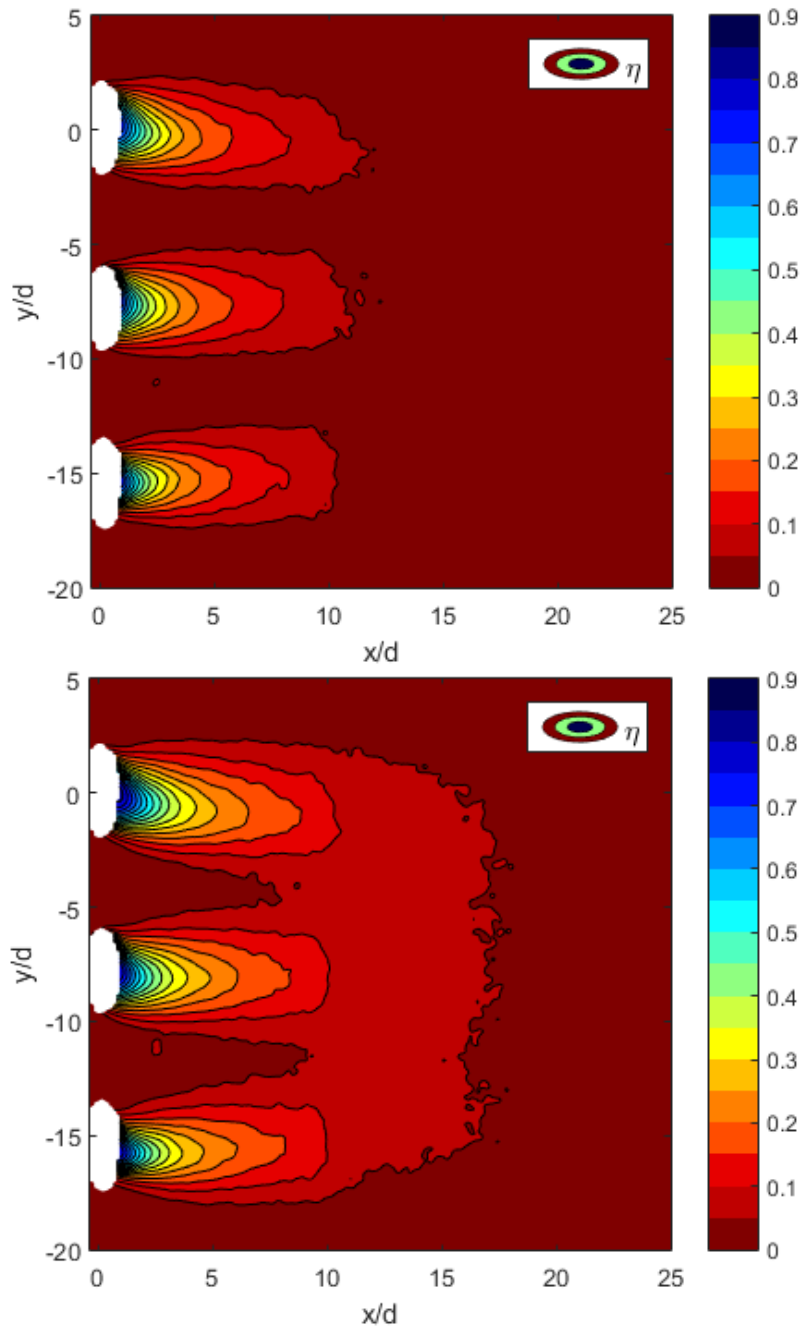


Figure 4-33: FCR adiabatic effectiveness contours for nitrogen, $I = 0.56$ (#5), 1.62 (#6)

Cases collected in the LEFCT at conditions matching those of Figure 4-33 are presented in Figure 4-34. Similar to the argon cooled cases of Figure 4-31 and Figure 4-32, the LEFCT $I \approx 0.5$ case features a strongly cohesive coolant jet producing larger η values than the FCR over the observed range ($0 < x/d < 9$). Increased coolant jet penetration for LEFCT data persists for these cases as well, particularly for $I \approx 1.5$. The $I \approx 1.5$ cases agree fairly well in terms of η magnitude and the rate of downstream η decay. The broader coolant jets present in the FCR data are again attributed to elevated freestream turbulence levels in the FCR.

