



**RELATIVE CONTRIBUTIONS TO OVERALL EFFECTIVENESS IN
GAS TURBINE COOLING**

THESIS

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AFIT-ENY-MS-17-M-248

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THESIS

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Carol E. Bryant, BS

Second Lieutenant, USAF

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Abstract

Gas turbine engine hot gas path components are protected via coolant that travels through internal passageways before being ejected as external film cooling. Efforts aimed at improving cooling are often focused on either only the internal cooling or the film cooling; however, the common coolant flow means the internal and external cooling schemes are inextricably linked and the coolant holes themselves provide another convective path for heat transfer to the coolant. The relative influence on the overall effectiveness of internal cooling, external cooling, and convection through the film cooling holes is not well understood.

Large scale, matched Biot number experiments were performed to identify each cooling component's impact on overall effectiveness (ϕ) using the $\Delta\phi$ method, which allows for comparison of two schemes by subtracting one scheme's ϕ from another. Several impingement designs were tested, but all had low $\Delta\phi$ values. Blocking coolant holes resulted in significant changes in ϕ , however it was physically impossible to decouple the effects of cooling within the hole and film cooling. Ingestion of the freestream flow was observed at low blowing ratios, significantly reducing the overall cooling.

Physical limitations were overcome by performing computational fluid dynamics (CFD) simulations to isolate each cooling component, and thereby determine their relative effectiveness. The conjugate simulations were of flat plates with all three types of cooling. The internal cooling effects were isolated by selectively setting individual surface boundaries as adiabatic. The film cooling effects were isolated by setting an artificial mass

flow sink at the exit of the film cooling holes. Hole cooling was found to have an equal or greater impact on overall effectiveness as film cooling, while impingement had little to no effect.

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Carol Bryant

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Nomenclature

A	=	area
Bi	=	Biot number
c_p	=	specific heat at constant pressure (J/kg-K)
D	=	diameter
DR	=	density ratio
h	=	heat transfer coefficient, (W/m ² -K)
H	=	spacing between jet plate and impacted surface
I	=	momentum flux ratio
IR	=	infrared
k	=	thermal conductivity (W/m-K)
L	=	characteristic length
M	=	blowing ratio
Nu	=	Nusselt number
Pr	=	Prandtl number
q''	=	heat flux, (W/m ²)
Re	=	Reynolds number
s	=	entropy
T	=	temperature, (K)
V	=	velocity (m/s)
w	=	width
μ	=	dynamic viscosity (N-s/m ²)
α	=	thermal diffusivity (m ² /s)

$\Delta\phi$	=	change in overall effectiveness
η	=	nondimensional adiabatic effectiveness
θ	=	nondimensional surface temperature
ν	=	kinematic viscosity (m ² /s)
ρ	=	density (kg/m ³)
σ	=	standard deviation
ϕ	=	overall effectiveness
χ	=	warming factor

Subscripts

0	=	reference condition
∞	=	freestream
aw	=	adiabatic wall
c	=	coolant
ce	=	coolant at the exit of the film hole
ci	=	coolant at the inlet to the film hole
cp	=	coolant in the plenum
f	=	film
h	=	hole
i	=	inner
imp	=	impingement
LE	=	leading edge
o	=	outer

p = predicted

ref = reference

s = solid

RELATIVE CONTRIBUTIONS TO OVERALL EFFECTIVENESS IN GAS TURBINE COOLING

I. Introduction

1.1 General Issue

Gas turbine engines have four primary components. The first stage is the compressor where gas is compressed, increasing pressure and temperature. Next, the gas enters the combustor, where it is mixed with a fuel and ignited, again causing an increase in temperature. In the third stage, the gas encounters the turbine, where work is extracted. Finally, the gas exits the engine through a nozzle. An ideal Brayton cycle diagram, such as in Figure 1, can show the ideal temperature and entropy changes through the engine.

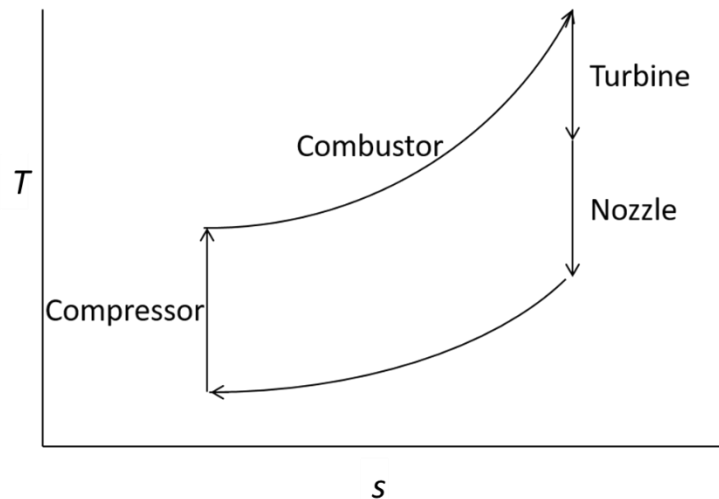


Figure 1: T - s diagram for a Brayton cycle

The first turbine stage is subject to the hottest gases directly from the burner. This environment will destroy most materials. However, it is desirable to operate gas turbines as hot as possible because the thrust and power an engine can produce is directly limited

by the energy imparted by the combustion gases. Therefore, it is necessary to cool the turbine blades to get maximum performance from an engine.

There are several methods to cool a turbine blade. The cooling methods can be broken down into two general categories: external and internal cooling. External cooling is any method which directly affects the outside surface of the blade via convection, such as film cooling. Internal cooling is any method which takes place inside the blade; the effect is then conducted through the thickness of the blade material to affect the outer surface temperature. Impingement cooling and internal channels are examples of internal cooling.

The coolant air for all methods of cooling must come from the compressor after much work has been applied. This valuable air skips the burner stage and flows through the turbine blades and mixes with the combustion gases reducing the temperature which decreases the overall efficiency of the engine. So, while the turbine would be quite cool if it were constantly awash in enormous amounts of coolant air, the engine would nearly cease to provide any thrust. Therefore, it is evident that coolant air should be conserved to keep the performance of the engine as high as possible. The best turbine cooling schemes must make efficient use of coolant.

The leading edge of the turbine blade is a concern because the freestream gases from the burner are hot enough to melt the blade. Three forms of cooling are used to cool this area. Internal cooling, such as impingement, cools the inside of the blade. The coolant then flows through film cooling holes, cooling them via convection. Finally, external cooling, such as a showerhead of film cooling holes, injects a thin film over the top surface. The interaction of each of these factors, depicted in Figure 2, and their relative contribution

to cooling the surface of the blade is unknown. In Figure 2, the blue arrows show internal cooling, green arrows show cooling within the film holes, red arrows show film cooling, and the yellow arrow shows the freestream.

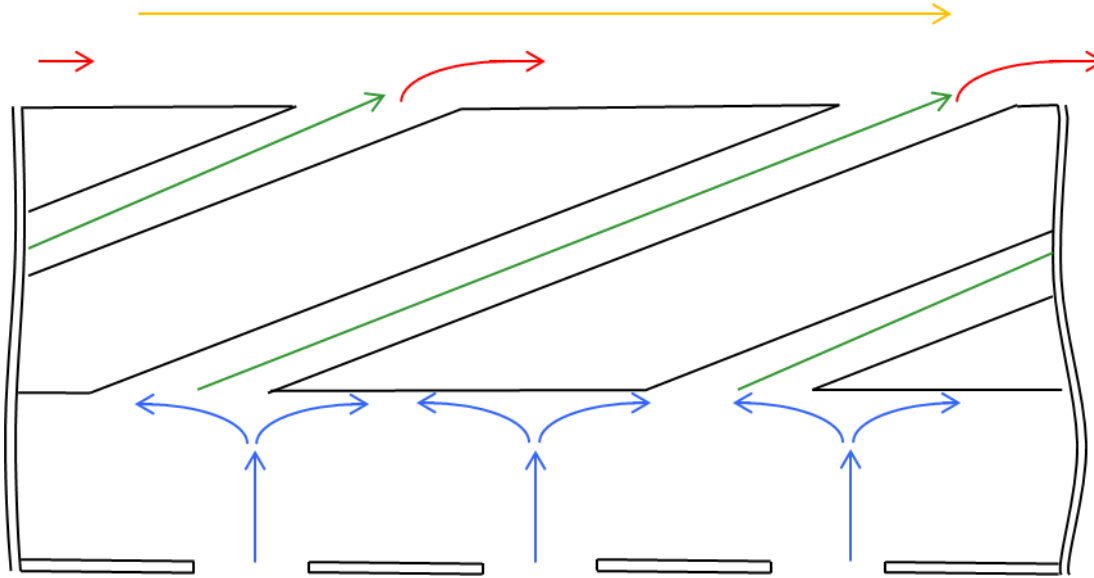


Figure 2: Diagram of different types of cooling.

1.2 Research Objectives

The objective of this research is to answer three questions on the interactions of the three types of cooling.

- What are the relative contributions by each part of cooling to the overall effectiveness of the cooling design?
- What are the effects of different impingement plates?
- What areas are most affected by each type of cooling?

These questions are answered through a novel combination of experiments and CFD simulations. Accomplishing these objectives will allow for improved research focus

on the most impactful forms of turbine cooling while potentially saving weight and thermodynamic losses on techniques that are relatively ineffective.

This document will first discuss important parameters and relevant previous research. Next, experimental methodology and results will be discussed. Computational methodology and results are then discussed. Finally, conclusions and suggested for future work are covered.

II. Literature Review

Most previous research in leading edge cooling has been focused on film and impingement cooling. Very little research has been conducted on the third cooling effect, cooling within the film holes. The purpose of this section is to review previous work on all three areas with respect to leading edge gas turbine cooling.

2.1 Film Cooling

Film cooling is based on the interactions of jets in crossflow. The injection of a jet of coolant and freestream results in a film of mixing layers along the downstream surface. The typical geometry for film cooling is shown in Figure 2. Per Bogard and Thole [1], there are three primary factors which effect film-cooling performance: coolant and freestream conditions, hole geometry and configuration, and the airfoil geometry. To further discuss film cooling, the basic parameters of film cooling must be discussed.

2.1.1 Important Parameters

The fundamental principle of all three cooling forms is convective heat transfer which obeys Newton's law of cooling:

$$q'' = h(T_{aw} - T_{\infty}) \quad (2.1)$$

where q'' is the convective heat flux, h is the convective heat transfer coefficient, T_{aw} is the adiabatic wall temperature, and T_s is the solid temperature. The layer of air comprising the film is a mix of coolant air ejected from the turbine blade and the hot freestream gases from the burner. If the wall is adiabatic, the temperature in the film directly above the surface is called the adiabatic wall temperature.

To determine how effective the film cooling is, a nondimensional parameter, adiabatic effectiveness η , is used. It is defined as

$$\eta = \frac{T_{\infty} - T_{aw}}{T_{\infty} - T_{ce}} \quad (2.2)$$

where T_{ce} is the temperature of the coolant as it exits the film cooling hole. To determine the adiabatic wall temperature, most experiments use a low conductivity material and apply a conduction correction. The low conductivity material is used because adiabatic effectiveness only shows the effect from film cooling, excluding all internal and hole cooling. An IR camera is typically used to record the surface temperature in the experiments, therefore a low conductivity material is needed. However, there is no perfectly adiabatic materials therefore, a conduction correction is applied to correct for any conduction due to internal cooling within the model. Film effectiveness varies widely based on the coolant hole shape, flow geometry, as well as the ratio between the coolant and the freestream density, velocity, mass flux, and momentum flux [1].

Another parameter, $NHFR$, shown in Equation 2.3, describes the reduction in internal cooling heat load by adding film cooling, without changing the surface temperature. It is a nonphysical representation; the wall temperature would decrease if there was a reduced heat flux. But the parameter has merit because it describes the hypothetical decrease in internal cooling necessary to maintain an acceptable outer surface temperature.

$$NHFR = 1 - \frac{q_f}{q_0} = 1 - \frac{h_f}{h_0} \left(1 - \frac{\eta}{\phi} \right) \quad (2.3)$$

The f subscript denotes the condition with film cooling while the 0 means without the film. The ratio of heat transfer coefficients is usually greater than one due to turbulence in the boundary layer, therefore negative values of NHFR are possible.

Overall effectiveness, ϕ , is a nondimensional parameter that describes the cooling effectiveness of a cooling scheme on a conducting material. It relates the coolant temperature in the plenum, as it enters the cooling channels, to the airfoil surface temperature compared to the freestream and surface temperatures:

$$\phi = \frac{T_{\infty} - T_s}{T_{\infty} - T_{ci}} \quad (2.4)$$

Overall effectiveness is a useful parameter because it shows the effects of all three types of cooling, not just film cooling as in adiabatic effectiveness.

There are several other important parameters in convective heat transfer. The Reynolds number is the ratio of the inertial to viscous forces in a fluid flow. It is primarily utilized to characterize the flow in the freestream. For leading edge film cooling, the characteristic length, L , is typically the diameter of the cylindrical leading edge.

$$Re = \frac{VL}{\nu} \quad (2.3)$$

The Nusselt number is the ratio of convection to pure conduction heat transfer.

$$Nu = \frac{hL}{k_f} \quad (2.6)$$

Where h is the convection heat transfer coefficient at the surface, L is a reference length, and k_f is the thermal conductivity of the fluid. In internal and external flows, a high Nusselt

number means an increased heat transfer coefficient, which is commonly the result of impingement or film cooling.

There are several ways the cooling flow rate is nondimensionalized in film cooling literature. Freestream gases in the turbine are much hotter than the coolant, therefore the coolant flow characteristics are very dependent on temperature. A frequently varied parameter in scaled experiments is density ratio:

$$DR = \frac{\rho_c}{\rho_\infty} \quad (2.7)$$

Other flow parameters which describe various other portions of the flow are the mass flux, or blowing ratio M , the momentum flux ratio I , and the velocity ratio VR .

$$M = \frac{\rho_c V_c}{\rho_\infty V_\infty} \quad (2.8)$$

$$I = \frac{\rho_c V_c^2}{\rho_\infty V_\infty^2} \quad (2.9)$$

$$VR = \frac{V_c}{V_\infty} \quad (2.10)$$

The blowing ratio, M , is the ratio of the coolant mass flux to the freestream mass flux. It partially represents the ability of the coolant to transfer energy. It is evident from the definition of the parameters that M and I cannot be matched simultaneously if DR is not matched.

Now that the basic parameters have been discussed, previous research will be discussed. First, adiabatic effectiveness, or the effect of the film, is most common parameter studied when looking at film cooling configurations.

2.1.2 Adiabatic Effectiveness on a Flat Plate

Teekaram et al. [2] studied adiabatic effectiveness for slot injection. They found η was 1.0 directly after the slot, then decreased downstream. They also found increasing the blowing ratio increased adiabatic effectiveness downstream of the slots for blowing ratios up to 0.52. The higher blowing ratio tested, $M=0.72$, did not significantly improve η . Their results are shown in Figure 3.

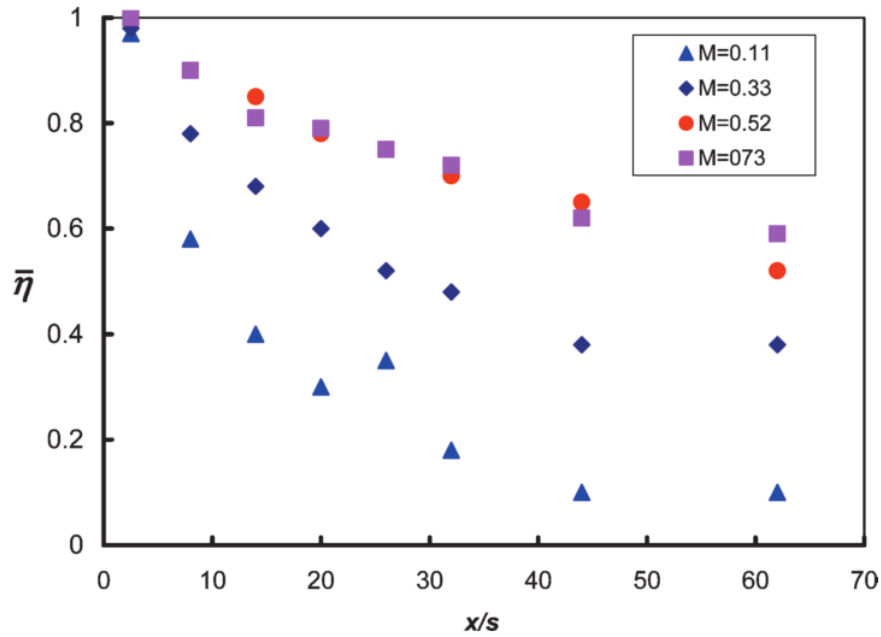


Figure 3: Average adiabatic effectiveness for 30-degree slot injection.

Figure from [1], data from [2]

Baldauf et al. [3] measured spanwise adiabatic effectiveness downstream from a single film cooling hole. They found increasing the blowing ratio moved the peak effectiveness downstream, from 1.9 hole diameters at $M=0.4$ to 8.5 hole diameters at $M=2.2$ due to separation and reattachment. Another important result trend was decreased peak adiabatic effectiveness with increased blowing ratio. The local η distribution for range of

blowing ratios tested are shown in Figure 4, with white triangles indicating the locations of peak η . They also researched the effects of changing the density ratio. They found increasing the density ratio slightly increased the peak η at blowing ratios greater than unity. Another effect of increasing density ratio was moving the peak effectiveness location further downstream.

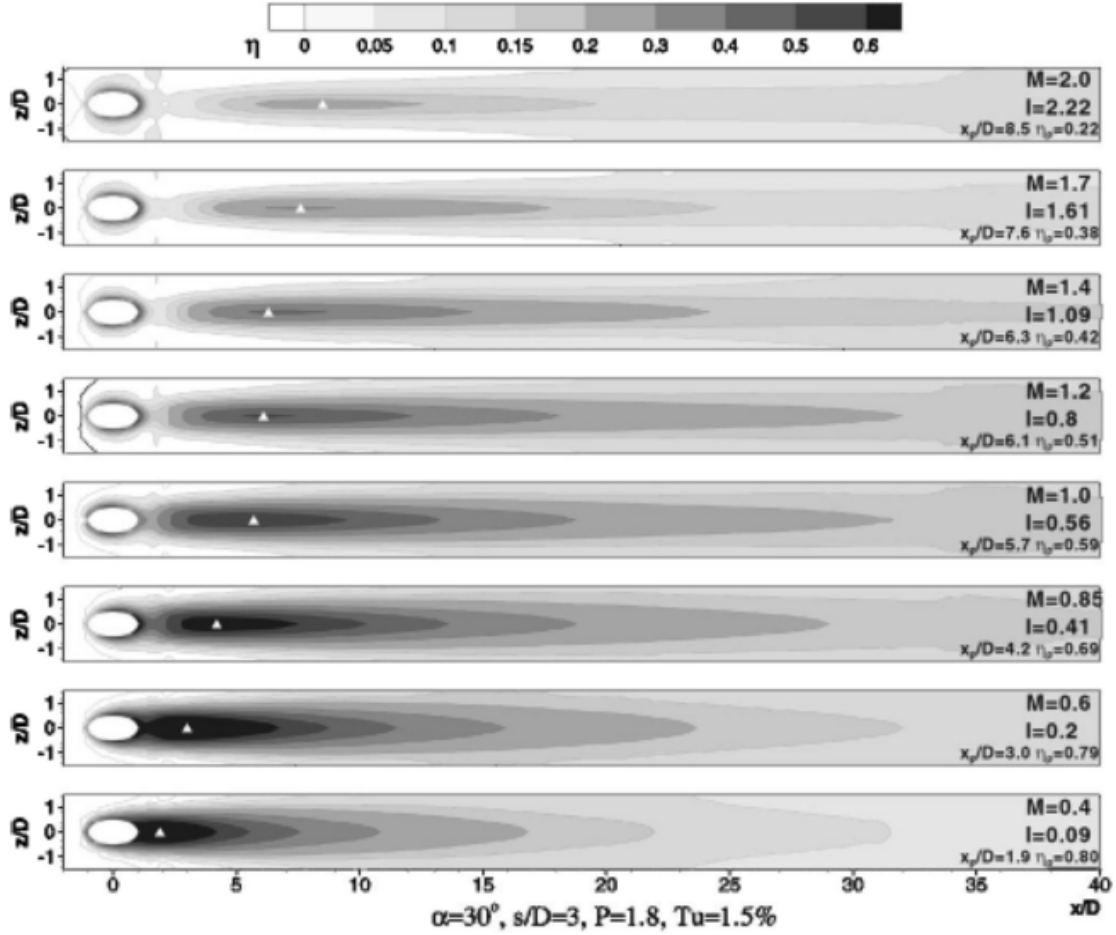


Figure 4: Adiabatic effectiveness contours for a single film cooling hole [3]

When the coolant jet does not stay attached to the outside surface and form a film, it is called jet liftoff. Coolant jet separation is evident in the η distributions by Baldauf et al. between $M=1.0$ and 1.2 . The coolant distributions are nearly identical; however, the

magnitude decreases at $M=1.2$, indicating jet liftoff. Figure 5, from Thole et al. [4], shows overall effectiveness profiles of an (a) attached coolant jet, compared to (b) a separated coolant jet.