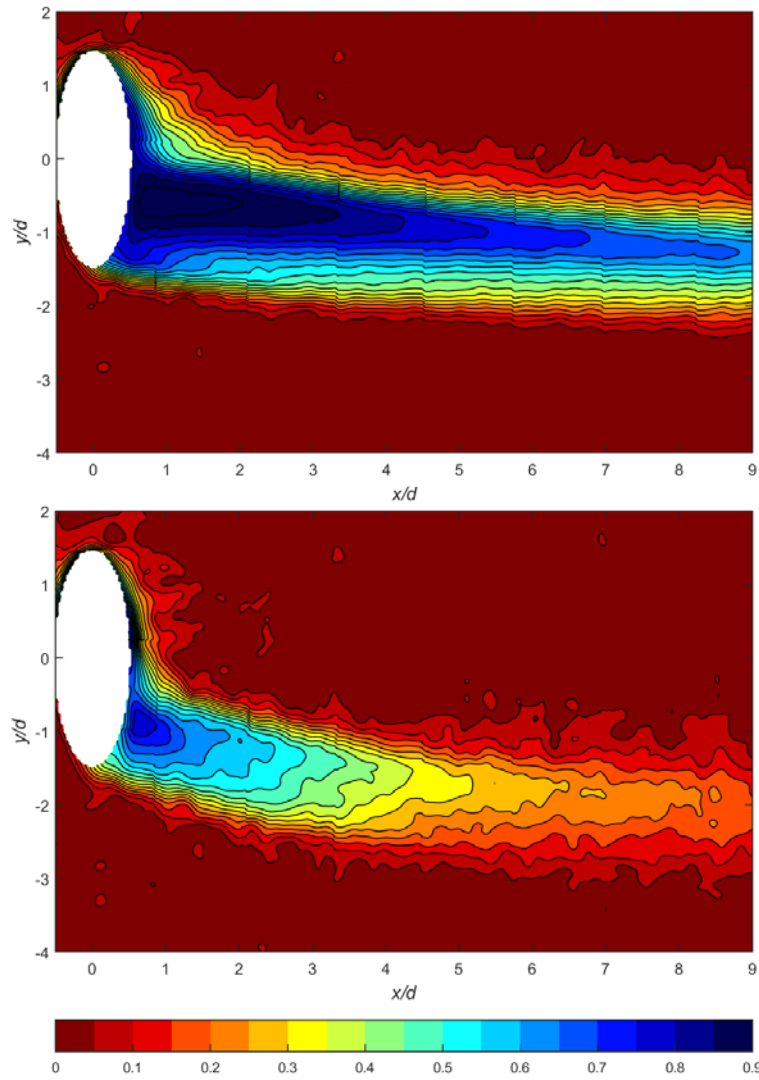


Figure 4-34: LEFCT adiabatic effectiveness contours for nitrogen, $I = 0.49$ (#13), 1.50 (#14)

Helium cooled η contours are presented in Figure 4-35 for experimental conditions $I = 0.55$ (#7) and 1.69 (#8) and $Re_{\infty,D} \approx 9600$ as shown in Table 4-19. The lack of coolant effect for these two cases is striking. It is possible that the greatly reduced blowing ratio due to the very low molecular weight for helium could play a role. The M values (Table 4-19) for the helium cases are 0.28 and 0.49 compared to 0.73 and 1.26 for the nitrogen cases. As molecular weight is greater for argon and carbon dioxide, M would be greater than for nitrogen and should result in the greatest M values. Also, the decrease in η with increased I is surprising. For the greater I values considered (≈ 1.6), the coolant jet could lift off from the surface, but jet lift off should be present for all gases considered if present at all. The M values of 0.28 and 0.49 may explain the reduced magnitude of cooling effect, but the I values of 0.56 and 1.69 indicate the coolant jet placement should match fairly well with cases for the other gases.

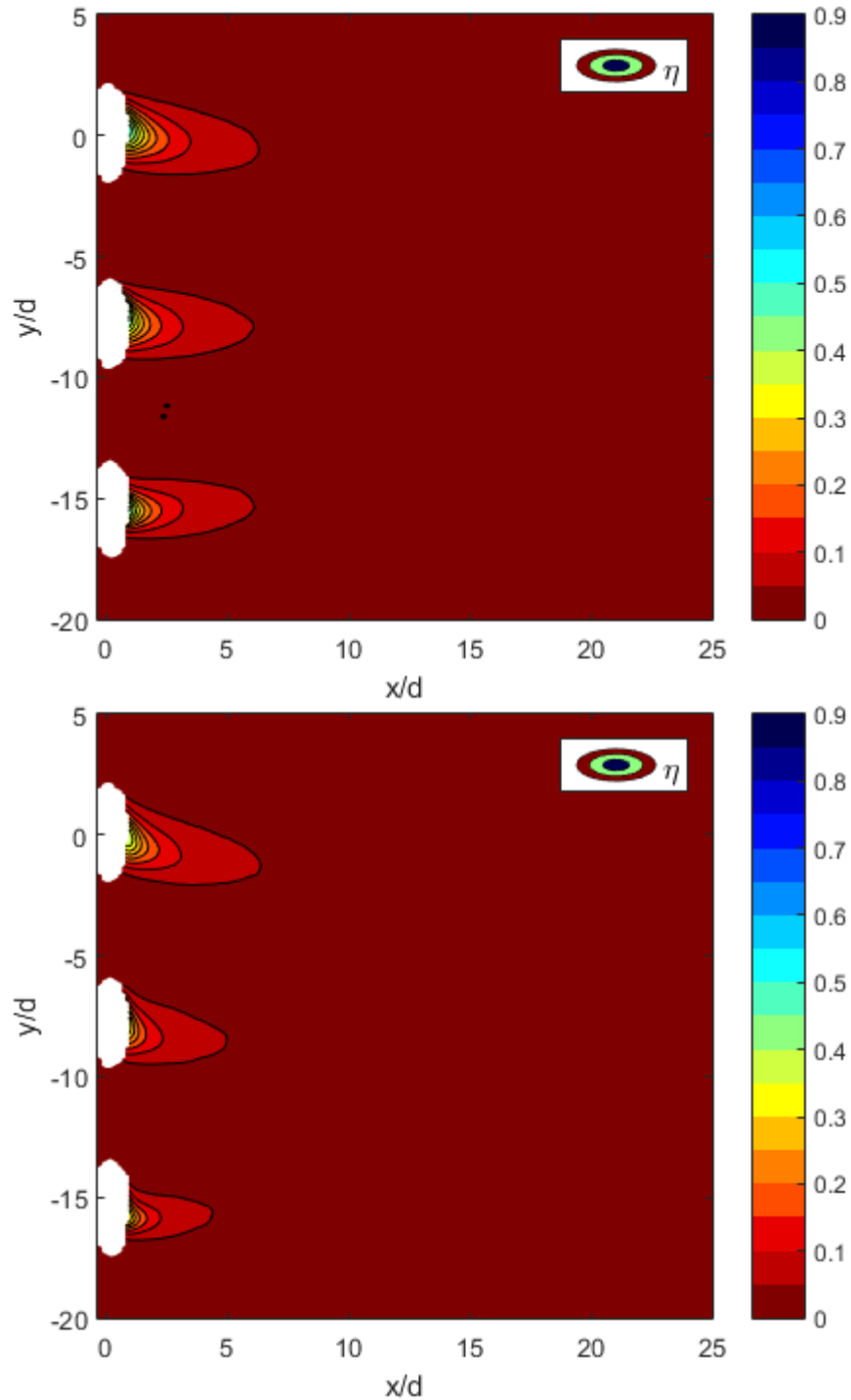


Figure 4-35: FCR adiabatic effectiveness contours for helium, $I = 0.55$ (#7), 1.69 (#8)

The most likely explanation for the η contours of Figure 4-35 is lack of coolant flow. This lack of flow could arise due to an error in mass flow control, a physical blockage in the coolant feed line, or, most likely, a flow leakage somewhere in the coolant gas feed system. Due

to its nature as a monatomic gas with a very small atomic diameter, helium may escape through gaps too small for larger gas molecules, such as diatomic oxygen and nitrogen, to pass through. While the exact locations of such leaks are unknown, they could be ascertained using a leak detection method such as soapy water application. Leaks within the test section could also be detected using PSP to detect oxygen displacement near a leak. Areas suspected for leakage include any tubing connections, the point of coolant entry into the main cooling block, the interface between the test plate and coolant baffle or main cooling block, and any blocked cooling holes.

However, considering η contours from matched conditions collected in the LEFCT (shown in Figure 4-36), it seems more likely that the observed η contours from Figure 4-35 are accurate. While the LEFCT $I \approx 0.5$ case still features greater peak η values and a slower rate of downstream η decay, the increased rate of η decay for the FCR data likely results from elevated freestream turbulence. Additionally, the $I \approx 1.5$ cases agree reasonably well between the two experimental rigs. Both feature cooling effect essentially limited to $0 < x/d < 5$ and indicate a severely separated coolant jet. With this reasonably good agreement in mind, the previously hypothesized flow leaks may not be as large a concern as initially thought. However, the FCR should still be examined for flow leakage any time the rig is dismantled or tubing connections are detached and reattached.

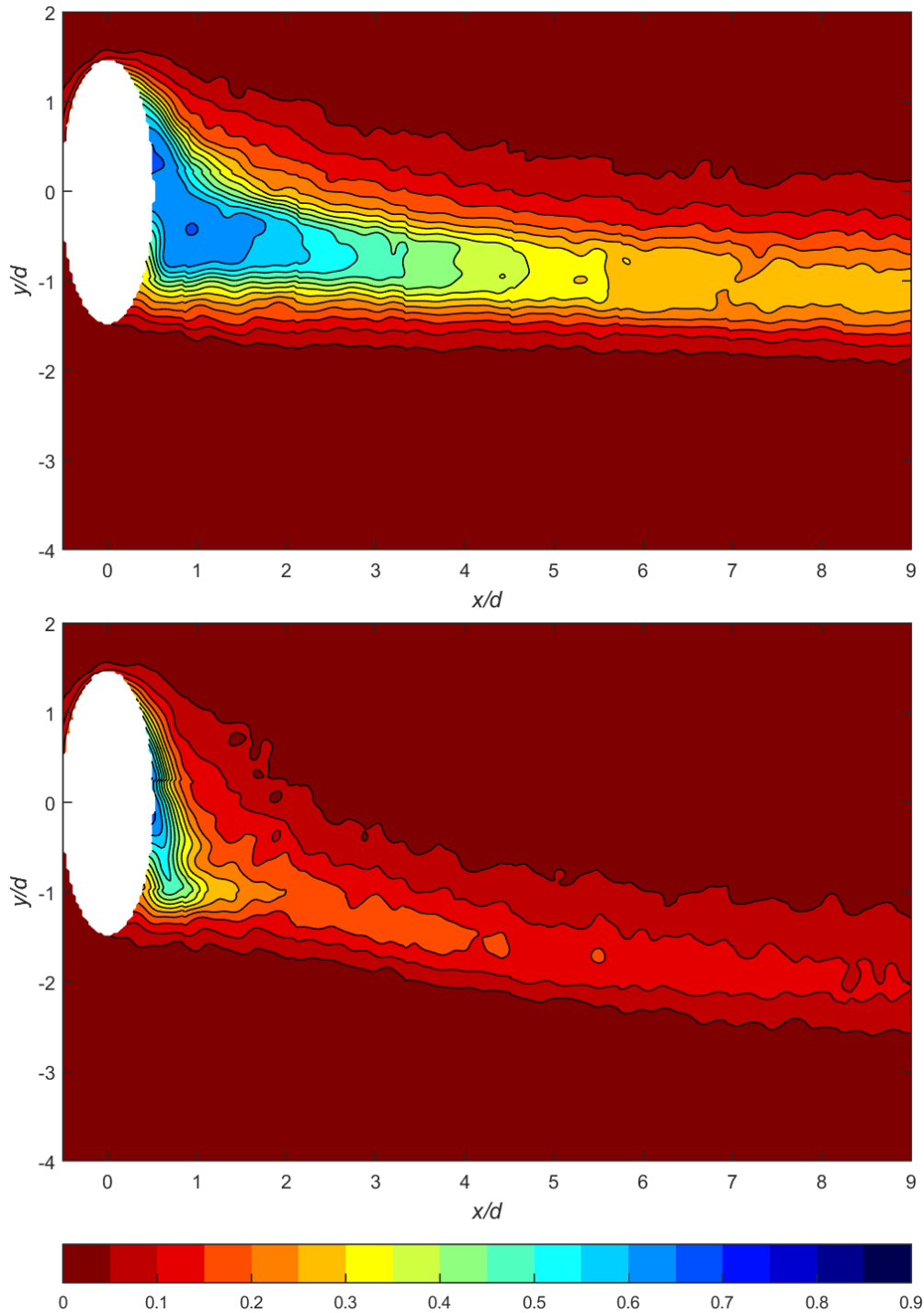


Figure 4-36: LEFCT adiabatic effectiveness contours for helium, $I = 0.50$ (#15), 1.50 (#16)