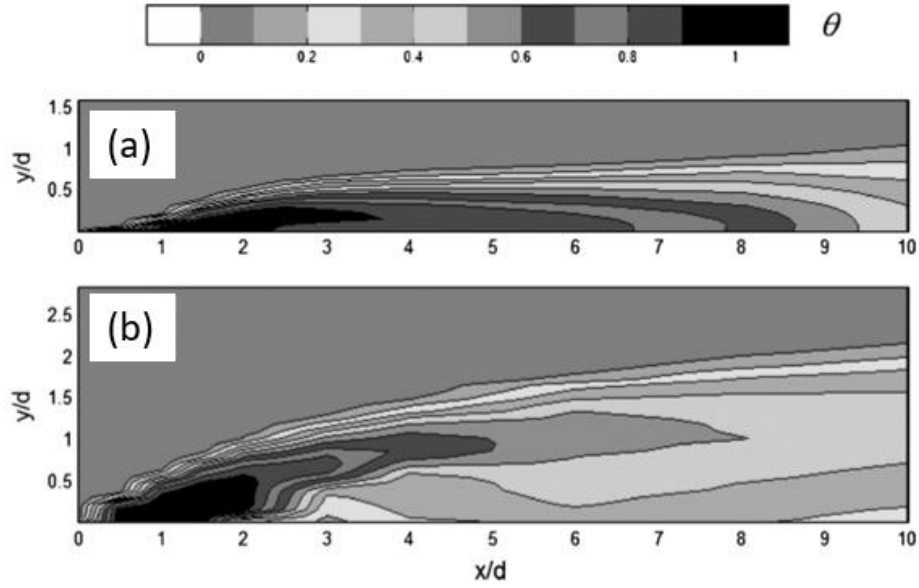


Figure 5: Overall effectiveness profiles along the centerline of a coolant jet.

(a) attached jet and (b) separated jet [4]

Numerous studies have examined adiabatic effectiveness in the leading edge region of simulated turbine vane. Film holes in the leading edge region are usually closely grouped in staggered arrays. One study on the leading edge region by Cutbirth and Bogard [5] measured adiabatic effectiveness in the stagnation region for a range of blowing ratios. They found the coolant jets separated from the surface at low blowing ratios, but the interaction of the jets still created a barrier to the freestream resulting in high η even with separation.

2.1.3 Effects of Curvature and In-Hole Roughness

Because the leading edge of the turbine blades are convex, it is important to understand the effect curvature has on film cooling performance. Per Bogard and Thole [1], when a jet is injected along a curved wall, the ratio of the coolant to freestream momentum determines whether the jet will liftoff from the surface. If the jet momentum is greater than the freestream, the radius of curvature for the jet will be greater than the radius of curvature for the wall and the jet will separate from the wall. Ito et al. [6] measured adiabatic effectiveness downstream for a range of blowing ratios, shown in Figure 6. They found blowing ratios greater than 0.5 correlated to jet liftoff from the surface.

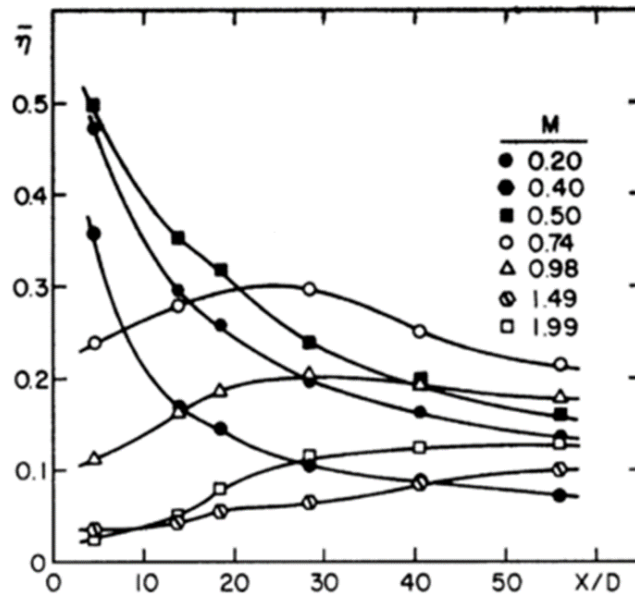


Figure 6: Adiabatic effectiveness downstream on a convex wall [6]

There has not been much research on the effects of roughness inside film cooling holes on overall cooling effectiveness. One paper by Stimpson et al. [7], measured the roughness inside channels manufactured using Direct Metal Laser Sintering (DMLS). DMLS is

frequently used to manufacture modern turbine blades with complex cooling schemes. They measured the roughness in the channels using an optical profilometer. They used a range a test coupons, manufactured from a variety of materials, in a variety of sizes. For each test coupon, they reported the mean arithmetic roughness nondimensionalized by the hydraulic diameter of the cooling channels, R_a/D_h . This value varied from 0.009 to 0.025, with a mean of 0.017. Another important result from this paper is a correlation relating equivalent sand grain roughness to the nondimensionalized roughness. The correlation was created by comparing four previous experimental results on rough channels to the results for DLMS channels.

$$\frac{k_s}{D_h} = 18 \frac{R_a}{D_h} - 0.05 \quad (2.11)$$

This correlation is only applicable for $R_a/D_h > 0.0028$ because any values less than this will result in a negative k_s/D_h .

Schroeder and Thole [8] studied the effect of roughness in a film hole on adiabatic film effectiveness. They performed a literature review on the in-hole roughness. They found the typical R_a/D_h for aero engines is 0.002-0.020. To test the effects of roughness they used a row of shaped film cooling holes with varying roughness machined in polystyrene. They manufactured the holes using a CNC mill and then measured the resulting roughness using an optical profilometer. Testing the holes showed decreasing area average adiabatic effectiveness with roughness and blowing ratio, as shown in Figure 7.

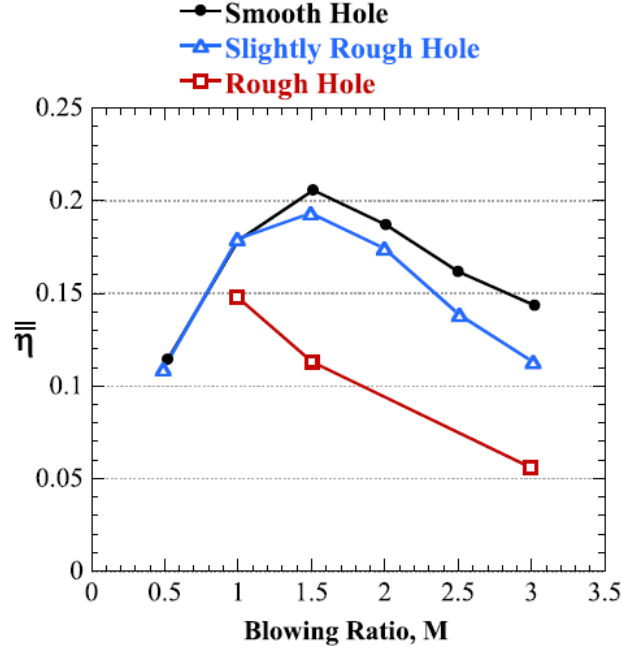


Figure 7: Area average adiabatic film effectiveness [8]

2.1.4 Overall Effectiveness

In order to obtain overall effectiveness data in laboratory conditions, a representative model must transfer heat at room temperature the same way it would at engine conditions. It is often desirable to test a cooling design in laboratory conditions at low temperatures, compared to engine conditions because a larger model is easier to instrument and allows easier optical access. To use a large-scale model to find overall effectiveness distributions, it must be constructed from a matched Biot number material. The Biot number is a ratio of the internal thermal resistance of a solid to the boundary layer thermal resistance. A matched Biot number model will transfer heat at large scale conditions, the same way the metal blade will transfer heat at engine conditions.

$$Bi = \frac{hL}{k_s} \quad (2.12)$$

The equation for overall effectiveness can then be re-written to include the Biot number and several other parameters used in scaling cooling designs.

$$\phi = \frac{1 - \chi\eta}{1 + Bi + h_f/h_i} + \chi\eta \quad (2.13)$$

To get the most accurate overall effectiveness distributions for scaled experiments, all four of these parameters, Bi , η , χ , and h_f/h_i , must match engine conditions. For representative experiments, η is expected to match due to similar geometry and Reynolds number. The Biot number can be matched by constructing the model from an appropriate material. The coolant warming factor, χ , is expected to match by both matching conductivity and by matching internal and external flow conditions. The coolant warming factor χ , is defined as the amount the coolant heats up from when it enters the plenum to when it exits the film cooling holes.

$$\chi = \frac{T_\infty - T_{\text{hole exit}}}{T_\infty - T_{\text{plenum}}} \quad (2.14)$$

The ratio of h_f/h_i can be matched by matching the internal and external Reynolds numbers and by also matching the mass flux and viscosity ratios. Because of difficulties matching these four parameters simultaneously, scaled experiments often have error due to imperfect matching. Weise [9] explored the effects of matching various ratios and their effects on adiabatic effectiveness in depth. Polanka et al. [10] describes a method of using a thermal barrier coating to correct turbine thermal behavior using overall effectiveness measurements.

Greiner et al. [11] performed a computational study on the scaling of film cooling from ambient to engine temperatures on a flat plate geometry with a single shaped film cooling hole. To appropriately scale, momentum flux and density ratios must match. Additionally, freestream and coolant Reynolds and Prandtl numbers should match, but this is frequently impossible experimentally. The authors suggest that if a tradeoff must be made, it is better to match coolant Reynolds number to improve scalability.

Rutledge et al. [12] also used CFD to show the influence of matching M and DR between laboratory and engine representative temperature flow conditions with matching freestream or coolant Re . They found the centerline η distribution matched well with matched coolant Re . The spanwise average η distribution was most closely matched with matching freestream Re .

Albert et al. [13] performed experiments comparing adiabatic effectiveness data from an approximately adiabatic model with the overall effectiveness data gathered directly from the normalized surface temperature of the matched Biot number model of a leading edge experiment, shown in Figure 8. They found overall effectiveness to be significantly higher than adiabatic effectiveness over most of the leading edge. This was attributed to convective cooling by flow through the cooling holes and flow at the inside of the model, though the model did not have any specific internal cooling scheme. It was also noted a significant drop in η between coolant holes, where ϕ did not exhibit a similar drop. This effect is contributed to three-dimensional conduction effects. Another experiment was performed on the same model where one coolant hole was plugged at $M=2.0$. Results show

a region of lower ϕ localized to the immediate vicinity of the plugged coolant hole, but there were no other effects elsewhere in the leading edge.

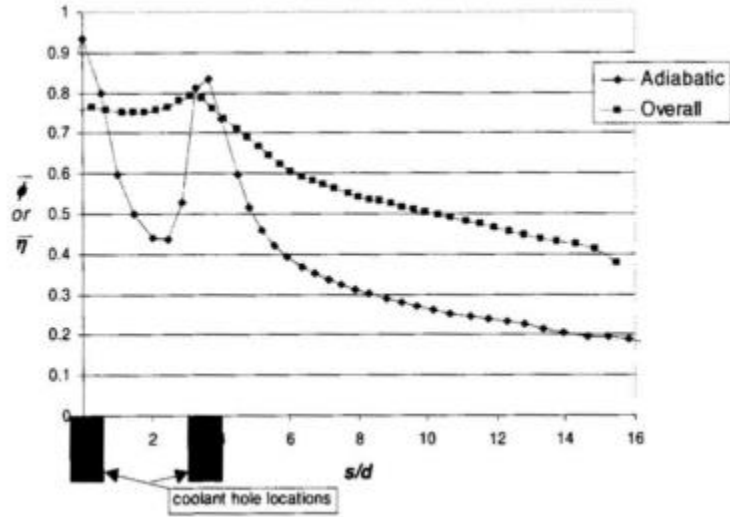


Figure 8: Comparison of ϕ and η at $M=2.6$, $DR=1.8$ [13]

Nathan et al. [14] also performed a study comparing adiabatic and overall effectiveness in just the leading edge of a turbine vane. The experiment used a matched Biot number model with impingement cooling and a full showerhead. They found overall effectiveness increased with momentum flux ratio for all test points, even those where jet liftoff clearly occurred. At high momentum flux ratios, coolant jets began to merge preventing the freestream from penetrating and significantly increasing adiabatic film effectiveness. Like Albert et al., overall effectiveness values were much higher with less spatial variation than η due to impingement cooling.

2.2 Impingement Cooling

Impingement cooling is the phenomena where a free jet hits a surface, increasing the overall heat flux due to an increased heat transfer coefficient. An example of an impingement jet is shown in Figure 9 with the jet potential core labeled.

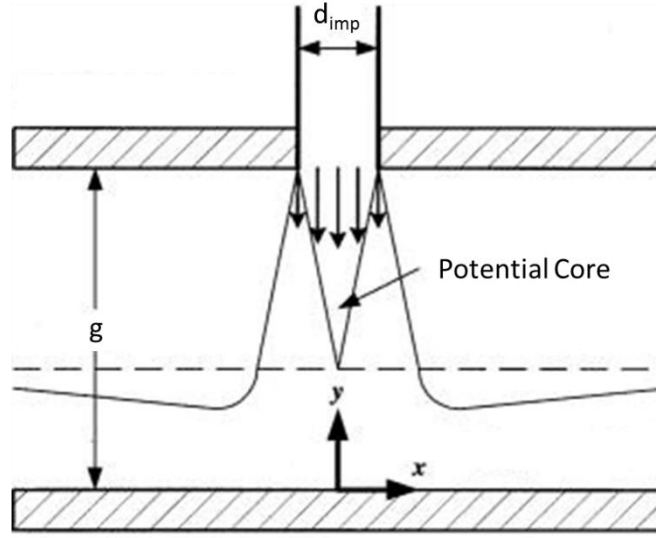


Figure 9: Impinging jet cooling geometry. Adapted from [15]

One early work on impingement by Martin et al. [16] focused on finding predictions for the performance of various impingement configurations. He created several correlations based on flow parameters and impingement geometry which predicted the effect a configuration will have on the internal Nusselt number.

$$Nu = f\left(Re, Pr, A, \frac{g}{d_{imp}}\right) \quad (2.15)$$

Cho et al. [17] did a study using an impingement plate and an effusion plate which allowed air to flow through, similarly to film cooling holes. Any conduction error was eliminated by using the naphthalene sublimation method, where a naphthalene coated

surface is impinged on and the sublimation depth is measured. Thus, the surface boundary condition is analogous to an isothermal surface. Their results for the small gaps between the two plates showed minimal jet interaction, thus the flow pattern was like duct flow. The heat/mass transfer at the stagnation point was highest at $g/d=6$, with overall heat/mass transfer rate increasing as H/d decreases. The authors also concluded that increasing the Reynolds number increased heat/mass transfer monotonically. Lastly, with a shifted array, a low momentum region grows between the effusion holes.

Lee et al. [18] did a study investigating the effects of gap spacing and Reynolds number on jet array impingement heat transfer. These two parameters were chosen to be investigated because previous research by Kercher and Tabakoff [19] indicated they are the dominating factors. The study found average Nu increases substantially with Re. The highest Nusselt numbers occurred with a gap spacing of 1.5 or 3.0. It was also found increasing the gap led to the local maximum in Nu to shift due to cross flow effects. Finally, at a gap of 8 hole diameters, Nusselt number decreased at each x/D location and with Reynolds number due to cross flow.

In gas turbine blade cooling, impingement is typically used in conjunction with film cooling. It can be used anywhere: end wall, pressure and suction sides, trailing edge, and leading edge. Mensch and Thole [20] developed a conjugate heat transfer model for the end wall of a seven-blade cascade. They tested external film cooling and internal impingement cooling separately and together. Film cooling only showed high cooling effectiveness around the hole exits, which was hypothesized to be due to convective cooling within the holes. Internal impingement cooling provided more uniform

effectiveness than film cooling and impingement effectiveness increased with blowing ratio. Combining the two cooling methods gave an overall effectiveness close to 0.4 on average.

Williams et al. [21] studied impingement effects on the suction side of a turbine vane. They measured adiabatic and overall effectiveness for a wide range of momentum flux ratios on a matched Biot number model. They found that adiabatic effectiveness decreased with increasing I due to jet liftoff, but overall effectiveness continued to increase. They blocked single film cooling holes and found the average overall effectiveness remained the same on the outside surface, called ϕ_0 . They credited increased internal cooling for compensating for the lack of coverage. They also came up with a 1D correlation for predicting the overall effectiveness due to internal cooling only

$$\phi_0 = \frac{1}{1 + Bi + \frac{h_f}{h_0}} \quad (2.16)$$

Using this correlation, they came up with a predictor for overall effectiveness based on ϕ_0 , adiabatic effectiveness, and coolant warming.

$$\phi_p = \phi_0 (1 - \chi\eta) + \chi\eta \quad (2.17)$$

Ravelli et al. [22] investigated the effects of varying the impingement on a leading edge model with showerhead film cooling. They performed both experimental and computational tests using three different impingement plates. They also tested the models without any impingement. They found overall effectiveness was only slightly improved by impingement. One theorized reason for the underwhelming effect due to impingement is

convective cooling within the coolant holes has a much greater effect than previously recognized.

Dyson et al. [23] measured adiabatic and overall effectiveness data for a fully cooled turbine vane, with internal cooling. They constructed models from both a matched Biot number material and a low thermal conductivity material. They also made a simplified model to find measurements of overall effectiveness due to internal cooling alone, ϕ_0 . They found overall effectiveness increased with momentum flux ratio in the leading edge region, even after jet separation. This was because internal cooling continued to improve, while adiabatic effectiveness remained constant. Overall, they found a 20-50% increase in overall effectiveness for full cooling vs. internal cooling only.

The relative impact of various parts of cooling has been the subject of many experiments. Rutledge et al. [24] developed the $\Delta\phi$ approach to better analyze the effect of a film cooling scheme. The approach allows for comparison of two schemes by subtracting one scheme's ϕ from another. The result is the contribution to the overall effectiveness by only the difference between the two schemes. The authors were primarily interested in the contribution of film cooling to overall effectiveness, so they assumed characteristic values for the internal heat transfer coefficient, Biot number, and coolant warming factor and input only the external heat transfer coefficient and adiabatic effectiveness distribution. Using this approach, an experiment showed $M=0.5$ was the most effective blowing ratio for increasing $\Delta\phi$ due to film cooling.

2.3 Hole Cooling

Terrell et al. [25] did a series of experiments to determine the influence of convective heat transfer through film cooling holes in a leading edge. The experiment used thermocouples to determine the coolant warming through the coolant holes. A complimentary CFD experiment found the overall effectiveness contours for a similar model. For both these models, without any impingement cooling, the results indicate cooling within the film holes accounts for 50% to 80% of the heat transfer to the external surface. They looked at similar experiments run in the same facility and concluded convective cooling in the holes would account for 30% to 70% in a model which included impingement.

Chavez et al. [26] did a similar experiment comparing the effects of internal and film cooling on the overall effectiveness on a fully cooled turbine blade. They found the large spacing of holes in the showerhead region yielded low adiabatic effectiveness, but the impingement and through hole convection provided adequate cooling. Along the stagnation line, where there is zero adiabatic film effectiveness, there was still significant overall effectiveness due to internal and convective cooling. They point to convective cooling within the holes as particularly important in the showerhead region due to the spacing of coolant holes.

These experiments show a need for more understanding as to the relative contributions of the different aspects of turbine cooling. There have been numerous studies on the effects of film cooling on the adiabatic wall effectiveness, and on the effects of

impingement on the internal surface. However, the effect of through hole cooling and the effects of varied impingement on the outer surface of the turbine material are unknown.

III. Experimental Methodology

To determine the contribution from each aspect of cooling, a physical model with thermal conductivity such that the Biot number approximately matches engine conditions was constructed and tested in a wind tunnel. The purpose of this chapter is to describe the leading edge model and testing methods.

3.1 Model

The leading edge model used in this experiment had full coverage film cooling holes, but is based on the single coolant hole geometry used by Ekkad et al. [27]. The model used a semi cylinder with a flat afterbody to replicate the leading edge region of a high pressure turbine blade. A diagram of the model is shown in Figure 10.

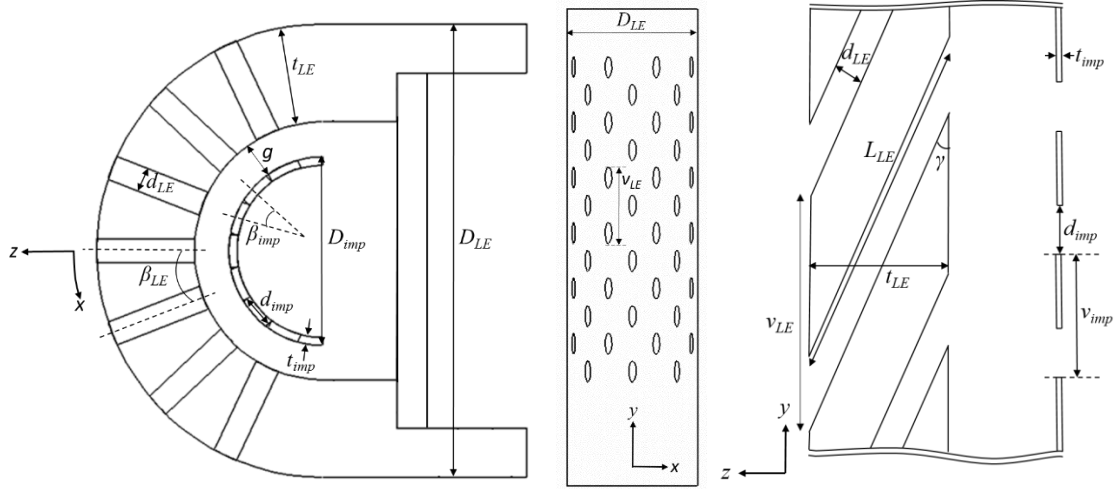


Figure 10: Top, front, and cross section views

The showerhead consists of 42 cylindrical holes in seven vertical rows at 21.5 degree intervals around the circumference. The cooling holes are arranged in a staggered array at a 20-degree angle, γ , to the surface. The diameter of the film cooling holes was 0.48 cm

and the diameter of the semi cylinder was 8.89 cm. The ratio of the leading edge to coolant hole diameters, D/d , was 18.52. The length of the coolant hole was 5.61 cm which yields a coolant hole length to diameter ratio of 11.63. The design parameters can be found in Table 1.

Table 1: Model dimensions

LEADING EDGE			
OUTER DIAMETER	D	8.89	cm
COOLANT HOLE DIAMETER	d	0.48	cm
WALL THICKNESS	t	1.93	cm
INJECTION ANGLE	γ	20	deg
ANGULAR SPACING	β	21.5	deg
HOLE SPACING	v	3.78	cm
HOLE LENGTH	L	5.61	cm

An important aspect of this model is how dense the cooling holes are. A cross section in the middle of the leading edge in the x - z plane, as in Figure 11, shows that there are multiple cooling holes between the inner and outer walls of the model. The holes overlap each other, such that the outlet and inlets of successive holes in a vertical row overlap. Additionally, the holes are in a staggered array: where the upstream row has a hole inlet and an outlet through the thickness of the leading edge material, the next row downstream has a coolant hole in the middle of the material. All these holes through the thickness of the material have coolant flowing through them, cooling the hole walls, and conducting to the outer wall. Whatever effect the impingement plates has on the internal surface must conduct past all these conducting holes to affect the external surface. The predicted local heat transfer coefficients are further discussed in Section 3.2.

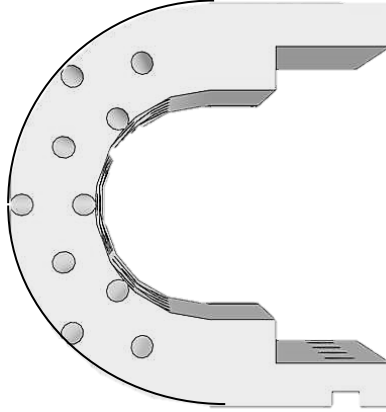


Figure 11: Cross section of leading edge showing cooling holes

The model also had the capability to accept one of four interchangeable impingement plates. Three plates consisted of an array of cooling holes, all with a 0.33 cm diameter. The ratio of leading edge coolant hole diameter to impingement plate hole diameter, d_{LE}/d_{imp} , was 1.46. The gap between the leading edge and impingement plate, as shown in the top view in Figure 10, was 0.64 cm, which resulted in a ratio of gap distance to coolant hole diameter of 1.92. Finally, the distance between holes was 3.78 cm so the ratio of hole spacing to hole diameter was 7.84. The leading edge model is identical to the model used by Wiese [9] except his model had only a single coolant hole at 21.5 degrees from the stagnation line.

Several impingement plates, shown in Figure 12, were tested. The first impingement plate used consisted of 46 holes in a staggered array. Each vertical row was spaced at 21.5 degrees circumferentially, and the holes were designed so that each hole would impinge directly between cooling hole entrances in the leading edge. Unfortunately, during the manufacturing process the design was modified and thus, the impinging holes nearly lined up exactly with the leading edge coolant hole entrances. The next plate consisted of 20

holes, also in a staggered array. The circumferential spacing for this plate was 43 degrees, but the vertical spacing remained the same. In fact, the row of holes down the centerline of the plate is identical to the 46-hole plate.

The third plate had only two holes. These were inline on the centerline, where the 3rd and 5th holes had been on the 46 and 20-hole plate. This plate was intended to produce poor impingement, to draw a contrast with the 46-hole plate, which was expected to have a much better effect. The flow was not choked travelling through only two holes. The fourth plate used consisted of four parallel vertical slots. The width of the slots was equal to the diameter of the coolant holes in the other three plates. The slot height was equal to the height of the array of 46 holes. The slots were spaced at 43 degrees circumferentially, with no slots on the centerline. A table with the dimensions of each plate is shown in Table 2 and a front view of the impingement plates is shown in Figure 12.

Table 2: Impingement plate dimensions

		PLATE A	PLATE B	PLATE C	PLATE D	
SCHEME		4 slots	46 holes	20 holes	2 holes	
OUTER DIAMETER	D	1.94	1.94	1.94	1.94	cm
HOLE DIAMETER	d	0.32	0.32	0.32	0.32	cm
WALL THICKNESS	t	0.05	0.05	0.05	0.05	cm
HOLE SPACING	y	N/A	3.78	3.78	7.57	cm
ANGULAR SPACING	β	43	21.5	43	N/A	deg
SLOT HEIGHT	h	22.64	N/A	N/A	N/A	cm

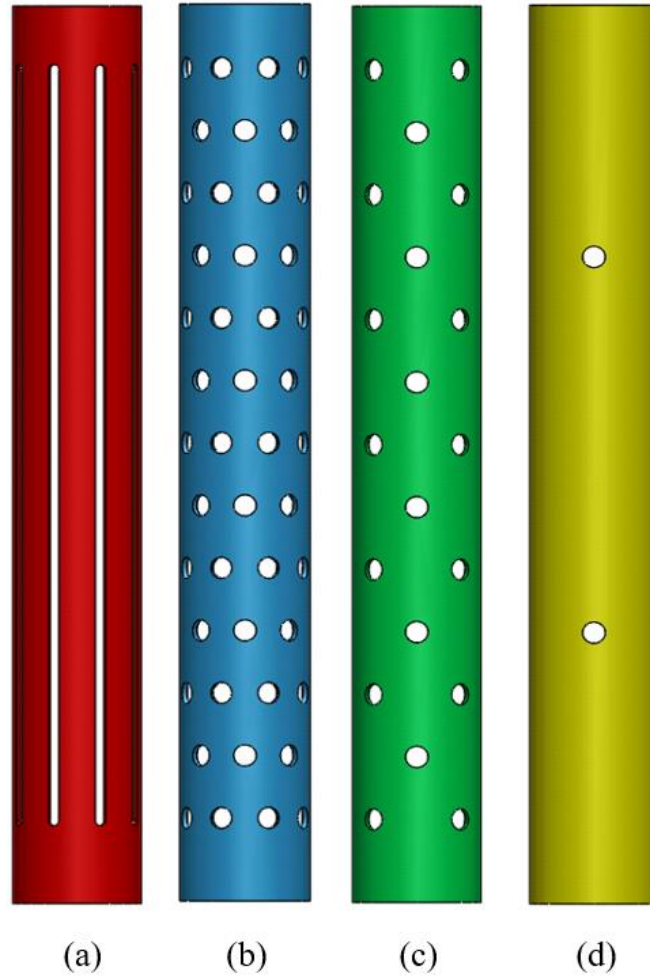


Figure 12: Schematic of the impingement plates used. Plate (a) is the slots plate, (b) is the 46-hole plate, (c) is the 20-hole plate, and (d) is the 2-hole plate.

As previously mentioned, due to a manufacturing error, the film cooling holes did not end up in the designed location. An unfortunate result of this mishap was the impingement plate exits frequently aligned with cooling hole inlets. Figure 13 shows an overlay of the impingement plates onto the film cooling hole inlet pattern, shown in black. The slots plate aligned with four of the inlet rows. The 46-hole plate aligned with all the cooling inlets except for the top row. The 20-hole plate only aligned with the center row,

except for the top inlet. The two other rows of impinging holes align between cooling inlets, as it was designed. And finally, the two-hole plate aligns with two center row inlets.

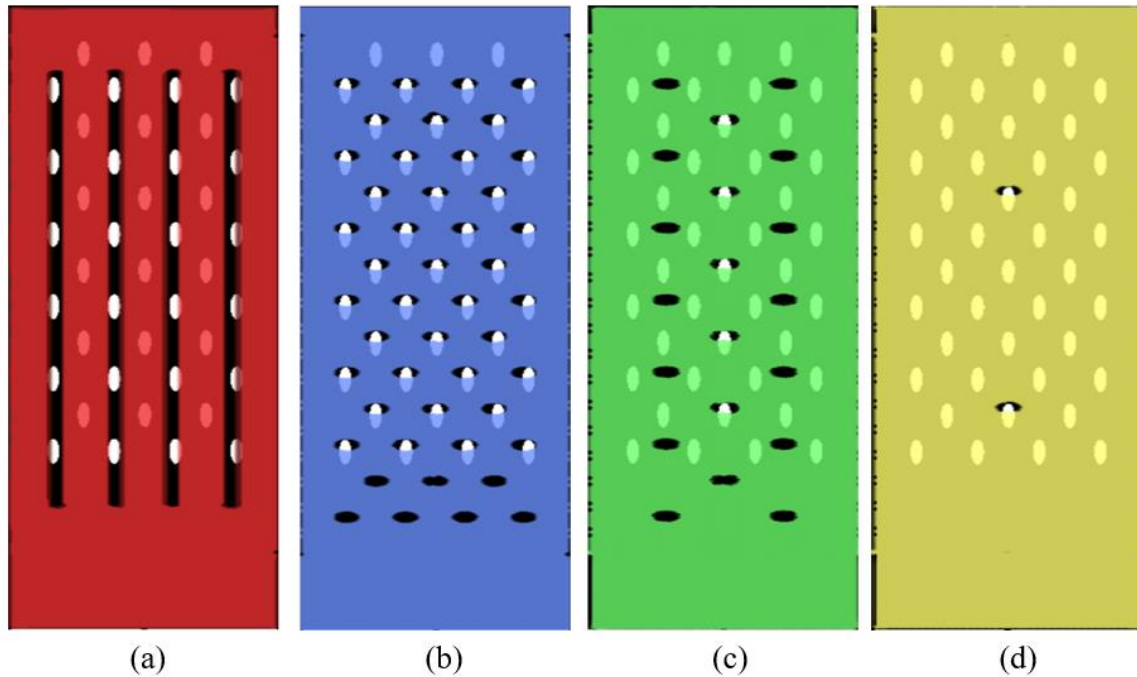


Figure 13: Impingement plate hole overlaps with cooling hole inlets. Plate (a) is the slots plate, (b) is the 46-hole plate, (c) is the 20-hole plate, and (d) is the 2-hole plate.

The assembled model consisted of several separate parts. The top and bottom of the plenum were created by 0.95 cm thick plates which slid into grooves in the leading edge. Two spacers 0.635 cm wide clipped to the impingement plates at the top and bottom to keep the plate at the correct gap from the leading edge. A back plate 0.51 cm thick sealed the plenum by four screws into the end caps and leading edge sides. The impingement plate sides were separated from the back plate by 1.45 cm on both sides.

Coolant was fed to the plenum by a tube running through the center. The tube had air flowing from both the top and bottom via a split in the feed line. The tube within the

plenum was full of holes at regular intervals to get uniform flow. Before testing began, porous foam was added to the center of the tube to increase the pressure requirement and encourage a more even flow distribution. A back view of the partially assembled model is shown in Figure 14. The leading edge is shown in gray, end caps are in yellow, the impingement plate is blue, the spacers are red, the back plate is pink, and the coolant feed is green.

The leading edge, end plates, and back plate were made of DuPont Corian® material, to match the Biot number. Numerous studies have used Corian before, such as Mensch and Thole [20] and Nathan et al. [14]. The material properties are known from an internal study conducted by DuPont and are: $\rho = 1700 \text{ kg/m}^3$, $c_p = 1379 \text{ J/kg-K}$, and $k = 1.009 \text{ W/m-K}$. All the impingement plates were made of 0.05 cm thick aluminum. The spacers were made of acrylonitrile butadiene styrene, or ABS, using a Makerbot 3D printer. The coolant feed loop was made of 0.95 cm diameter copper tubing. The entire model was painted matte black for uniform emissivity.

Several J-type thermocouples were placed on the model to obtain temperatures at specific locations. One thermocouple, T_{cp} , was placed in the plenum, directly where the coolant enters the plenum. Another thermocouple, T_{ci} , was placed at the inlet to a film cooling hole at 43 degrees from the stagnation row. Another thermocouple, T_{ce} , was placed at the exit of a different film cooling hole, also at 43 degrees from the stagnation row. Six more thermocouples were placed on the model: three on the inside of the leading edge surface, between coolant hole inlets, and three on the outside of the leading edge between coolant hole exits. All the thermocouples were placed on the side of the model opposite of

the IR camera. The thermocouple locations are shown below in Figure 15, with the area viewed by the IR camera, used to measure surface temperature, shown in a red square. A top view of the thermocouple locations is shown in Figure 16. Note the thermocouples inside and outside thermocouples are attached to the walls while the coolant in the plenum thermocouple is unattached, yet close to the soaker hose.

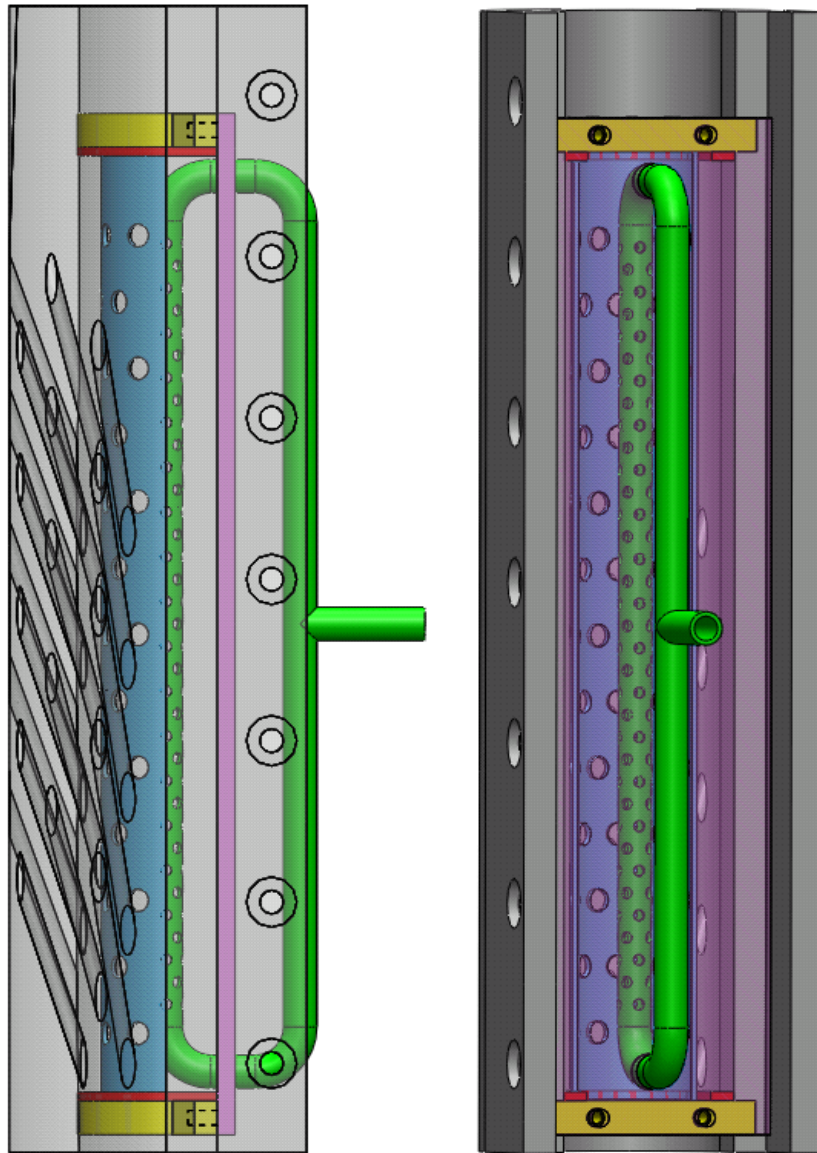


Figure 14: Assembled model

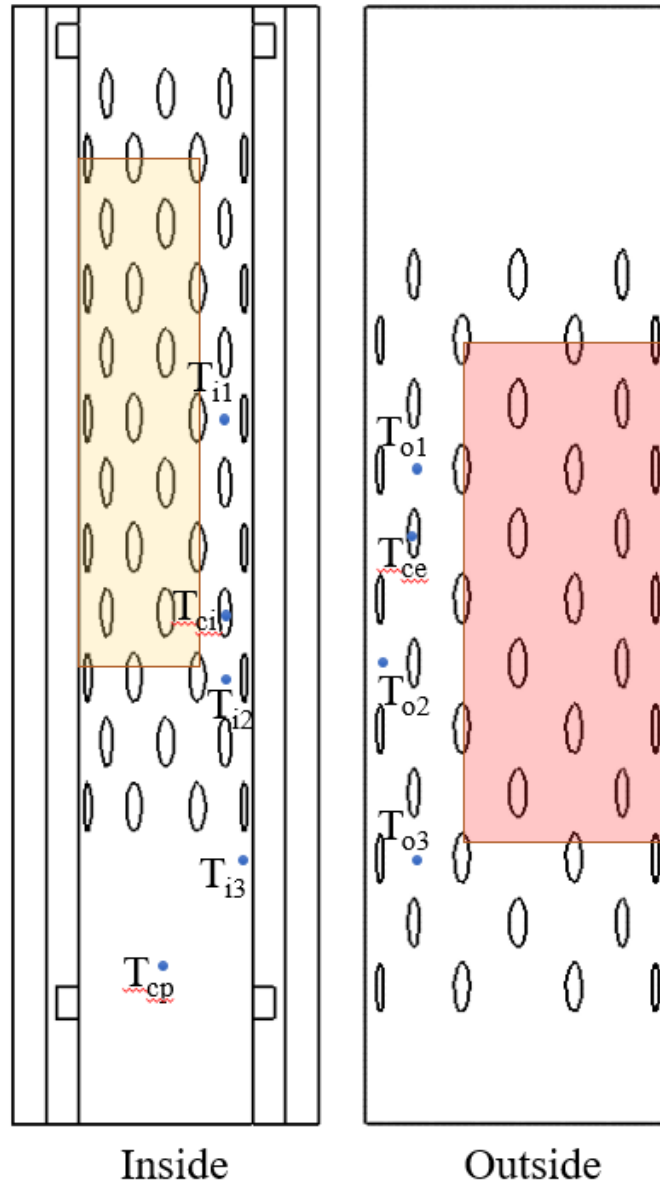


Figure 15: Thermocouple locations, with area viewed by IR camera marked by a red rectangle, and corresponding inlet area inside marked by a yellow rectangle

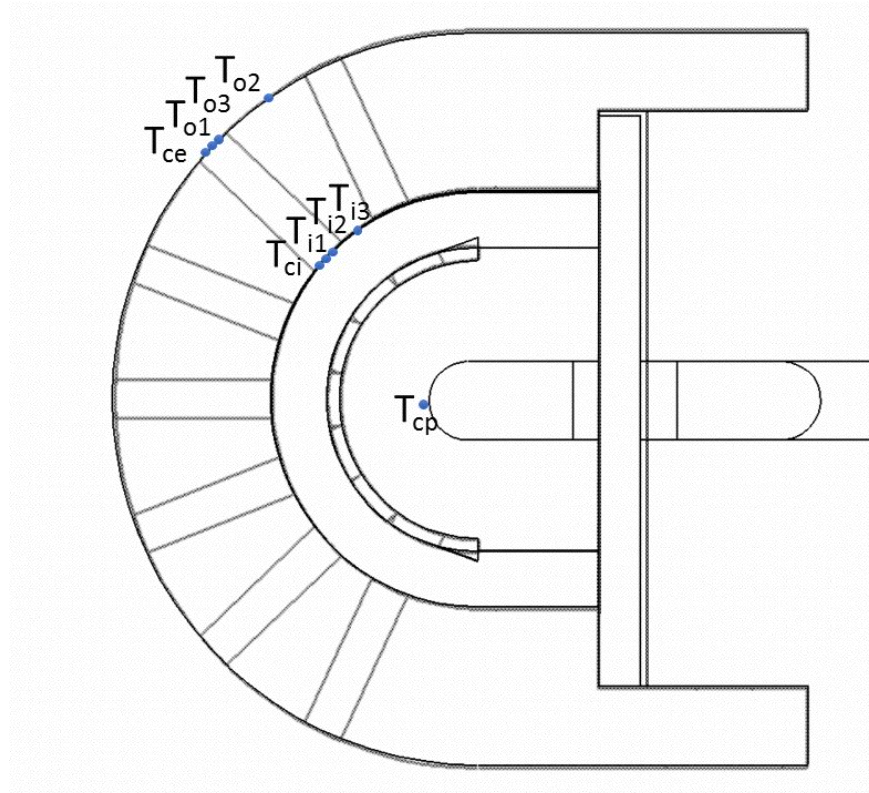


Figure 16: Top view of thermocouple locations

3.2 Predicted Local Heat Transfer Coefficients

The heat transfer within the cooling holes was approximated using convection heat transfer theory. Several assumptions were made to simplify the governing equations. First, the flow was assumed to be fully developed, both hydrodynamically and thermally. Second, the walls are assumed to apply a constant heat rate to the flow, like a heat exchanger. This assumption was made because the walls are not a constant temperature, the other standard boundary condition, instead they became warmer the closer to the freestream surface due to conduction. Next, the flow was assumed to be turbulent. Finally,

the flow was assumed to have constant properties. Per Kays and Crawford [28] such flow in a circular tube will have a constant Nusselt number approximated by

$$Nu = 0.022Pr^{0.5}Re^{0.8} \quad (3.1)$$

The local heat transfer coefficient was calculated from the Nusselt number using the definition of the Nusselt number (Equation 2.6).