The PSP results presented here are promising and invite an array of new investigations to better understand both the PSP method and FCR performance during PSP testing. However, it is

noted that the primary goal of the recommended investigations is to optimize FCR operation to eliminate confusion from future studies on film cooling behavior. These investigations include assessing the effect of coolant jet dynamic pressure on η , evaluating the effect of elapsed time between wind on and wind off images, and determining the cause of the apparent lack of coolant for helium cooled cases. Investigations of coolant jet dynamic pressure and image collection timing will both provide valuable information when quantifying experimental uncertainty for the PSP method. Further, the current camera and lens setup only provides spatial resolution of about 10 pixels per hole diameter. Adjusting the setup to provide greater spatial resolution would provide greater insight into flow phenomena as well as minimize the wasted space in images. Finally, placing the camera at an angle may allow a better view of the coolant holes and test plate leading edge. While not critical, these setup adjustments could help in making comparisons with data obtained from the AFRL Leading Edge Film Cooling Tunnel. All recommended future actions will be required to accomplish the objectives of evaluating physical model size effects and making effectiveness predictions.

5. Conclusions

Because the AFIT FCR is a fairly new test rig with relatively little experimental background, the data collected throughout the course of this investigation provided many valuable insights about its performance. Many of these discovered phenomena present challenges that must be overcome or at least understood for meaningful, novel work to proceed. Conclusions are presented based on the objectives of investigating physical model size effects on scaling adiabatic effectiveness and scaling film cooling performance between near ambient and engine representative conditions. As a subset of these objectives, experimental capability development was also a heavy emphasis area. Recommendations for future work follow with an emphasis on a path forward to complete the overall objectives of this study.

Physical model size effects on adiabatic effectiveness were assessed by measuring η distributions on scaled models in the AFIT FCR and AFRL LEFCT using the PSP method. The PSP method had never before been used in the FCR, so development of that experimental capability was required. Scaling film cooling performance between near ambient and engine representative conditions was investigated in the FCR using IR thermography to measure model surface temperatures and then calculate ϕ distributions. Measuring ϕ at near ambient conditions in the FCR was also a previously unexplored test method. Thus, experimental capability development was critical to both the objectives of investigating physical model size effects on η and scaling ϕ between near ambient and engine representative conditions.

5.1. Engine Representative Conditions

Experiments were conducted on a representative turbine airfoil leading edge model consisting of a quarter cylinder followed by a flat section and sloped after body at $Re_{\infty,D} = 3800$

with varied DR and M values and using air as the film coolant gas. A single row of six fan-shaped, laidback film cooling holes oriented in the streamwise direction was located on the flat section of the test plate and inclined 30° to the surface. The fan half angle and layback angle were both 10 degrees. Engine representative conditions included freestream temperatures between 1460 and 1500 K, affording a range of DR values between 1.5 and 1.8. IR thermography was used to determine test plate surface temperatures and then calculate overall effectiveness. Two primary parameters were varied for engine representative conditions: blowing ratio and internal coolant mass flow rate.

Thermal data obtained via IR thermography at engine representative conditions served as a baseline to evaluate film cooling behavior, both globally and specifically in the FCR. These data points were also used as a baseline when attempting to match film cooling performance in the form of ϕ between engine representative and near ambient conditions. A lack of thermal control caused by setup-specific heat transfer issues resulted in significant T_c and DR variations when varying blowing ratio and internal coolant mass flow rate.

For all conditions examined, ϕ was found to increase slightly well downstream of the film cooling holes ($x/d > 20$). While this result runs counter to expectations, it is consistent with FCR geometry, dimensions, and heat loss to surroundings. As internal coolant flow rate was increased, coolant temperature decreased. The combined effect of increased internal coolant flow and decreased coolant temperature yielded increased heat flux and decreased surface temperatures as expected. However, the cases with lower surface temperature also exhibited lower overall effectiveness values. This trend occurred over the full range of blowing ratio considered. Although counter to expectation, most of these results did not exceed the overall effectiveness uncertainty and technically cannot be said to differ. These strange trends further

reinforce the need for a thorough repeatability analysis. Additionally, a new experimental procedure is recommended wherein all test points for a single comparison should be collected in succession on a single day of testing. Further, the order of data point collection should be examined to ensure no hysteresis within the data.

An internal coolant flow sweep also revealed a trend of decreasing ϕ with increased excess internal coolant mass flow. Both heat flux and internal heat transfer coefficient increased as expected with excess internal coolant mass flow. However, density ratio was not held constant for all cases included in the comparison. While the comparison was useful in that it accurately represents heat transfer in a real system, too many parameters are varied. Thus, even though the $\bar{\phi}$ results differ conclusively, the net effect cannot be attributed to change in any single parameter.

Estimates of T_{aw} and h_f revealed that small differences in T_w and q'' are not significant enough to generate useful or accurate T_{aw} estimates. The spread in T_w was severely limited by the need to maintain constant DR values to preserve similitude of the external flow field. A T_{aw} estimate utilizing several DR values and a single M value generated spatial trends inconsistent with expectation: h_f was higher outside the film cooled region than within it. Further coolant temperature control measures should help increase the range of T_w values attainable for a given DR leading to more accurate T_{aw} and h_f estimates.

5.2. Near Ambient Conditions

Development of the near ambient experimental regime is critical to the objective of scaling film cooling performance. Thermal data were collected via IR thermography, and the near ambient regime was found to provide freestream temperatures as great as 400 K while

various foreign gas species provided DR values ranging from approximately 1.4 to 1.7. Analysis of baseline carbon dioxide and argon cooled near ambient conditions revealed several unexpected results. Initial inspection of representative near ambient conditions revealed very concerning trends for both coolant gas species considered: no apparent cooling effect and increasing ϕ with downstream position. However, closer analysis revealed a very slight region of cooling effect immediately downstream of the film cooling holes. Increasing the difference between T_∞ and T_w should make film cooling effect more apparent by expanding the observed range of T_w values. The increase in ϕ with downstream position is attributed to decreasing freestream temperature. This T_∞ decrease results from energy loss to surroundings.