The local heat transfer coefficients from each impingement plate were calculated using correlations provided by Incropera et al. [29] based on work by Martin et al. [30]. The correlations are based on flow characteristics and the geometry of the impingement plates themselves. The correlations used were for three different types of geometries: staggered arrays of round holes for the 46 and 20 hole plates, in-line array of round holes for the 2-hole plate, and an array of slots for the slots plate. Figure 17 shows the pertinent geometrical features for each of these cases.

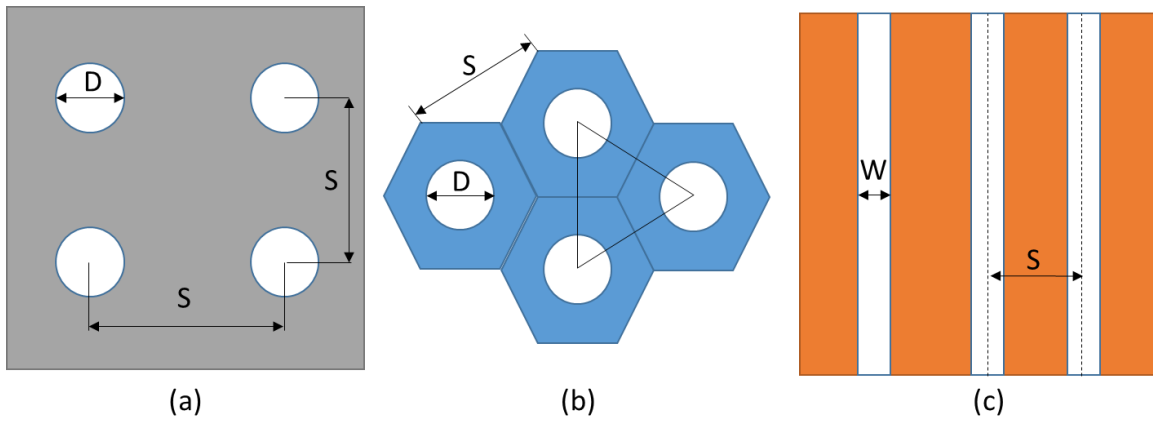


Figure 17: Impingement plate geometries for (a) an in-line array of round holes, (b) a staggered array of round hole, and (c) and array of slots

For a staggered array of round nozzles, the given correlation is:

$$\frac{\overline{Nu}}{Pr^{0.42}} = 0.5K \left(A_r, \frac{H}{D} \right) G \left(A_r, \frac{H}{D} \right) Re^{2/3} \quad (3.2)$$

where

$$K = \left[1 + \left(\frac{H/D}{0.6/A_r^{1/2}} \right)^6 \right]^{-0.05} \quad (3.3)$$

$$G = 2A_r^{1/2} \frac{1 - 2.2A_r^{1/2}}{1 + 0.2 \left(\frac{H}{D} - 6 \right) A_r^{1/2}} \quad (3.4)$$

$$A_r = \frac{\pi D^2}{2\sqrt{3}S^2} \quad (3.5)$$

For an in-line array of round nozzles, the same correlation applies. The only difference is the definition of A_r .

$$A_r = \frac{\pi D^2}{4S^2} \quad (3.6)$$

For an array of slot nozzles, a different correlation applies.

$$\frac{\overline{Nu}}{Pr} = \frac{2}{3} A_{r,o}^{3/4} \left(\frac{2 Re}{A_r/A_{r,o} + A_{r,o}/A_r} \right)^{2/3} \quad (3.7)$$

where

$$A_{r,o} = \left[60 + 4 \left(\frac{H}{2W} - 2 \right)^2 \right]^{-1/2} \quad (3.8)$$

$$A_r = W / S \quad (3.9)$$

The Nusselt numbers within the cooling holes and for each impingement condition were calculated at the three blowing ratios tested, $M=0.25$, $M=0.5$, and $M=0.9$. The local heat transfer coefficients were again calculated using the definition of the Nusselt number. They are shown below in Figure 18.

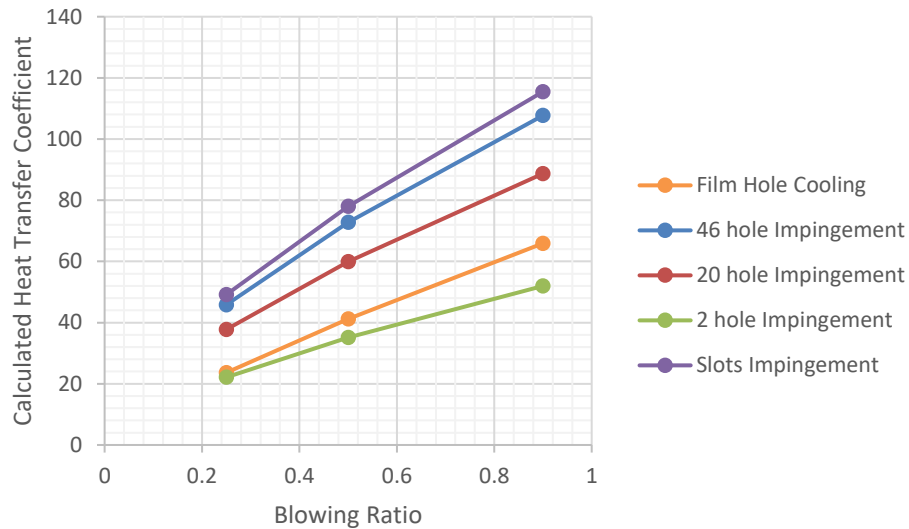


Figure 18: Calculated heat transfer for impingement and hole cooling

The heat transfer coefficient, h , within the cooling holes is similar to h for the 2-hole impingement plate. However, heat transfer within the holes is expected to be more influential on overall cooling because the cooling in the holes affects a greater surface area than the impingement plate. The sum of the surface areas of the cooling holes is 355.3 cm^2 , while the inner surface of the Corian is only 99.7 cm^2 . The cooling in the holes effects 3.5 times more surface as impingement effects. Additionally, per Figure 11 which shows the cross section of the model, the cooling holes are closer to the outer Corian surface, and will therefore conduct to the outer surface better than the inside surface.

3.3 Facility

The wind tunnel used in this study was an open loop blow down tunnel powered by two 50 hp blowers. A valve controlled the flow velocity immediately downstream of the blowers. The freestream temperature range was measured to be 320-330 K. Coolant was supplied to the model by a shop air line pressurized at 100-125 psig. The coolant rate was controlled and measured by an Omega FMA-2612A volumetric gas flow controller, with a maximum flow of 500 standard liters per minute. The coolant temperature range without any modifications was found to be 298-303 K. A diagram of the wind tunnel with the coolant flow path is shown in Figure 19.

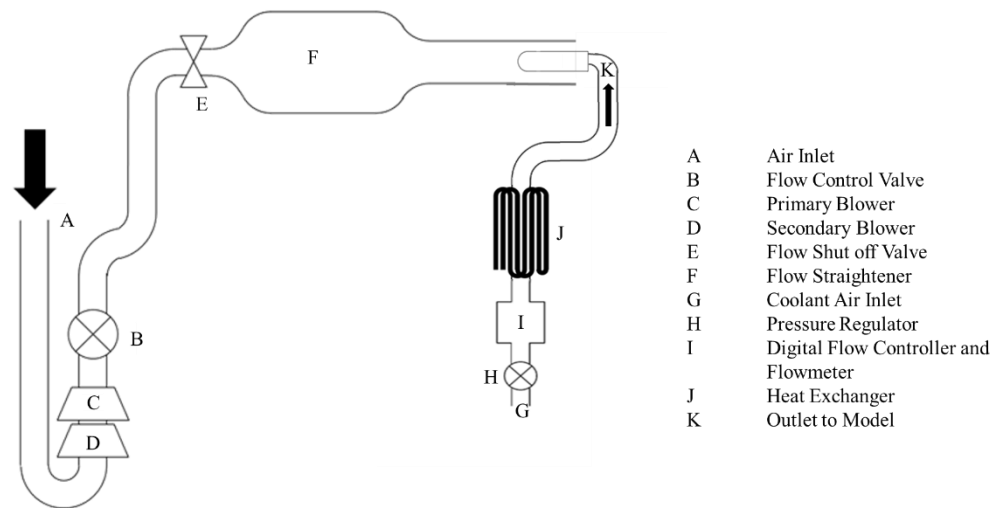


Figure 19: Wind Tunnel and Coolant diagram. Adapted from [9]

The coolant temperature could be adjusted using a counter flow, tube-in-tube heat exchanger. The heat exchanger fluid was a mixture of ethylene glycol and water, capable of a temperature range of 253-373 K. The coolant temperature range was found to be 270-320 K with the heat exchanger.

The tunnel test section was constructed out of clear acrylic measuring 36.8 cm by 40.6 cm. Flow velocity was determined using a pitot static probe 74 cm upstream of the model. The pitot-static pressure was measured by an GE Druck LPM 9381 pressure transducer, powered by a Tenma DC power supply set at 12 Volts and 0.01 Amps. The freestream density was determined using the known static pressure and T_∞ , found using a J-type thermocouple. The ambient pressure was measured at the beginning of each test and assumed to be constant for the duration of the test. The freestream viscosity μ , specific heat c_p , and thermal conductivity k , were determined via linear interpolation from published data sets based on T_∞ . The IR camera could view all the experiments through a sapphire window, as shown below in Figure 20. A picture of the test section set up is shown in Figure 21. This shows the location of the IR camera relative to the leading model. Note the test section is mostly blacked out to get rid of any reflections to the IR camera.

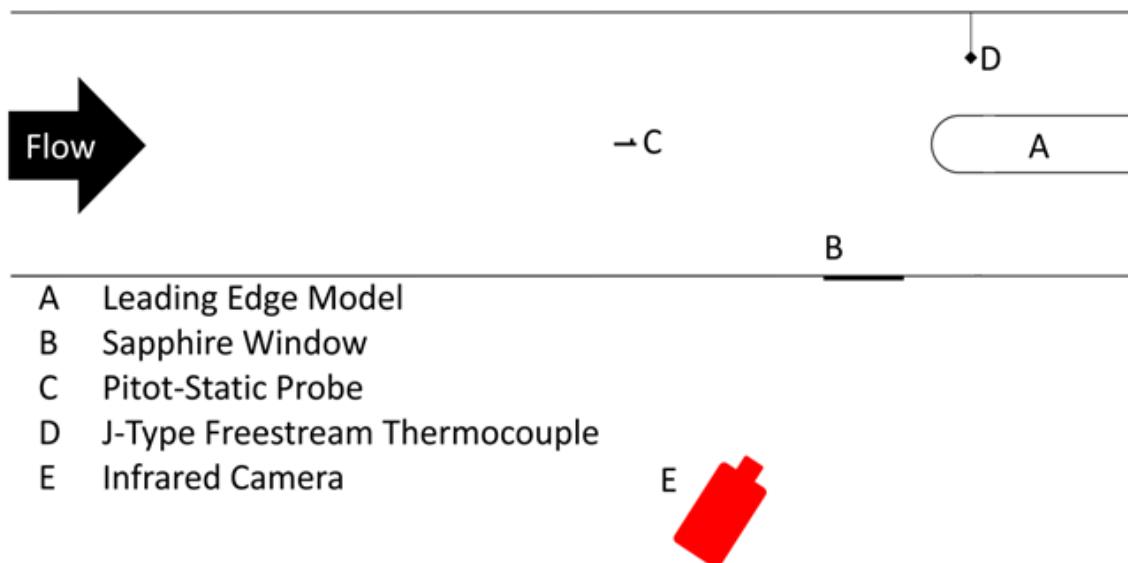


Figure 20: Test section schematic. Adapted from [9]

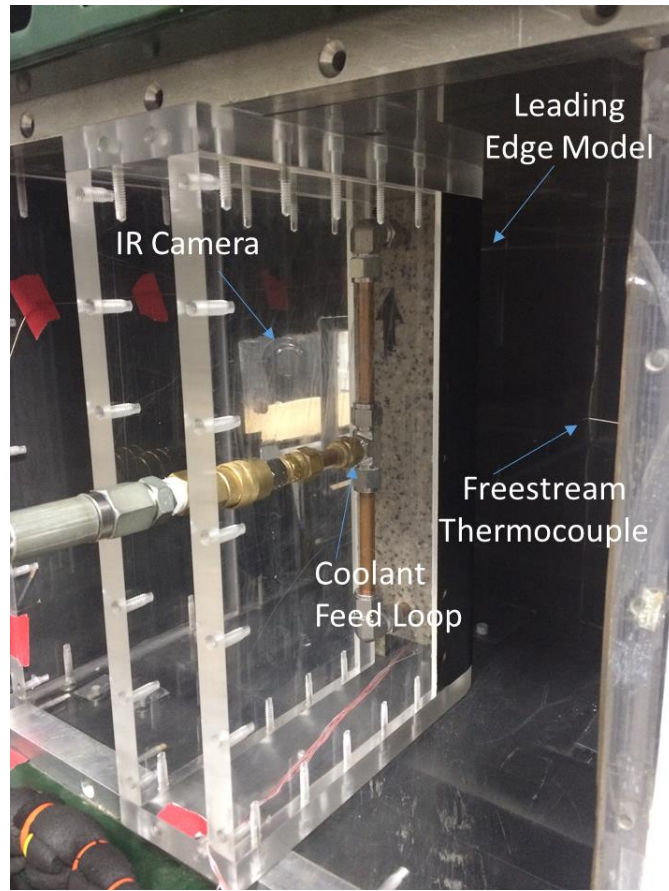


Figure 21: Test section set up viewed from behind

3.4 Test Procedures

Once the wind tunnel was set up for testing, the model itself was adjusted and the tests were completed. Each test required a slightly different modification to the leading edge model. Setting up the leading edge model was the first step. For the first set of tests, several hole inlets were blocked by placing tape over the inlets. The second set of tests, several impingement plates were installed and tested. The plates were inserted into the plenum and kept the desired distance from the inner Corian wall by spacers. One problem was the gap between the end of the impingement plates and the back plate, shown in Figure 14. The

impingement plate did not extend all the way around to the back plate, so there was no way to ensure all the coolant went through the impingement plate. To fix this issue, two wide pieces of electrical tape were placed on the outside of the plates, and the other side of the tape stuck to the corner, where the back plate joined the leading edge. The tape ensured no coolant could flow around the sides of the plate.

Once the necessary changes had been made, the plenum was sealed by screwing the back plate onto the model with four screws. The model was then secured to an acrylic frame, and the entire assembly was installed in the tunnel using three screws. All the model thermocouples were then plugged into J-type thermocouple extenders which were connected to the data acquisition computer.

Once the model was installed, the IR camera was turned on and focused. The camera used was a FLIR SC8340HD. It was connected to the computer via a Cat5E cable and was remotely triggered via a BNC cable. The camera was focused using a metal ruler placed on the surface of the model. When the fine print on the ruler could be clearly seen, the camera was in focus. The area viewed by the camera was 15.2 cm by 8.6 cm, as large as possible in the vertical direction and viewed the entire left side model in the horizontal direction. The area viewed by the IR camera is shown in Figure 15.

Next the flow controller, pressure sensor, and power supply were turned on. The pitot-static pressure was zeroed. The shop air line was opened and pressurized. Finally, the tunnel shut off valve was opened, and both blowers were turned on. The flow control valve was used to adjust the freestream Reynolds number to 60,000 for all tests.

For each test point the coolant blowing ratio was set and held constant for 10 minutes before data was recorded, to allow the model to reach thermal equilibrium. The transient thermal characteristics of the model were experimentally determined by taking data every ten seconds, immediately after starting coolant flow. The temperature at three places inside and outside the model were taken from thermocouples on the model, as shown in Figure 15 (T_{i1} , T_{i2} , T_{i3} and T_{o1} , T_{o2} , T_{o3}). The transient thermal results are shown in Figure 22. The model reached a steady thermal profile after 10 minutes, so that time was used for all tests.

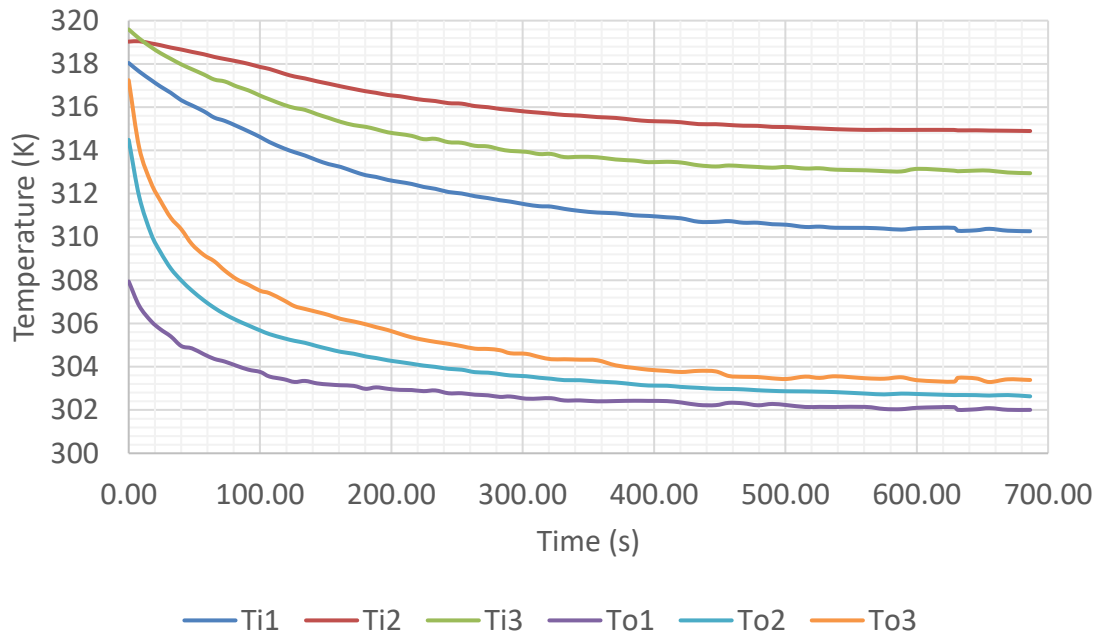


Figure 22: Transient thermal characteristics

After the desired time had elapsed, data was collected from the IR camera and a LabView system. A single frame of temperature data from the IR camera was recorded. The LabView system recorded all the thermocouple temperatures as well as the coolant flow rate and the Reynolds number. The coolant flow rate was then adjusted and the

thermal transient time elapsed before the next data point was recorded. After all the desired coolant flow rates had been tested for a configuration, the model was removed from the tunnel and reconfigured, then reinstalled and testing continued.

3.5 Calibrations

The IR camera was calibrated using a separate leading edge model. The calibration model had an empty plenum, or no impingement plate, and was made from thin stainless steel shim stock. It had the same outer diameter as the leading edge model tested. The steel had six J-type thermocouples adhered to the outer surface.

The calibration was conducted by adjusting the model surface temperature using the coolant temperature. The IR and thermocouple derived temperatures were recorded for the full range of model surface temperatures experienced during testing. The temperatures were collected in 1 K increments, with multiple measurements taken at each point. The IR data was recorded as counts, which was then converted to temperatures, and were fit to the thermocouple measurements using a second-order polynomial, shown in Figure 23. The IR calibration did not include a background image, or anyway to account for day to day ambient variations.

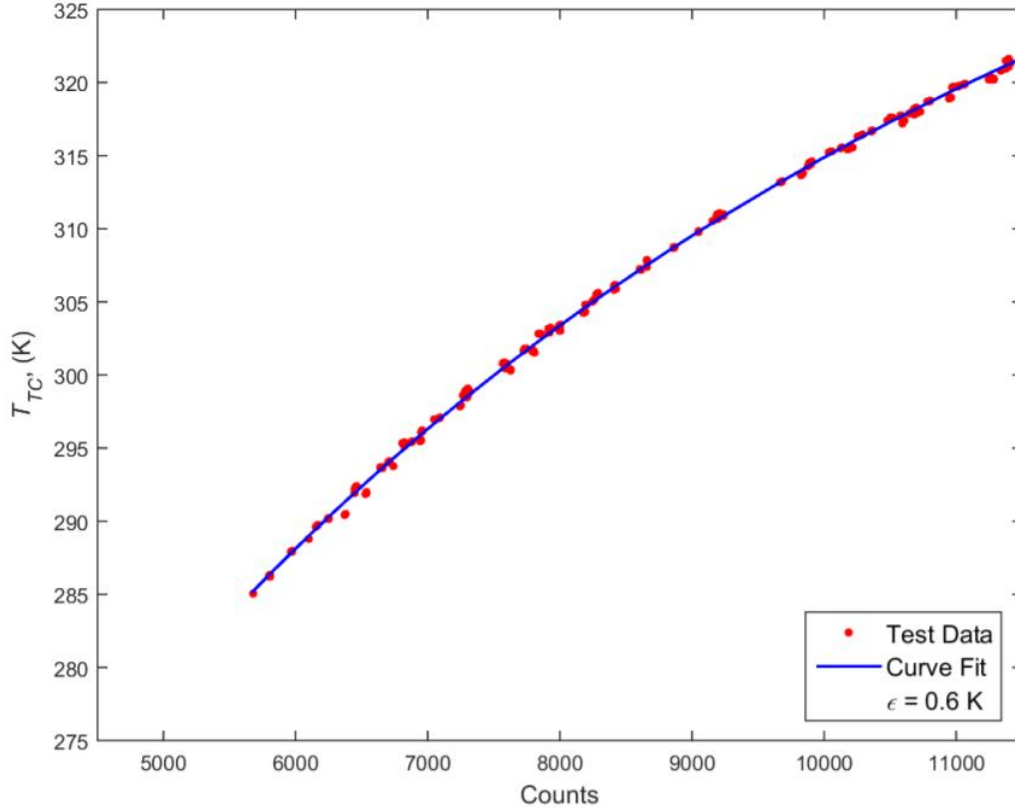


Figure 23: Temperature Calibration

The calibration was for a temperature range of 285 to 322 K. The temperatures tests were conducted at usually ranged from 300 to 320 K. The test data for the range of temperatures is in good agreement with the curve fit polynomial. The range with the greatest variance from the calibration is from 290 to 300 K, the errors are likely due to data taken before the thermal transient time had completely elapsed. The curve fit has an overall uncertainty of $\epsilon = 0.6$ K; therefore, the IR camera is well calibrated for the temperatures tested.

A spatial calibration was used to orient the data taken by the IR camera. To mark locations on the model surface, a tape was constructed using steel pins, to be thermally visible. The tape was then affixed to the model surface, directly above a cooling hole on

the stagnation line of the model. The locations of the pins in the x direction were fit to a third order polynomial to account for the surface curvature. The pins in the y direction were assumed to have constant spacing. Figure 24 shows the calibration with the pins highlighted with red ovals. The origin is in the yellow filled oval. The coolant hole exit on the stagnation row is highlighted in a green oval; the other exits are shown with white ovals. The image itself is a grayscale with white representing warm areas. These areas were warmed with a heat gun, to ensure the pins and coolant hole exits were clearly seen by the IR camera.

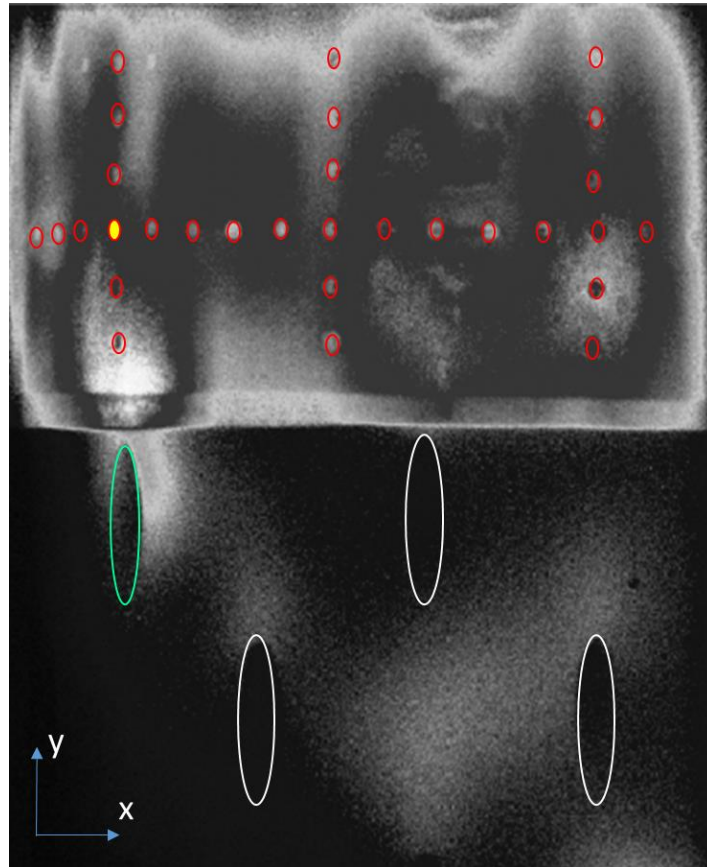


Figure 24: Spatial calibration

3.6 Uncertainty

Experimental uncertainty was characterized using the method pioneered by Kline and McClintock [31]. They assign R to be the result of an experiment, which is mathematically related to N factors X , as shown in Equation 3.1. The partial derivative with respect to each factor is the sensitivity coefficient. The root sum square of each of these factors is the overall uncertainty in R (Equation 3.2).

$$R = R(X_1, X_2, X_3, \dots, X_N) \quad (3.1)$$

$$\varepsilon_R = \left\{ \sum_{i=1}^N \left(\frac{\partial R}{\partial X_i} \varepsilon_{X_i} \right)^2 \right\}^{1/2} \quad (3.2)$$

The uncertainty in freestream Reynolds number was characterized by Rutledge [32]. The primary source of Reynolds number uncertainty was measurement using the pitot-static system. The error characterized to be 1%. Additionally, the Re was held within 2% of the desired value for all tests. Therefore, the maximum possible variation in Re is 3%.

The coolant blowing ratio was calculated using the actual calculated values of the freestream conditions. The flowmeter accuracy reported by the manufacturer was $\varepsilon_{\dot{m}} = \pm (0.8\% \text{ of reading} + 0.2 \text{ SLPM})$. All the tests were run at a flowrate above 50 SLPM which corresponds to a blowing ratio of $M=0.1$ and an uncertainty of 1.2%. For most coolant flow rates, the uncertainty was less than 0.9%, as shown in Figure 25.

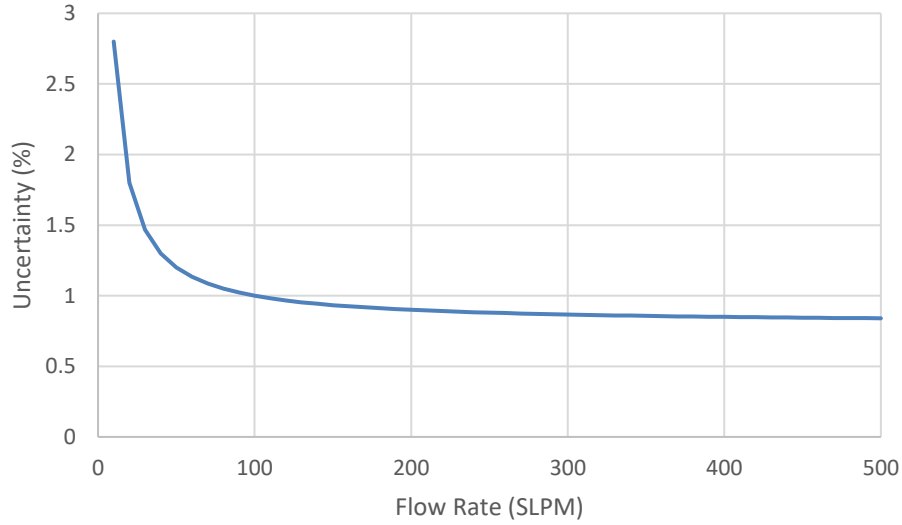


Figure 25: Uncertainty in coolant flow rate

Uncertainty in overall effectiveness was directly influenced by the surface temperature distribution. The temperature of the surface was determined using the IR camera, which, as previously discussed had $\varepsilon_{T_s} = 0.6$ K. Thermocouple uncertainty was determined by Wiese [9] to be 0.3 K, which is similar in magnitude to the reported 0.3 K uncertainty reported by Rutledge [32]. Using Kline and McClintock's method the total uncertainty in phi is $\varepsilon = 0.031$. The uncertainty in the freestream and coolant temperature is identical to thermocouple uncertainty because thermocouples directly measured them.

3.7 Repeatability

To establish the reliability of the experimental results, the exact same case was tested every day data was collected, in addition to the other test cases, for a total of six days of repeatability. This was simply the baseline case at all three blowing ratios: $M=0.25$, 0.5, and 0.9. The overall effectiveness data was reduced, then the standard deviation between

every day's baseline case was computed. To find the acceptable margin, the 95% confidence interval was computed using 1.96 standard deviations for the six days of testing.

$$95\% \text{ C.I. } = 1.96 * \sigma \quad (3.12)$$

Figure 26 shows this value at each point on the model. The legend is the half width of the 95% confidence interval in overall effectiveness. Usually, the 95% confidence interval is presented as a number and a half width, i.e. 5 ± 0.2 , or as (4.8, 5.2). In Figure 26, 0.2 would be the number plotted. For example, at $M=0.9$, below the cooling hole at $x/d=0, y.d=8$, the half width of the 95% confidence interval in overall effectiveness is 0.01-0.02. However, at $M=0.25$, the same area now has a confidence interval half width of 0.07-0.08. The center of the field of view at $M=0.25$ has the most variance. Not allowing the model to fully come to thermal equilibrium, or variance in each day's conditions due to changing weather may have contributed to the increases in variance.

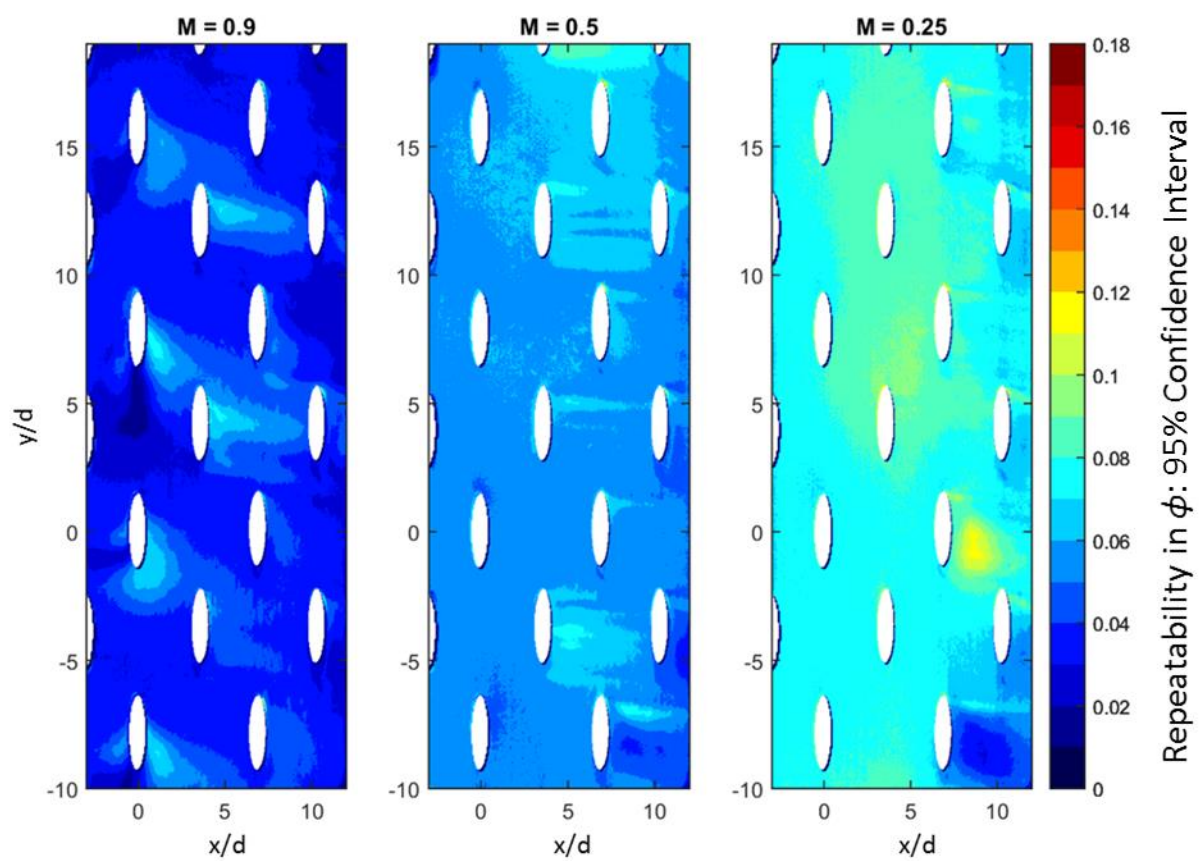


Figure 26: Repeatability for all days of testing

IV. Experimental Results

All the experimental results were processed using FLIR's image processing software and MATLAB. Much of the MATLAB code used was a slightly modified version of the code created by Wiese [9] for creating adiabatic effectiveness distributions using a very similar experimental setup.

4.1 Baseline effectiveness and freestream ingestion

The model was tested with various modifications, but always at the same three blowing ratios. The first ratio tested was $M=0.9$, chosen because it was the maximum mass flow the coolant flow meter/controller could accurately measure. The second blowing ratio was $M=0.5$, which per previous experimental work [33] is the most effective blowing ratio for adiabatic film effectiveness on the second row of cooling holes from the stagnation row, using an identical leading edge geometry and the same Reynolds number. It was decided the last blowing ratio tested should be less than $M=0.5$ to see the effects of the film and hole cooling at another lower blowing ratio, so $M=0.25$ was the last point tested. Only three blowing ratios were tested because of the lengthy transient thermal time, 10 minutes for each test point.

Future tests should be conducted at higher blowing ratios to test at conditions for the highest adiabatic film effectiveness. At higher blowing ratios, the film from one hole lifts off and covers the exit for the next cooling hole downstream. A full showerhead of film cooling holes will form a barrier of coolant between the surface and the freestream in these conditions [34]. The low blowing ratios tested in this research are not high enough for this effect to occur.

The first step in collecting experimental data was to establish a baseline. The baseline for these tests consisted of the leading edge model with full film cooling and an empty plenum, i.e. no impingement plate. The surface temperature was recorded using an IR camera. That data was recorded as counts which was read into MATLAB and nondimensionalized to overall effectiveness using Equation 2.4. Overall effectiveness close to 1 corresponded to a value where the temperature of the surface was nearly identical to the coolant temperature. A value of zero is where the surface temperature is equal to the freestream temperature. For the overall effectiveness plots, the leading edge's stagnation row is at $x/d=0$ on the horizontal axis. $y/d=0$ on the vertical axis occurs at the center of the showerhead cooling hole exits. Figure 27 shows a set of baseline ϕ contours.

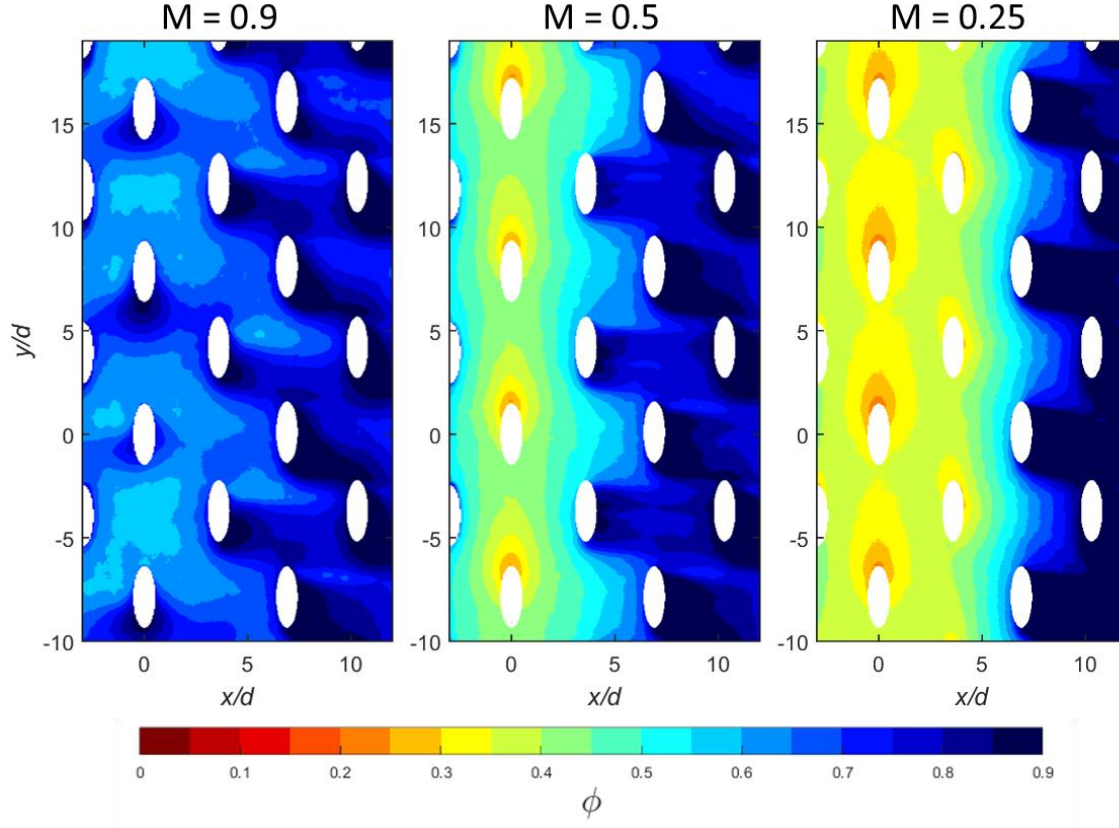


Figure 27: Overall effectiveness contours for the baseline condition

The left plot shows overall effectiveness contours at $M=0.9$. Overall, the coolant flow is fairly uniform. The lowest effectiveness is between coolant holes on the stagnation row, where the freestream impacts the surface. Further downstream, ϕ increases as the coolant film builds. The middle plot is of overall effectiveness at $M=0.5$. Here a marked difference from $M=0.9$ is the lower effectiveness on the stagnation row. In this row, none of the holes are blowing because they are all ingesting freestream. Each coolant hole is normal to the freestream, at a 20-degree angle to the outer surface. As the hot freestream travels through the coolant holes, the cooling hole walls are heated, which is then conducted to the top surface. This is visible directly above each coolant hole, where the

Corian is thinnest. In the right plot, at $M=0.25$, coolant is ingested in three rows: the stagnation row and each row at 21.5 degrees away on each side. Only the two cooling rows on the far outside of the leading edge are still blowing. This results in a low cooling effectiveness in the stagnation area, with heating of the top surface due to ingestion evident above each cooling hole exit.

To further investigate freestream ingestion, a study was conducted to find the ‘critical blowing ratio’ for both the stagnation and 21.5 degree rows of holes. The critical blowing ratio was designated as the rate where the entire row was neither blowing coolant nor ingesting freestream air. The study was conducted by making slight adjustments to the coolant blowing ratio, waiting for the thermal transient time, and observing whether the coolant holes were ingesting or blowing. A coolant hole was determined to be blowing if there was an area of higher effectiveness downstream of the coolant hole. A hole was ingesting freestream if the area above the coolant hole had decreased effectiveness due to conduction to the top surface from the warmed coolant hole.

The critical blowing ratios were determined to be $M=0.64$ for the stagnation row and $M=0.32$ for the row at 21.5 degrees. Overall effectiveness contours for both blowing ratios are shown in Figure 28. Unfortunately, the coolant distribution in the plenum was not even, so a rough average of the holes determined the critical blowing ratio. The takeaway from this study is the critical blowing ratios are between the blowing ratios tested, so ingestion is to be expected in one row at $M=0.5$ and two rows at $M=0.25$.

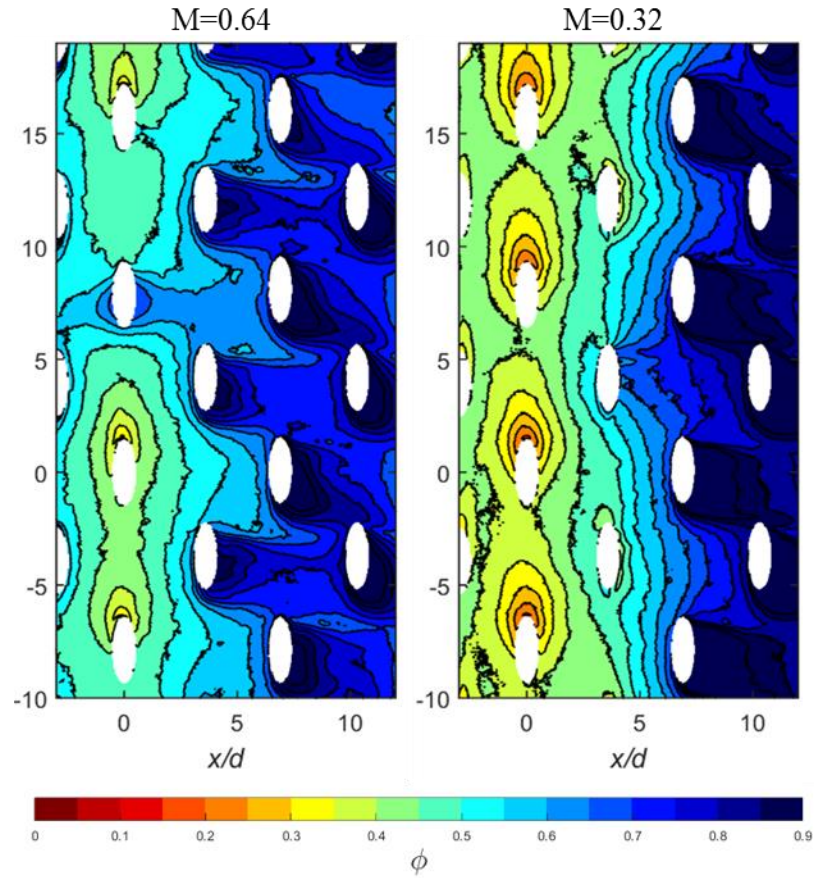


Figure 28: Overall effectiveness contours for the critical blowing ratio at the stagnation row and the row at 21.5 degrees

4.2 Blocked Coolant Holes

Once the baseline was established, the effects of various modifications to the leading edge were tested. The first parameter tested was the effect of blocking certain cooling holes. By blocking a few holes, the effect of film cooling and hole cooling for just those holes could be determined. Holes were blocked by putting tape over the coolant hole inlet, in the plenum. Testing was conducted after the model's baseline effectiveness data had been gathered for each testing day. The assembled model was removed from the tunnel and

dismantled, tape was placed over the appropriate holes, and the model was assembled and reinstalled in the tunnel. The blowing ratio calculations were adjusted for the reduced number of holes. After effectiveness data was gathered for all three blowing ratios, the data was reduced.