These results pose a serious challenge to achieving objectives requiring thermal data collected at near ambient conditions. Fortunately, a relatively straightforward solution may exist and is proposed. Replacing the foreign gas coolant with air at a temperature appropriate to produce an equivalent density ratio would increase the energy difference between freestream and coolant flows, resulting in a greater range of T_w values across the test plate. Additional complications encountered when applying the IR thermography method for near ambient conditions arose from suspected background radiation influence, optical distortion, and radiation reflected off the IR window. Solutions to these problems include isolating the FCR from its surroundings or subtracting a background IR radiation image from results, correcting for optical distortion using a black body, and acquiring an IR window with an anti-reflective coating. Isolating the FCR from surroundings and installing an anti-reflective IR window should provide great benefits with relatively little monetary and labor expenditure. The optical correction should also be implemented, but will likely be more costly in both time and money.

5.3. Matched Engine Representative and Near Ambient Conditions

Matching film cooling performance at engine representative and near ambient conditions directly supports the objective of scaling film cooling performance. Engine representative conditions were treated as a baseline to be matched in the near ambient operating regime. Both carbon dioxide and argon were used as coolant gases in the near ambient regime to maintain DR similitude. Air was not used as coolant for the near ambient conditions because T_c could not be controlled to provide the necessary values to match the DR values from the engine representative conditions.

The lone carbon dioxide cooled case yielded fairly good ϕ magnitude agreement between the engine representative and near ambient conditions. Argon cooled near ambient cases exhibited worse ϕ magnitude agreement with the respective engine representative conditions. The near ambient cases matching Re_∞ provided better ϕ magnitude agreement than the Re_c matching case. All the near ambient cases failed to reproduce the $\bar{\phi}$ trends present for the engine representative conditions. This trend error is believed to come from the use of foreign gas species to artificially augment the density ratio, resulting in decreased potential for energy transfer and limiting the range of T_w values.

The lack of agreement in ϕ magnitude and profile between the near ambient and engine representative conditions may also arise from incorrectly matched experimental conditions. Parameters such as the internal coolant Reynolds number, Re_{D_h} , were ignored in favor of excess rows of coolant. While this approach is very practical and likely for employment in a real system, it fails to ensure similitude in critical flow field parameters such as the Biot number and internal heat transfer coefficient. Future investigations should carefully consider Reynolds

numbers for the freestream, film coolant, and internal coolant flows as well as internal and external heat transfer coefficients to ensure appropriate matching conditions are achieved.

5.4. Preliminary PSP Testing

In order to fulfill the objectives of investigating physical model size and predicting effectiveness values, the PSP method was developed for use in the AFIT FCR. A setup providing satisfactory spatial resolution and adequate lighting was achieved, and this setup was used to generate preliminary results to evaluate use of the PSP method in the FCR. In addition to the software suites provided by ISSI, codes were written in MATLAB to streamline and simplify the PSP data reduction process. The test article was a representative turbine airfoil leading edge model comprised of a quarter cylinder, a flat section, and a tapered after body. Eight film cooling holes were located at 21.5° of arc on the quarter cylinder, oriented at a 90° compound angle, and inclined 20° relative to the surface.

Preliminary testing utilized carbon dioxide, argon, nitrogen, and helium as coolant gases at conditions of $I \approx 0.5$ and 1.5 with $Re_{\infty,D} \approx 10,000$. Results from these tests generally adhered to expected trends, including increased η values with increased M or I . However, unexpected results included a slight increase in coolant coverage when decreasing DR from 1.55 to 1.39 (changing from carbon dioxide to argon film coolant) and a lack of cooling effect when using helium as film coolant. The increase in coolant coverage is most likely not a physical reality, but an artifact of the experimental procedure used. It is recommended that the timing of wind off and wind on images be examined to minimize uncertainty in experimental results. Also, the physical rig should be examined to ensure there are no leaks in the coolant delivery system.

Comparison with data acquired in the LEFCT with a model 10 times the size of the FCR model revealed relatively good agreement for high I cases. However, elevated FCR freestream

turbulence levels are suspected to have caused broader coolant jets for all conditions. This elevated turbulence is also believed to have caused reduced peak η values and more rapid downstream η decay for low I FCR cases. Additionally, FCR cases exhibited less coolant jet penetration in the direction of coolant ejection. The cause of this apparent lack in jet momentum is not known but loss of coolant flow from leakage would be a likely culprit.

5.5. Future Work Recommendations

Because this study did not accomplish its overall goals, future work should first and foremost aim to complete investigation of the objectives laid out in the current work. Additional experimental programs are also recommended as a natural progression of this study and the facility capabilities. Further characterization of the AFIT FCR is certainly warranted, and these efforts should reveal facility and procedural shortcomings in addition to those described needing improvement.

5.5.1. Experimental Work

The first experimental program for future work should focus on completing the overall objectives of this study: investigate physical model size, scaling film cooling performance from near ambient to engine representative conditions, and effectiveness predictions. The objective of scaling film cooling performance should focus on parametric studies similar to that accomplished computationally by Greiner et al. [43]. Effectiveness predictions should seek to follow the procedure laid out by Williams et al. [8] for predicting ϕ based on η and ϕ_0 measurements as well as making η predictions from ϕ , h_f , h_i , Bi , and χ measurements. These predictions should be compared against actual measured ϕ and η values obtained using appropriate heat or mass

transfer methods. To assist in completing these objectives, the T_{aw} and h_f estimation method described by Greiner et al. [30] should be further developed for use in the FCR.

Smaller focus areas similar to the blowing ratio and internal coolant flow rate sweeps examined include studying freestream velocity, flow rate, and temperature effects. Additional focus on density ratio effects would also be well warranted. This study of density ratio should include both foreign gas species and air as a coolant gas, reaching equivalent density ratio values for each gas species. Each of these studies would serve to characterize the performance of the FCR and its particular design features. These studies will also satisfy the larger goal of gaining valuable insight on film cooling behavior.