Rutledge's $\Delta\phi$ method [24] was used to compare the baseline case and the modified case. Figure 29 shows how $\Delta\phi$ contour plots were created, by subtracting the baseline ϕ distribution from the modified ϕ distribution. A $\Delta\phi$ value of zero is an area where the modification has no effect. A positive $\Delta\phi$ is indicated where the modification makes the area cooler, while a negative $\Delta\phi$ is a hotter area due to the modification. Because $\Delta\phi$ is just a modification of the nondimensional surface temperature, the exact increase in surface temperature due to blocking certain coolant holes can be determined by multiplying the $\Delta\phi$ value by the difference between the freestream and coolant temperatures. The details of Figure 29 are discussed later.

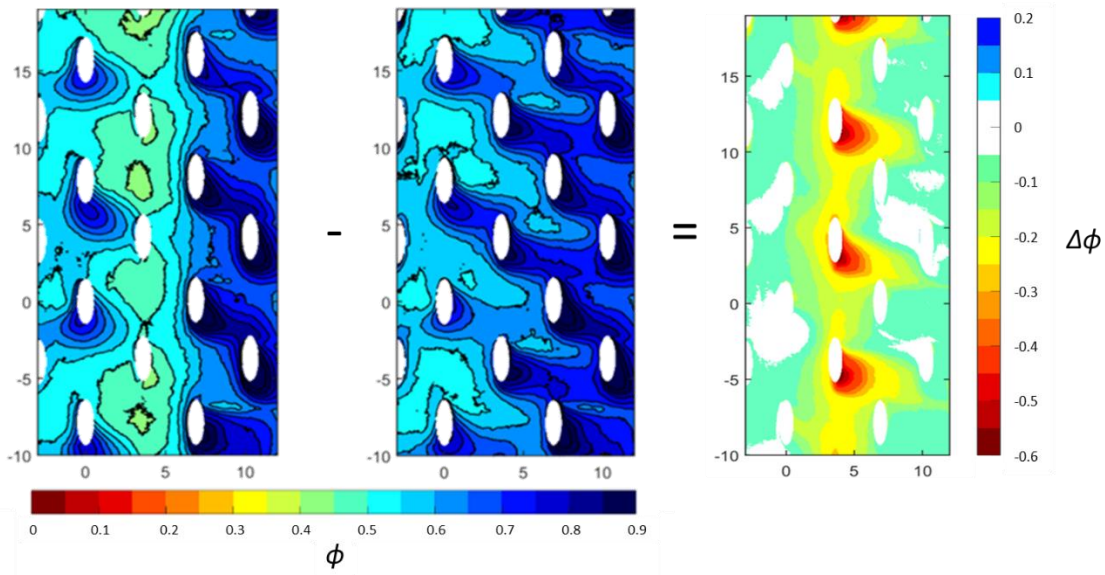


Figure 29: Modified model ϕ – baseline model $\phi = \Delta\phi$

The first modification tested was blocking the coolant holes at $x/d=4$. Before testing, the coolant flow rate was adjusted so that the blowing ratio was computed and controlled for only 36 coolant holes, because six holes were taped over. Figure 30 shows the overall effectiveness contours for all blowing ratios. Note that in all three plots, there is no blowing or ingestion for this row. Because there is no ingestion in that row, at $M=0.25$, there is higher effectiveness than in the baseline case (see Figure 27). One interesting effect is because only the one row of holes was blocked, creating an asymmetry on the leading edge model, the effectiveness contours on the stagnation line shift to the left because ingestion is still occurring on that side.

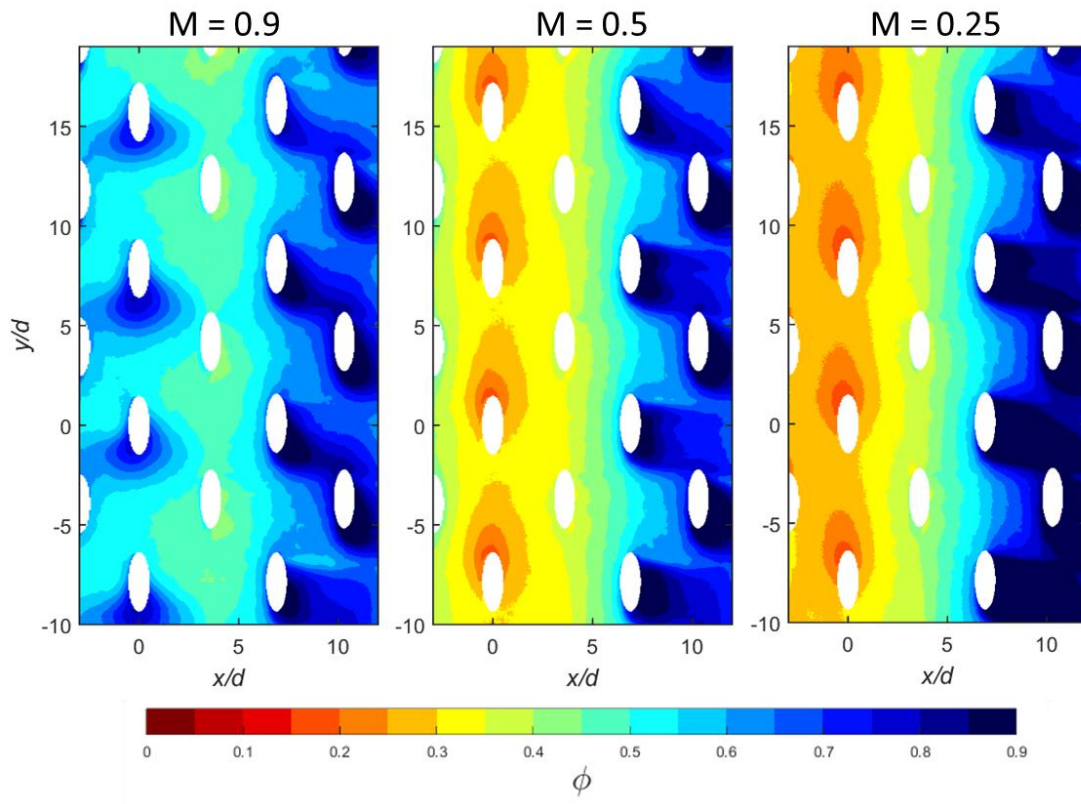


Figure 30: Overall effectiveness for blocked vertical row of holes at $x/d = 4$

$\Delta\phi$ contours for these cases are shown in Figure 31. At $M=0.9$, the large negative effectiveness contours downstream of the blocked coolant holes are where the film is effective in the baseline case. There are also areas of decreased $\Delta\phi$ above each coolant hole exit and in general in the vertical row. This is due to conduction within the Corian from the cooling of the hole walls. The film from the row at $x/d=4$ is expected to be more effective at $M=0.5$, than at $M=0.9$ [33]. The center plot shows more negative $\Delta\phi$ downstream of each coolant hole exit due to the increased η , which lead to increased ϕ . There is less effect from hole cooling in the vertical row at $x/d=4$ at $M=0.5$, than at $M=0.9$, shown by a slight increase in $\Delta\phi$ above each coolant hole. This indicates hole cooling effectiveness increases with blowing ratio.

At $M=0.25$, ingestion would normally occur in both the stagnation row and the 21.5-degree row. This would bring hot air into the plenum, heating the coolant. The right $\Delta\phi$ plot shows large areas of positive $\Delta\phi$ because there is now less hot air entering the plenum, so the coolant stays cooler and cools the surface more effectively as it travels through the coolant holes and as it is ejected as a film at the $x/d=7$ row of holes. The area directly around the blocked holes is significantly cooler because there is no hot air traveling through these holes.

The average $\Delta\phi$ was calculated for each plot by averaging the surface $\Delta\phi$ s, the hole exits were ignored. The average was -0.11 for $M=0.9$, -0.09 for $M=0.5$, and 0.003 for $M=0.25$. This value represents the average effect of the modification. A positive average is especially notable because it means blocking certain cooling holes made the surface of the material cooler.

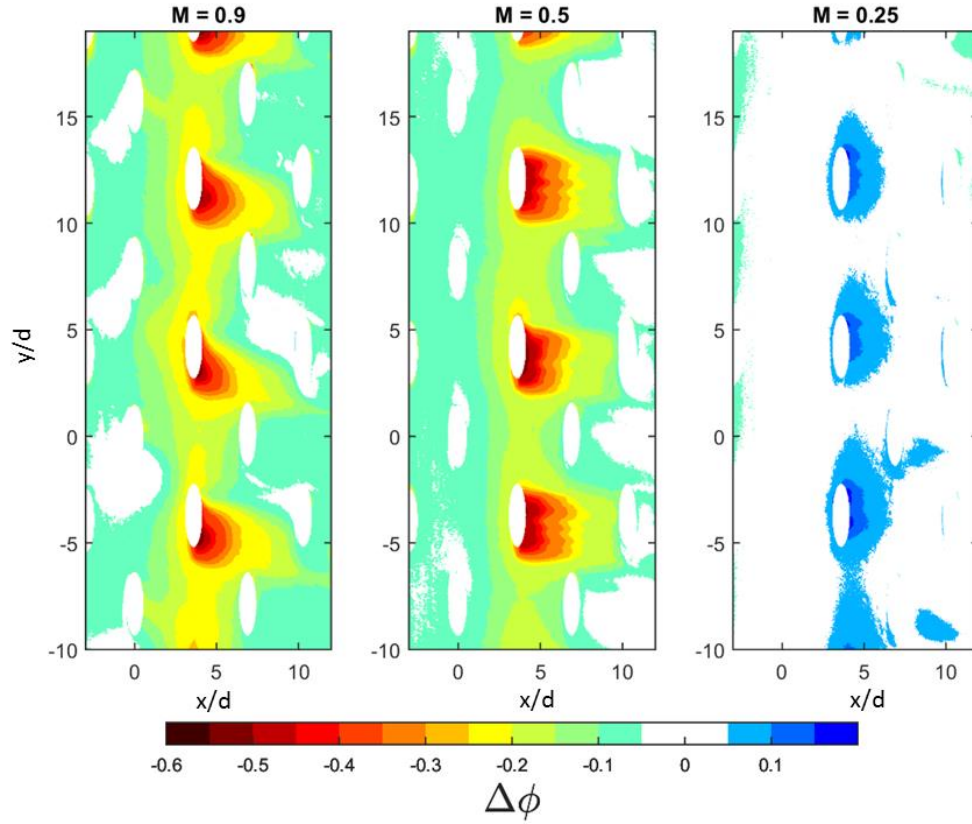


Figure 31: $\Delta\phi$ for blocked vertical row of holes at $x/d = 4$

The next modification tested was blocking the vertical row of holes on the stagnation line. The overall effectiveness plots for that case are shown in Figure 32. At $M=0.9$, there is no cooling around the blocked cooling holes. There is a small area of lower effectiveness directly over each cooling hole caused by the freestream heating the thin Corian directly before the exit. Because there is no air flowing through the hole underneath, it is neither heated nor cooled by through hole convection. But because the material is thin, it is more easily heated by the freestream compared to the full thickness Corian directly downstream each hole. The effectiveness elsewhere is unaffected because no ingestion occurred in the baseline case.

At $M=0.5$, ingestion occurred in the stagnation row for the baseline case, however with that row blocked there is no extra heating of the stagnation area. The film from the other holes are now at their best film coverage, so the entire scheme is optimized for $M=0.5$. However, the hole at $x/d=4$, $y/d=12$ is not blowing any coolant and the hole at $x/d=4$, $y/d=-4$ is only barely blowing any coolant. This is because for this exact configuration, $M=0.5$ is very close to the critical blowing ratio. Adjusting the coolant flow rate showed that slightly more coolant resulted in even blowing from the entire row and slightly less coolant resulted in ingestion for the entire row. For the baseline case, the second row's critical blowing ratio was at $M = 0.32$, but that was for 42 holes. When the stagnation row was blocked, the coolant flow rate was recalculated for 36 holes, so that the flow rate from each coolant hole would be identical. But that didn't consider the flow the stagnation row had been ingesting, which was then mixing in the plenum and going back out through the other coolant holes. So, though the coolant going into the plenum was only at $M=0.32$, when the ingested air was added, the blowing ratio out the holes on the side was higher. In the case where the stagnation row was blocked, no air could enter the plenum at $M=0.5$, so there was no extra coolant being ejected out the next row, thus why the second row is at a critical blowing ratio, much higher than the critical blowing ratio calculated when the stagnation row was not blocked.

At $M=0.25$, the second row of vertical holes begins to ingest freestream flow. Effectiveness drops above and around these holes because of the heating. The only effect on the stagnation area is slight heating due to conduction from the slightly heated material above each stagnation row cooling hole.

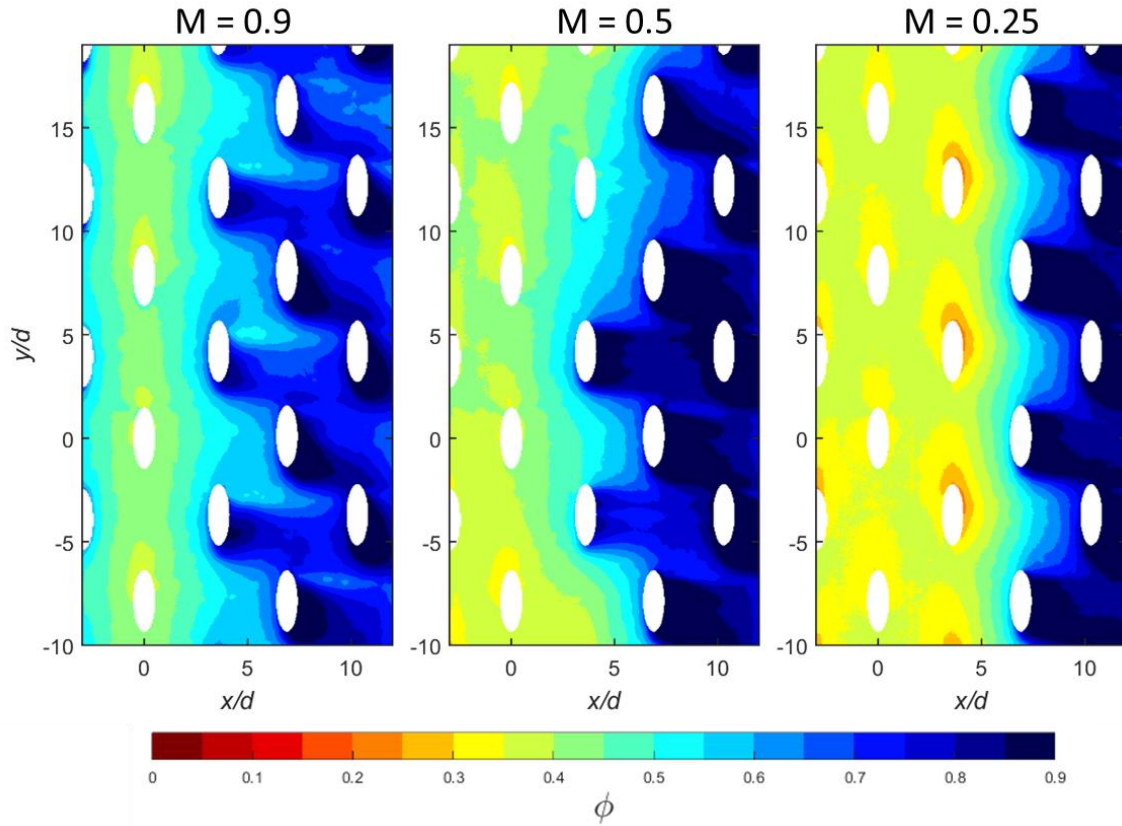


Figure 32: Overall effectiveness for blocked vertical row of holes on the stagnation line

Figure 33 shows the $\Delta\phi$ due to blocking the stagnation row film cooling holes. At $M=0.9$ the film from the blocked holes and the conduction above each hole is clearly visible. These are the areas which are cooled by film and hole cooling in the baseline case. At $M=0.5$, the results are more confusing. The area above the stagnation row holes has a positive $\Delta\phi$ because the holes are not ingesting any freestream, thereby causing hole warming. The area above the holes are therefore cooler than in the baseline case by an effect which is solely due to hole cooling. The second row is at its critical blowing ratio, as previously discussed. So, some holes are blowing only a small bit of coolant, and others

are not blowing, but also not ingesting much if any freestream. The area of decreased $\Delta\phi$ downstream of the hole at $x/d=4$, $y/d=12$ shows where the film is effective in the baseline case. Note there is little negative $\Delta\phi$ above this coolant hole because it is not ingesting flow. The hole immediately below, at $x/d=4$, $y/d=4$, has a very positive $\Delta\phi$ because this hole is now blowing more than usual (seen by comparing Figure 32 to Figure 27). The uneven flow in this row is expected at its critical blowing ratio, as shown in Figure 28. The plot on the right for $M=0.25$ shows little difference from the baseline case except in the stagnation row where there is no ingestion, meaning no hole warming in the stagnation row. Hole warming still occurs, like in the baseline case, for the row at $x/D=4$. The average $\Delta\phi$ was -0.09 for $M=0.9$, 0.01 for $M=0.5$, and 0.02 for $M=0.25$.

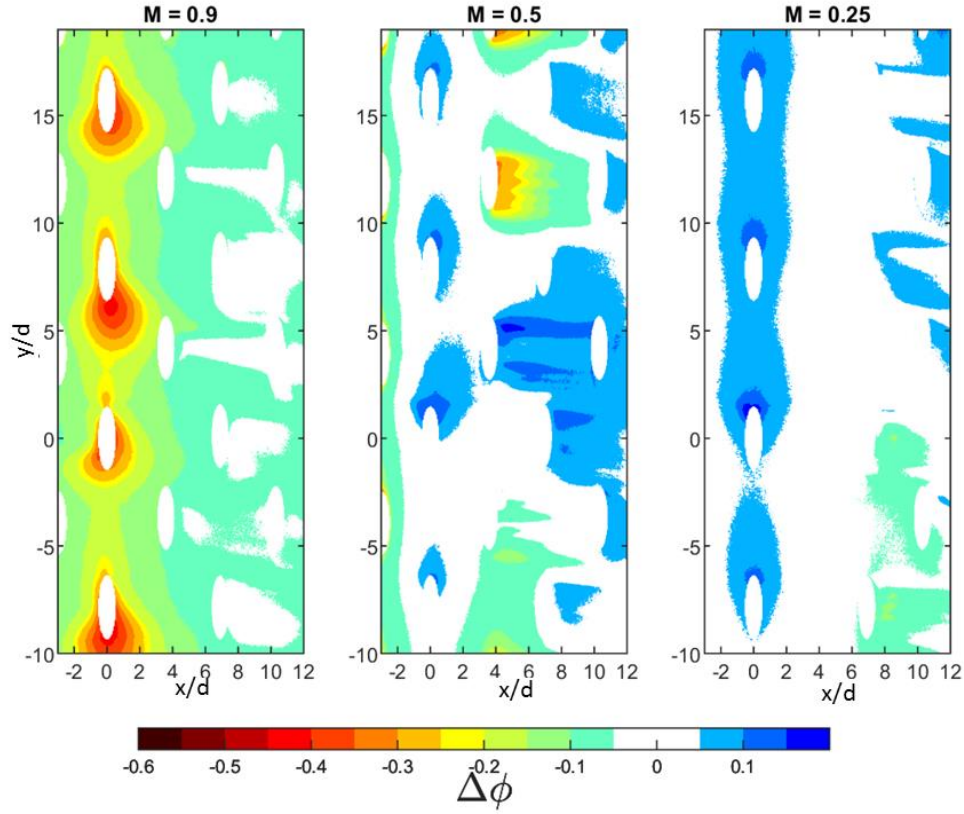


Figure 33: $\Delta\phi$ for tape over stagnation row

The next modification tested was blocking a horizontal row of cooling holes. In order to match the number of cooling holes blocked in view of the IR camera to the vertical case, the ‘row’ is actually two rows of holes at $y/d=0$ and 4. For all three blowing ratios, the unblocked holes in the overall effectiveness plots, shown in Figure 34, look like the baseline case. The blocked hole in the stagnation row is identical to the corresponding hole in Figure 32, with slightly decreased effectiveness above the hole from localized heating due to the thinness of the material above the hole, and otherwise matching the effectiveness around it. The hole in the next row, at $x/d=4$, $y/d=4$, is identical to the corresponding hole

in Figure 30, where it was blocked. The other two holes behave similarly, neither blowing or heating.

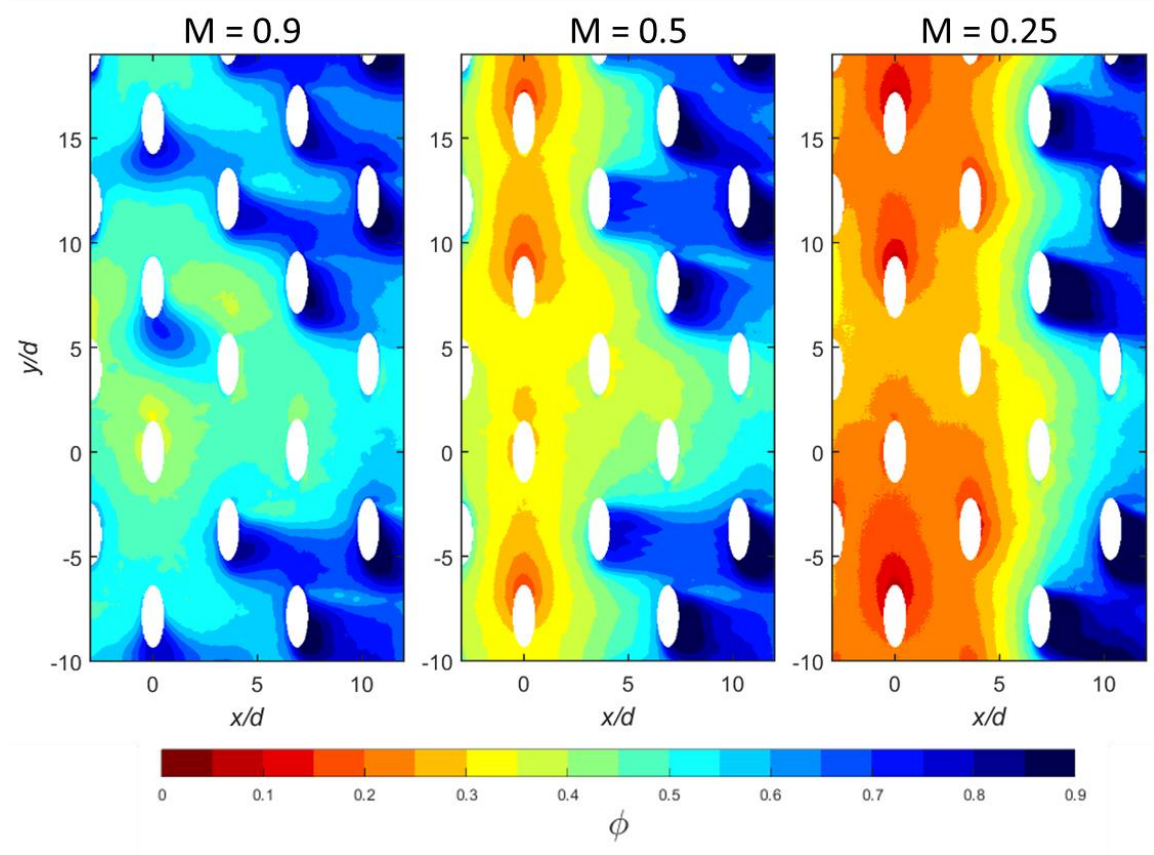


**Figure 34: Overall effectiveness contours for blocked rows of cooling holes
at $y/d=0$ and 3**

The $\Delta\phi$ plots for this case are shown in Figure 35. The plot on the left shows $\Delta\phi$ at $M=0.9$. The film cooling plumes are shown below and downstream of each cooling hole by areas of decreased $\Delta\phi$. Hole cooling is evident at the top of each of the blocked coolant hole exits. At $M=0.5$, there is no film cooling $\Delta\phi$ plume for the stagnation row, because this hole ingests flow in the baseline case. In fact, there is an area of increased $\Delta\phi$ directly

above it from no hole warming. The other three cooling holes behave as they did at $M=0.9$, but with a slightly less $\Delta\phi$ because $M=0.5$ is the optimal blowing ratio for film cooling.

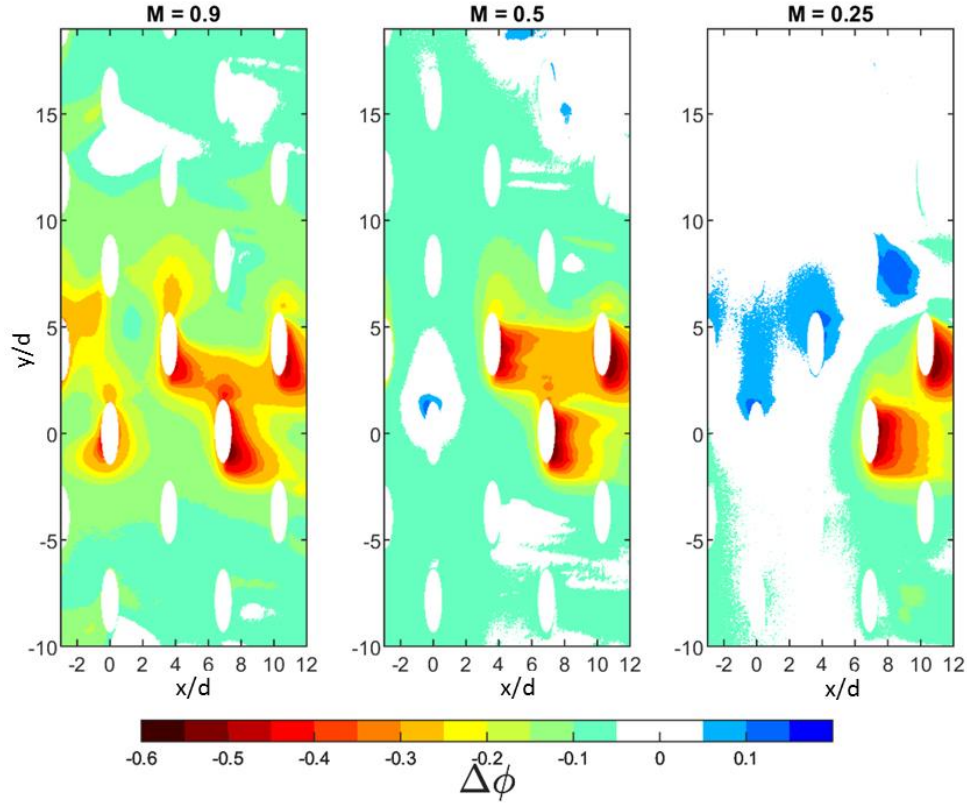


Figure 35: $\Delta\phi$ contours for blocked rows of cooling holes at $y/d=0, 3$

At $M=0.25$, both the hole at the stagnation row and the hole at $x/d=4$ are not ingesting freestream, like the baseline case. There is increased effectiveness directly above each hole because the hole walls are not being warmed by the freestream, as they would in the baseline case. The two remaining holes have the same film cooling decreased $\Delta\phi$ evident downstream of each hole.

The hole at $x/d=7, y/d=8$ has a very positive $\Delta\phi$ directly downstream of it, where its film cooling plume is effective. This is because the coolant exiting this hole is cooler than

it is the baseline case. At $M=0.25$ in the baseline case, hot freestream is ingested through the stagnation and second row of cooling holes. This hot freestream enters the plenum, mixes with the coolant, warming it, and then exits through the third and fourth row of cooling holes. But when there is no hot freestream entering the plenum from the holes directly below and upstream, like the baseline case, the coolant remains cold when it exits the cooling hole at $x/d=7, y/d=8$. Thus, the surface is cooled more than in the baseline case and the $\Delta\phi$ value increases.

Figure 36 shows the inlet and exits for this case, to better explain the colder than usual film from hole $x/d=7, y/d=8$. The black holes represent the holes that were blocked. The yellow shows the holes directly above. The solid lines around the hole denote hole exits, while the dashed lines denote hole inlets. At $M=0.25$, in the baseline case, hot freestream air would travel in the two black ovals and enter the plenum at the two black dashed ovals, from there, the hot air would mix a little before exiting through a nearby exit, the yellow dashed hole directly above. The air would then exit at the solid yellow oval and form a warm film. When the black holes are blocked, no warm air mixes in the plenum, which means the air exiting the yellow hole is cooler than in the baseline case.

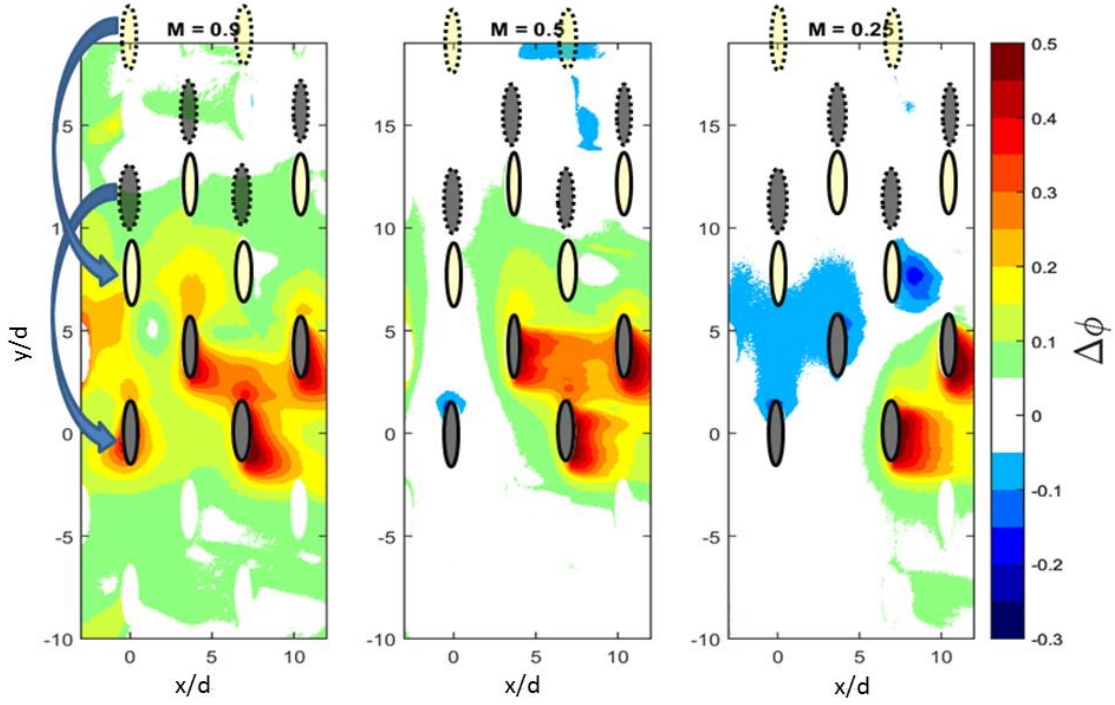


Figure 36: Hole inlet and exit locations. The black holes were blocked.

The average $\Delta\phi$ for each blocked hole case, at every blowing ratio, is plotted in Figure 37. This plot shows the effect ingestion has on the leading edge model. At $M=0.25$ the most positive delta phi is the blocked stagnation row, where the first row is prevented from ingesting freestream. The next most positive is the blocked row at $x/d=4$, where the holes in the second row are prevented from ingesting. Finally, the last case, where only 2 ingesting holes are blocked, has an overall negative delta phi. At $M=0.5$, the blocked row at $x/d=4$ drops to the most negative $\Delta\phi$ because the stagnation row is ingesting freestream, and the second row is at its critical blowing ratio. At $M=0.9$, the blocked hole only prevented the prescribed holes from blowing, no ingestion is prevented.

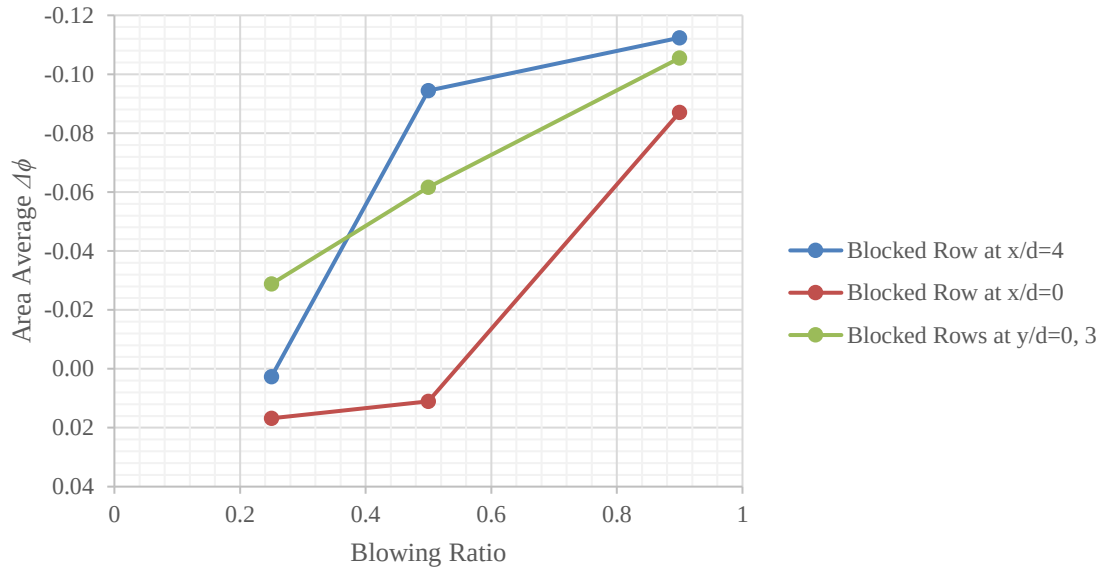


Figure 37: Average $\Delta\phi$ for the blocked holes

All three modifications show the effect of freestream ingestion. Ingestion is always expected to decrease overall effectiveness. It is never designed to occur in real gas turbines. Ingesting hot freestream is an interesting academic problem, which only occurs in the very front rows on a showerhead of cooling holes at very low blowing ratios. In a real cooling design, the designer would ensure ingestion would never occur by not placing a row of cooling holes on the stagnation line and using a blowing ratio higher than the critical blowing ratios for all the cooling holes.

4.3 Impingement

The next set of experimental tests was done using the impingement plates shown in Figure 12. No film cooling holes were blocked for any of the tests. The first plate tested was the 46-hole plate. This plate was expected to be the best plate at increasing overall effectiveness. However, due to the manufacturing flaw previously discussed, every hole

inlet but the top row was directly in front of a hole in the impingement plate. The overall effectiveness contours for this case are shown in Figure 38. These results show the same trends as the baseline case.

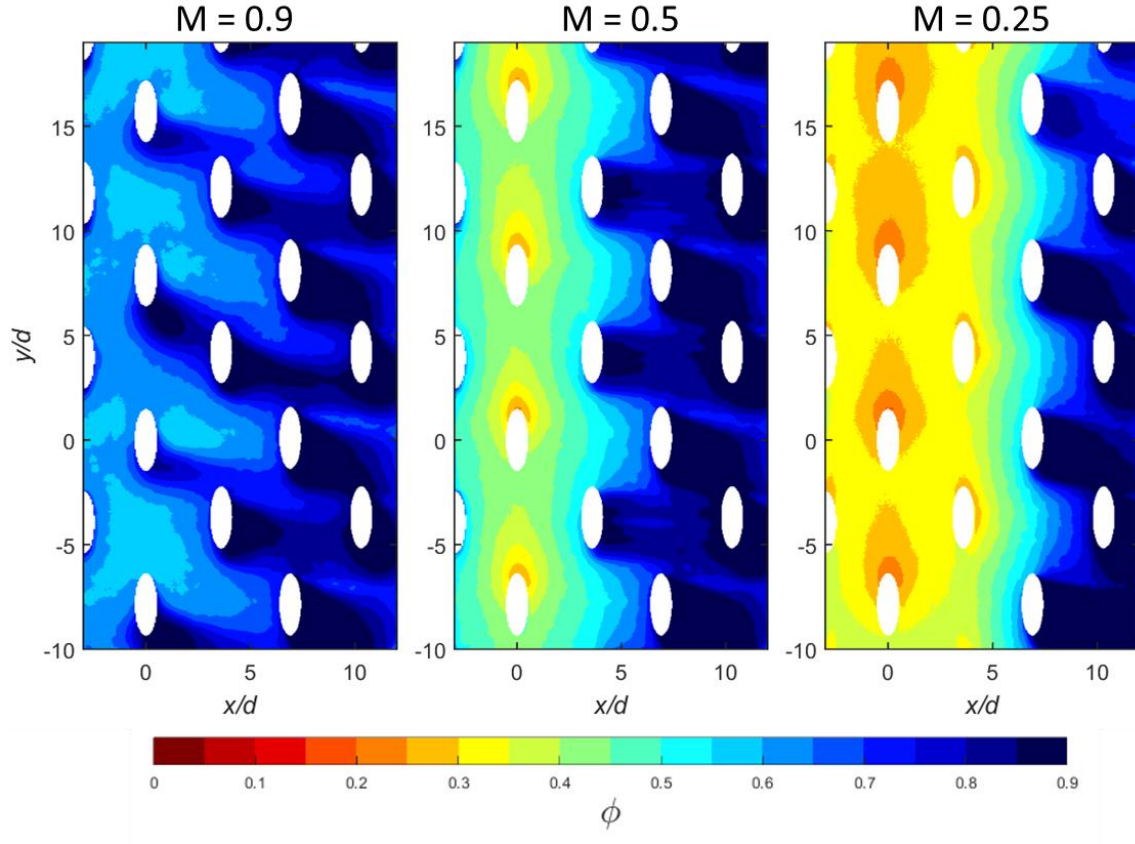


Figure 38: Overall effectiveness contours using 46-hole impingement plate

The $\Delta\phi$ contours for impingement with the 46-hole plate are shown in Figure 39. The average $\Delta\phi$ is -0.06 for $M=0.9$, -0.05 for $M=0.5$, and -0.10 for $M=0.25$. Recall the definition of $\Delta\phi$ is the modified condition, in this case impingement, minus the baseline case. We expect impingement to create a cooler outer surface temperature than the baseline case (without impingement), so the ϕ for impingement should be closer to one. Therefore, $\Delta\phi$ should, on average, be positive if impingement is aiding in cooling the surface.

However, per the mostly negative $\Delta\phi$ contours in Figure 39 and the negative $\Delta\phi$ averages, impingement does not help cool the outer surface in this case. Impingement is expected to increase the internal heat transfer coefficient and therefore decrease the inside wall temperature. There are a couple possible explanations for why this doesn't appear to be occurring. First, because there are so many cooling holes packed tightly, overlapping each other, any change on the internal surface may not be able to conduct past the cooling holes, which are also transferring heat from the coolant flowing through them. Second, the impingement holes exactly align with the film cooling hole inlets, So instead of impinging jets hitting the internal surface and changing the internal heat transfer coefficient, the jets of coolant directly entered the cooling holes without interacting with the internal surface at all.

At $M=0.9$, the film plume areas for each cooling hole is close to zero, or slightly positive $\Delta\phi$. The reason the film has a slightly positive effect may be because the coolant is being injected straight into the coolant holes, rather than circulating in the plenum and warming before entering the coolant holes as it would in the baseline case. This may also explain why the surface is hotter than the baseline case between the film areas. The coolant in the plenum can circulate, cooling the inner surface in the baseline case. The 46-hole plate causes the coolant to flow directly into the cooling holes, rather than flow over the inner surface cooling it.

At $M=0.5$, there are fewer areas of positive $\Delta\phi$. And at $M=0.25$, $\Delta\phi$ is nearly entirely negative. The trend, therefore seems to be increasing the blowing ratio increases $\Delta\phi$. Yet

for all three blowing ratios, a lack of impingement is harmful to overall effectiveness. Ingestion appears to have no effect.