Based upon past experimentation in the FCR by Lynch [48], future work should also examine film cooling schemes with additional rows of cooling holes as well as different cooling hole designs. A test plate buildup of one, three, and five rows of cooling holes for simple cylindrical, fan-shaped laidback, and trenched cooling holes would provide a thorough study of cooling hole shapes and impacts from additional rows of cooling holes. However, this study will likely require facility modifications to ensure consistent density ratio and internal coolant flow rate across all test cases.

5.5.2. Facility Improvements

In order to attack the experimental programs recommended above, several facility modifications should be made. These modifications will be described as either necessary or simply beneficial for the pursuit of these programs.

The design of the FCR representative turbine airfoil test plate was found to have significant impact on the cooling effects observed. Previous work did not examine the leading edge region of the test plate, and two major flow field features were not discovered. First, a

major bias in cooling effect to one side of the test plate revealed a definite coolant distribution issue within the internal coolant feed system. This coolant distribution issue was satisfactorily remedied with implementation of a baffle plate within the coolant feed, but further work is recommended to seal this internal coolant flow path. Second, two areas of coolant impingement on the internal test plate surface were observed. Cooling effect from the upstream impingement region overshadowed the film cooled region, so if future FCR work focuses explicitly on film cooling, this design may require modification.

Two of the greatest obstacles to film cooling research in the FCR are coolant temperature and mass flow control. Coolant temperature control is inherently limited because the FCR functions as a heat exchanger, meaning low coolant mass flow rates yield high coolant temperatures. While this influence cannot be eliminated, it should be reduced as much as feasible. To assist with maintaining consistent coolant temperature, a wider range of control for coolant mass flow is required. Excess coolant removal from the FCR is currently limited to approximately 22 SLPM of air. Increasing the capacity to remove excess coolant could open up a large operating regime necessary for matching film cooling performance in separate temperature regimes.

To achieve the objective of matching film cooling performance at engine representative and near ambient conditions, a new test plate material will most likely be required. Further experimentation should examine changes in h_f from near ambient to engine representative conditions to determine if this less conductive test plate is strictly required, but an analysis of the relevant equations and available data indicates that this would be a worthy investment for the FCR.

Finally, while not required to achieve any of the experimental objectives, a new routing and external support for the test plate thermocouples is strongly recommended. A new setup for these thermocouples could better seal the internal coolant passage and eliminate the need to pass thermocouples through a pipe barely larger than the thermocouple connectors. These thermocouples are also prone to breakage near the connector strain relief. A new patch wire interface could better support and protect the thermocouple wires to help prevent premature failure. If a redesign of the FCR main cooling block (see Figure 3-13) is accomplished in the future, these design features should be incorporated to save time and money.

Appendix A

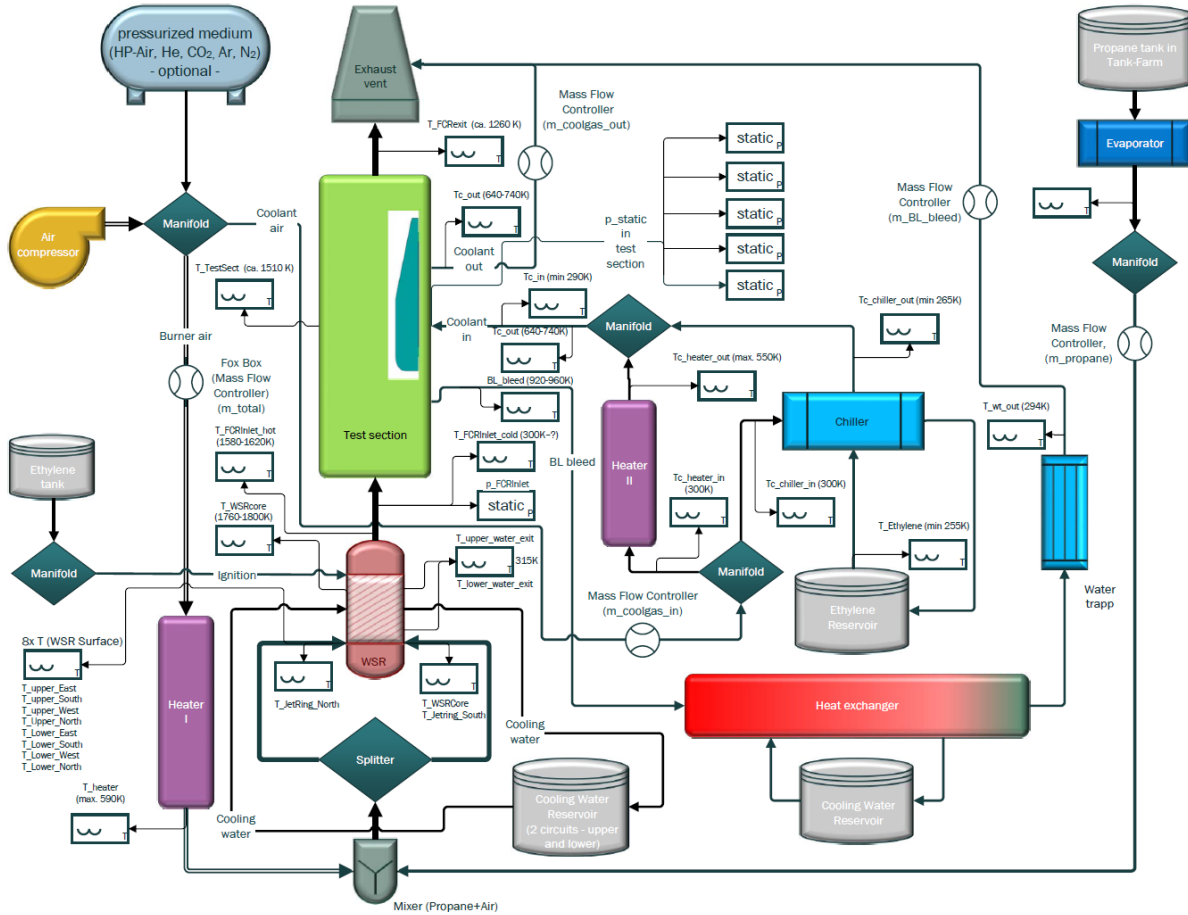


Figure A-1: AFIT FCR flow and instrumentation diagram

Appendix B

Well-Stirred Reactor (WSR) Operating Procedures

Before proceeding with the start-up procedure check the following items:

- 1. Ignitor assembly installed? (Location: below WSR)**
- 2. Turn on the MKS control unit and ensure all mass flow controllers are shut off**
- 3. Ensure the ethylene supply line is hooked up correctly to the ignitor and its mass flow controller**
- 4. Check position of the propane supply valves:** copper tubes between UCC and FCR and further downstream.
 - a. Propane supply to UCC must be shut off at all times when operating the FCR
 - b. Propane supply to FCR must be shut-off until shortly before ignition (see procedure). Is it hooked up to a mass flow controller?
- 5. Check if boundary layer bleed line and coolant gas line are correctly attached to the rig and respective mass flow controllers**
- 6. Check bucket water level:** Coolant exhaust tube should be well in the water

Observe the following temperatures continuously during rig operation, to ensure rig health:

- a) WSR Core temperature (check for flame out, Temp. range 1500-1800 K)
- b) FCR Inlet temperature (check for flame anchored in transition section, < 1600 K)
- c) Water Exit lower/upper (should not exceed 330 K each line)

Exceeding the limits of any item individually or in combination may cause considerable damage to the WSR.

The table below covers provides step by step starting procedures for both high and low temperature operating regimes for testing with the FCR. Steps which are essential to both regimes have a shared row.

Legend: * = optional, if needed

Step	Low Temperature Run	High Temperature Run
1*	Turn on the chiller. It will take at least 3h to cool the ethylene glycol reservoir and the surrounding equipment to a stable state. (Location: next to the FCR, below the laser table)	
1		Turn on one of the Zimmer-LPG/Propane-Vaporizers in the tank farm (turn red lever up) and allow it 30 minutes to warm up, continue start-up procedure while it warms up. (Location: in tank farm, right of tank farm door, white plastic)
2	Turn on Fox Box switch next to “Fan 1B” switch. (Location: left side of UCC rig)	
3	Turn on power to DAQ tower (Switch location: between monitors at control desk)	
4	Turn on both control boxes in back of DAQ tower and press the output (“out”) button on each.	
5	Load the “WSR/FCR Rig control” LabVIEW GUI. Close the program if open, restart, select a new file to save the data to, and press “RUN” in the GUI.	
6	Set mass flow in LabVIEW control interface to 0% (for 0.75in air line), before continuing... Open air line main lever (lower yellow “WATTS”-Lever), turn it from 6h to 3h position. Turn on compressed air to 0.75in line (lower green “WATTS”-Lever) – turn gradually from 6h (closed) position to half open (4:30 position), waiting for the regulator (Fox Box) to fully close (reads “0”) before fully opening the lever to 3h position. Turn on the 3/8in line (white manifold, red lever – 6h to 3h position).	
7	Turn on the wall heater (black lever, turn counterclockwise and green power control button on/off) and set to 30% power (good for testing at 600 SLPM). (press rectangular arrow first, then set power level with up/down keys) (Location: grey box on wall on the left, reading: “secondary flow”; Note: for cold testing a different power setting may be required, temperature limit for the box is 600°F)	
8		Wait for the air line temperature to reach more than 425°F (see power control display of wall heater) and the jet ring temperature to exceed 375 K (check LabVIEW interface). Adjust wall heater if necessary directly at the grey box. (Limit: 600°F)

9		Turn on the Mokon Supply Unit and heat to 300°F (turn red general power switch, press green “start”-button) (Location: grey box, back left corner of the Lab)
10	Turn on the WSR water coolant (turn lever counterclockwise from 3 to 12 o’clock and set both flow rates to 0.1. GPM Note: open return line first and supply line after (Switch Location: left hand side of the CAI Tower at the wall, black lines with orange switches; flow meters at the left side of the CAI tower)	
11	In LabVIEW control interface, turn the $\frac{3}{4}$ in air line flow up to 650 SLPM (12%)	
12		Turn on switch damper for running the UCC (Switch Location: left hand side of the UCC rig on test rig housing wall)
13	Open damper above the FCR (turn counterclockwise from 9 to 6 o’clock) (Location: handle outside the exhaust duct on the outer wall)	
14		Put cover on UCC exhaust duct
15		Turn on Fan 2 (right from the UCC damper switch) and Fan 1B (right next to Fox Box switch)
16	Turn on the monitors on the control desk.	
		Continue only after vaporizers have warmed noticeably (minimum 20 minutes)
17		Turn on either propane tanks 1+2 connected to the upper vaporizer or propane tanks 3+4 connected to the lower vaporizer (Apollo Levers) and open the ball valve between the tanks and the vaporizer (main lever and red “UCC”-Lever). (Location: in tank farm, right hand side)
18		Open the ethylene bottle. (Location: in tank farm, middle)
19		Turn on the “Brooks” control device (controls propane mass flow only)
20		Open the propane line valve (90° turn from 3h to 12h position) with the black handle. (Note: lower vaporizer in tank farm feeds the black valve, the upper vaporizer in tank farm feeds red valve) (Valve location: between UCC and FC rigs, copper tubes)
21		Open the ethylene valves in the lab to provide ethylene for rig ignition (black handle, turn counterclockwise, 12h to 9h pos.). (Location: wall left hand side, at multi-manifold)
22		Open ignitor air flow valve. (Note: This valve is a 3-way valve: 3h position (arrow right) for ignitor air supply, 12h position (arrow north) valve closed, 9h position (arrow left) coolant air supply)
23	Turn on the boundary layer bleed vacuum pump. (Location: right of FCR on the floor below Laser desk)	

24		Check of mass flow controllers: 1. Open solenoids in LabVIEW interface 2. Open channels at the MKS control board - Channel 1 for Ignitor air flow (set to 25 SLPM) and coolant supply - Channel 4 for Coolant exhaust - Channel 5 for Ethylene supply (set to 3.5 SLPM) - Channel 6 for BL bleed (50 SLPM) (Example: Command: “ON 6” followed by “ON ALL”; close channel: “OFF ALL” followed by “OFF 6”) 3. Open channel 2 on the Brooks to check propane flow
		The system is now set up for ignition. The following steps need to be carried out in strict order and without delay.
25		LabVIEW: Set airflow to 400 SLPM (7.44%)
26		Open propane flow first (Channel 2 on Brooks controller). Turn on the propane to 13.4 SLPM ($\Phi = 0.8$).
27		Turn on the ignitor air (25 SLPM, Channel 1, MKS board) and ethylene flows (3.5 SLPM, Channel 5, MKS board). Note: set the channels first, then open all together “ON ALL”)
28		Turn on the ignitor and allow ignitor to run for a few seconds after ignition, then disengage. (big yellow button “Ignitor” in LabVIEW interface, must be disengaged to turn off ignition) (Note: compare $T_{WSR,core}$ vs $T_{FCR,inlet}$ for successful ignition: $T_{WSR,core} > T_{FCR,inlet}$)
29		Turn the WSR coolant water up to 0.6 – 0.7 GPM for the “bottom” and “top” water circuits (turn them all the way up). (Location: left hand side of CAI rack)
30	Open coolant air flow valve. (Note: There are 2 different valves employed for 1) heated coolant air (1st black handle towards the UCC rig) and 2) chilled coolant air (middle black handle) This valve is a 3-way valve, but only 2 modes are necessary when the rig is not going to be lit. 12h position (arrow north) valve closed, 9h position (arrow left) coolant air supply)	After successful ignition, turn off the ignitor, ethylene supply, and ignitor air supply. Turn handle for ignitor air supply by 180° to enable coolant air supply (3-way valve)
	Allow the rig to warm up (10-20 minutes at minimum – can take up to 45 minutes)	

31		When increasing propane flow, make small adjustments (1-2 SLPM) to prevent large overshoots which could result in unstable WSR performance
32		If testing at high fuel/air ratios, it is recommended to increase the rate above stoichiometric at 400 SLPM of air, to prevent overheating the WSR
33		Alternate increasing propane and air until 600 SLPM (11.1%) of air is achieved
34*	Turn on the heater. There are 2 heaters. (1200 W and 800 W) operate the heaters at 20-25 SLPM. Maximum Power Setting 8. The heaters are located between the FCR and the UCC rig. There is a power switch for each heater at the left hand side of the CAI rack.	
	Wait for Internal Channel temperatures, Backside and Surface temperatures levelled out – stabilized. (+ Heater exit temperature if turned on)	

Grouping of Steps for WSR Start-Up

<u>DAQ Systems and Rig-Subsystems</u>	<u>Ignition</u>	<u>Burner</u>	<u>Coolant Supply and Management</u>
			1*
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			35*

WSR Shutdown Procedure

The procedure covers both low and high temperature operating regimes

Step	Description (* indicates high temperature testing only)
1*	Reduce air flow to 300 SLPM, wait 3-5 minutes (set airflow to corresponding % in LabVIEW, reduce propane flow (SLPM) at “Brooks”-Controller) (*)
2*	Set propane flow to “0” (“Brooks”-Controller) (*)
3	Turn air up to 12% in 2 steps (10%, 12%)
4	Turn off both film coolant heaters.
	It is helpful to run some film coolant flow to cool the FCR.
5	Set the wall air line heater to 0% and turn it off (wait to see temperature starting to drop)
6	Turn the Mokon temperature down to 100°F or lowest possible value and turn on the water coolant supply for the Mokon Supply Unit (MSU) (Location: back left corner of the Lab, black lines, red levers; Note: open return line before open supply line)
7*	Shut the propane and ethylene tank valves. (*) (Note: keep the vaporizer heater on until after the propane lines have been depressurized)
	Wait until the WSR core temperature drops below 400 K to prevent spontaneous ignition during propane and ethylene supply line depressurization
8*	Depressurize (bleed) the ethylene line. (*) (Set mass flow to 3.5 and command “ON ALL”; turn off “OFF ALL” when zero has been reached)
9*	Turn the propane flow up to 20 SLPM (Brooks-Controller) while keeping the air flow at 12 %. (*) This can take several minutes.
10*	Once the propane line is bled, close the valve between propane tanks and the vaporizer. (*)

11*	Turn off the propane vaporizer. (*)
	Wait for the rig to cool. (approx. 10-15 minutes or more conservative until air temperature falls below 150°F and the jet ring temperature below 330 K)
12	Turn off the Mokon system, once the temperature drops below 125°F. Close the cooling water lines. (close supply line before return line)
13	Set mass flows at MKS control unit to 0. (Channels 1,4,6; “OFF ALL”) Set ¾ inch line mass flow to 0% in LabVIEW
14	Close the air supply valves. (1. Yellow lever; 2. Green lever; 3. Red lever; 4. Black lever)
15	Turn off the exhaust fans and turn the UCC damper control off. Close the FCR-Damper in exhaust duct.
16	Close the WSR coolant water supply.
17	Turn off the chiller.
18	Stop LabVIEW. Turn off equipment (Fox Box, DAQ Tower, Monitors at control desk, Brooks).
19	Empty water trap.
20	Run the vacuum pump for 10 to 15 minutes to remove any possible condensation from its cylinders

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