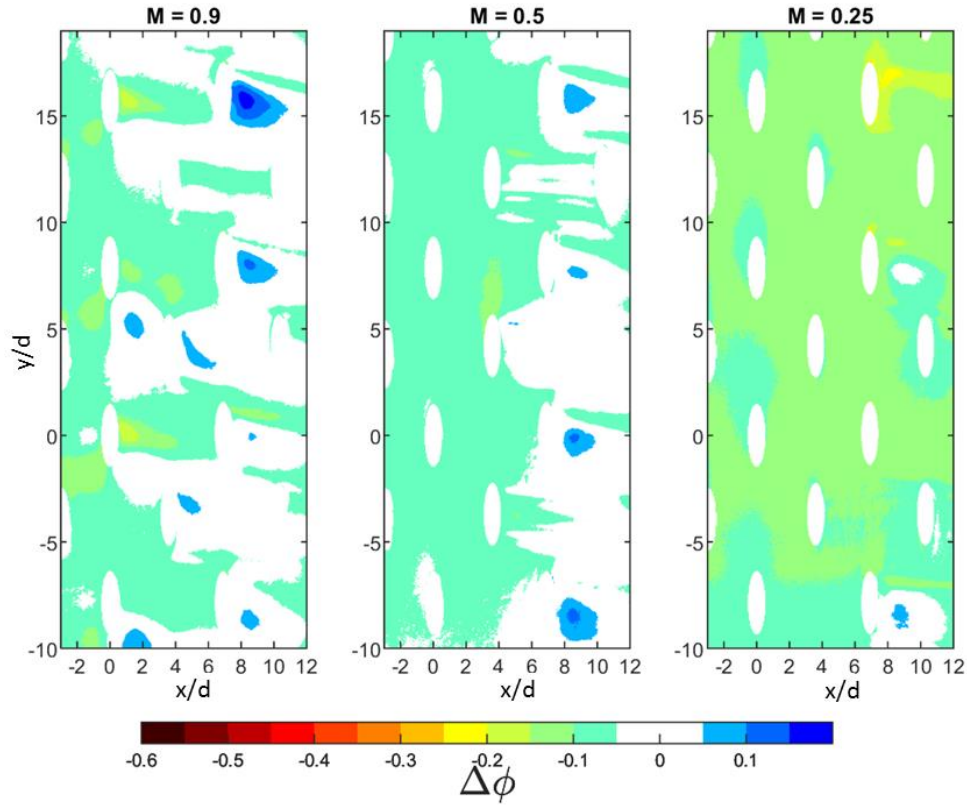


Figure 39: $\Delta\phi$ contours using 46-hole impingement plate

The next case tested used the 20-hole impingement plate, which was expected to be less effective at cooling than the 46-hole plate. The overlap between the hole inlets and impingement plate holes lined up only on the stagnation row, so it is expected there will be extra effective film for this row at $M=0.9$ only. The overall effectiveness contours for this case are shown in Figure 40. This plot again looks nearly identical to the baseline case.

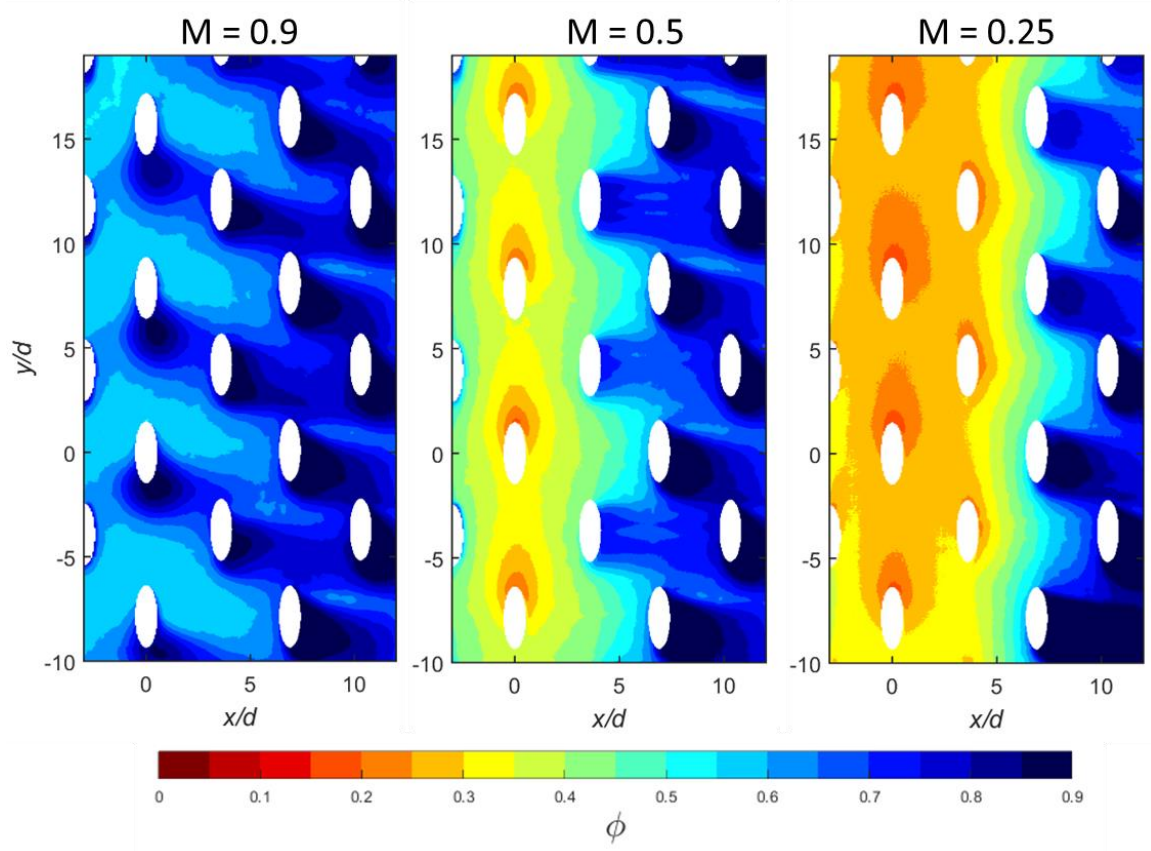


Figure 40: Overall effectiveness contours using 20-hole impingement plate

The $\Delta\phi$ plots for this case are shown in Figure 41. The average $\Delta\phi$ is -0.03 for $M=0.9$, -0.08 for $M=0.5$, and -0.11 for $M=0.25$. As expected, there are areas of increased $\Delta\phi$ in the film area for each hole in the stagnation row at $M=0.9$. There are also large areas of zero $\Delta\phi$ and a couple film areas with positive $\Delta\phi$ indicating cooling benefit from this impingement scheme. However, the average again is negative for all three blowing ratios. Compared to the 46-hole plate, the 20 plate was more effective at $M=0.9$, much worse at $M=0.5$, and about equal at $M=0.25$.

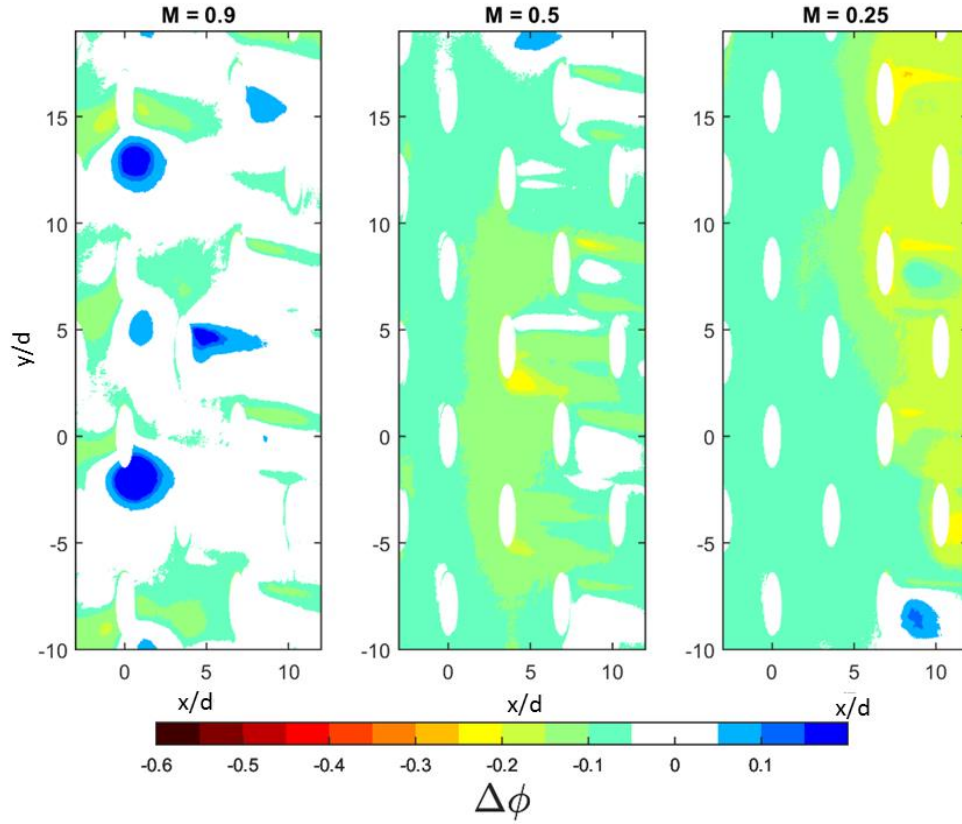


Figure 41: $\Delta\phi$ contours using 20-hole impingement plate

The third impingement plate tested was the two-hole plate. This plate was expected to provide the least effective impingement scheme. The impingement holes lined up with the holes in the model at $x/d=0$, $y/d=8$ and $x/d=0$, $y/d=-16$. The second hole is not in the IR camera's view. The overall effectiveness contours for this case are shown in Figure 42. At $M=0.9$, the hole at $x/d=0$, $y/d=8$ has increased effectiveness above the hole exit, where the extra coolant flowing through the hole has increased hole cooling and only very small film cooling due to jet liftoff. The rest of the holes in the stagnation row and a hole in the next row have less of a film than the baseline.

At $M=0.5$, the stagnation row is expected to be ingesting freestream, however the hole at $x/d=0, y/d=8$ has a large coolant plume and is contributing to both film and hole cooling. The other three holes in the stagnation row are ingesting freestream like the baseline. Another anomaly is the hole at $x/d=4, y/d=12$ is not blowing, nor was it ingesting. However, the hole directly below is blowing and has a significant coolant plume thanks to the overlap with the hole at $x/d=0, y/d=8$. Finally, at $M=0.25$, both the first and second rows were expected to be ingesting freestream. However, the hole in line with the impingement plate is still barely blowing and creating a very slightly effective film.

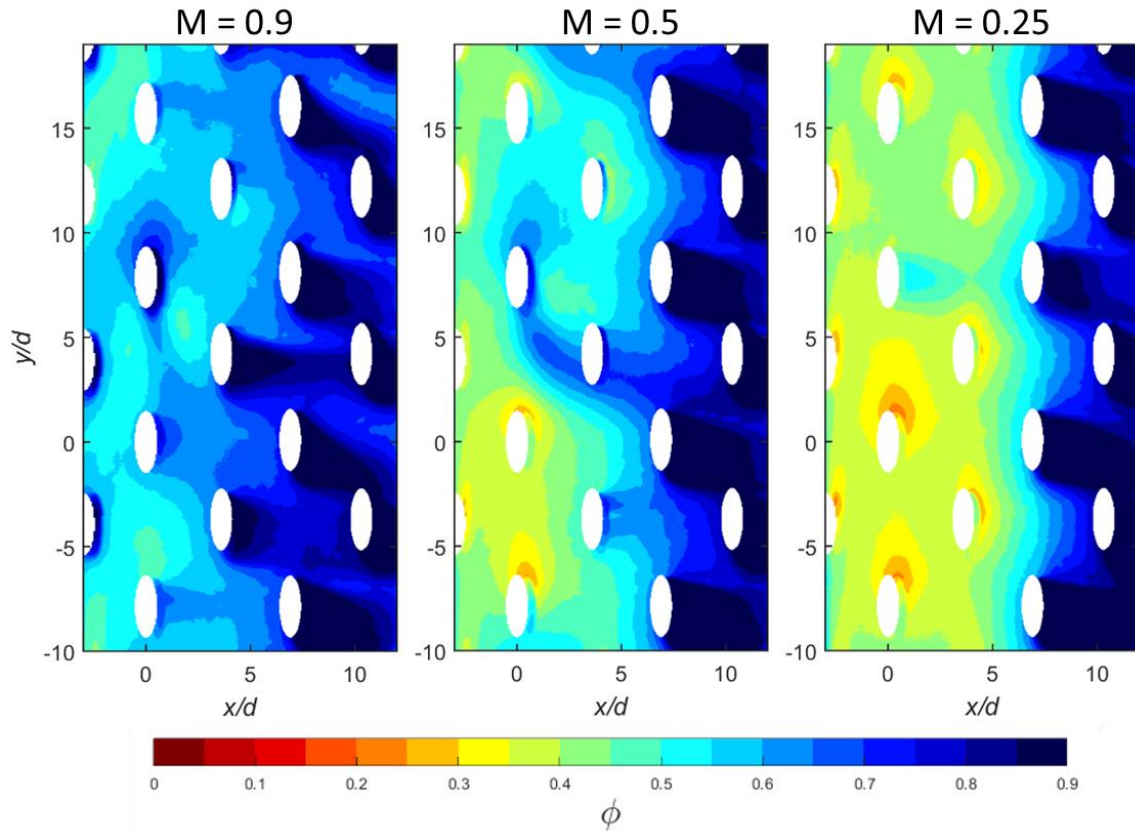


Figure 42: Overall effectiveness contours using the two-hole impingement plate

$\Delta\phi$ contours for the two-hole plate are shown in Figure 43. The average $\Delta\phi$ is -0.09 for $M=0.9$, -0.06 for $M=0.5$, and -0.04 for $M=0.25$. At $M=0.9$, the cooling hole at $x/d=0$, $y/d=8$ has reduced $\Delta\phi$ due to cooling jet lift off. The extra hole cooling due to the increased mass flow results in positive $\Delta\phi$ directly above the cooling hole. The other three holes in the stagnation row has negative $\Delta\phi$ because they are not blowing. The same is true of the hole at $x/d=4$, $y/d=12$ since it was not blowing. The other holes have a zero or slightly positive $\Delta\phi$ immediately downstream.

At $M=0.5$, the hole at $x/d=0$, $y/d=8$ has a positive $\Delta\phi$ in its coolant plume. It also has a significant positive $\Delta\phi$ directly above due to hole cooling, vis-à-vis the hole warming it experiences in the baseline case at this blowing ratio. The holes in the next row, at $x/d=4$, all have negative $\Delta\phi$ downstream due to less film cooling. Even the hole at $x/d=5$, $y/d=4$, which was covered by the film from the impinged cooling hole has negative $\Delta\phi$, though it appeared to be blowing coolant in the overall effectiveness contour. At $M=0.25$, though the hole in line with the impingement plate, at $x/d=0$, $y/d=8$, is still blowing. It only has a slightly increased $\Delta\phi$ above and downstream of it due to film and hole cooling, rather than the baseline ingestion. The rest of the model is largely warmed due to the impingement plate.

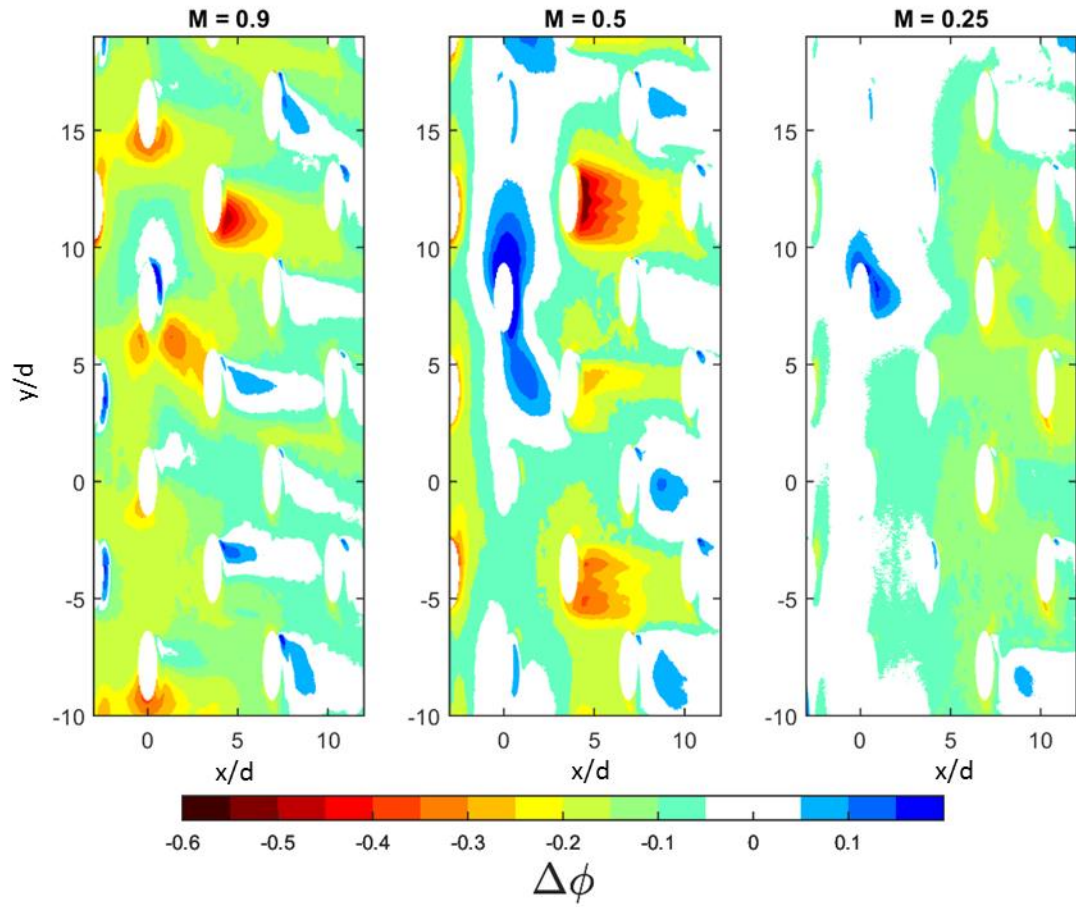


Figure 43: $\Delta\phi$ contours using two-hole impingement plate

The final impingement plate tested was the slots plate. The slots in this plate lined up the vertical row of holes at $x/d=4$ and 10 , so increased film and hole cooling is expected from these holes. The overall effectiveness contours for this case are shown in Figure 44. This case has very similar overall effectiveness contours to the baseline case.

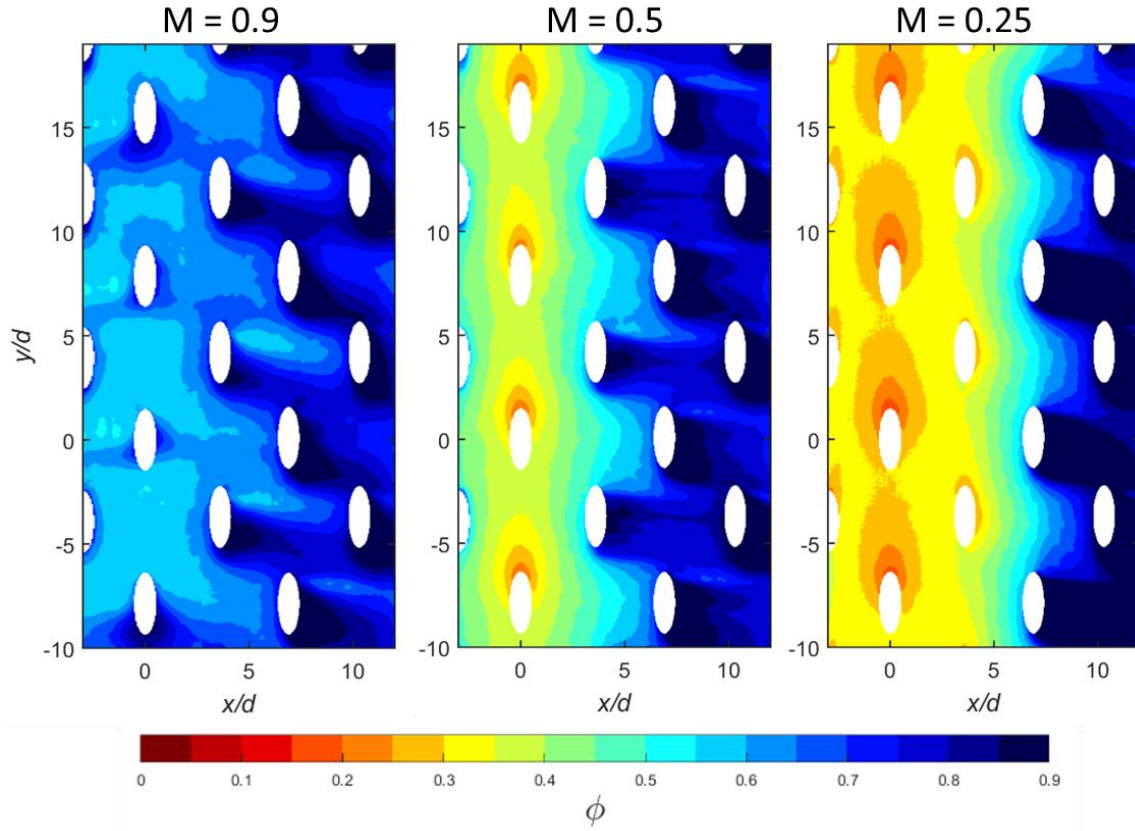


Figure 44: Overall effectiveness contours using slots impingement plate

The $\Delta\phi$ contours are shown in Figure 45. The average is -0.03 for both $M=0.9$ and $M=0.5$ and -0.06 for $M=0.25$. These are the smallest $\Delta\phi$ averages of any of the impingement plates, though they are still negative. There are several localized spots of both increased and decreased effectiveness. All of them are in the film areas downstream and below a few cooling holes, therefore the localized variations are likely due to differences in coolant flow in the plenum due to the slots plate.

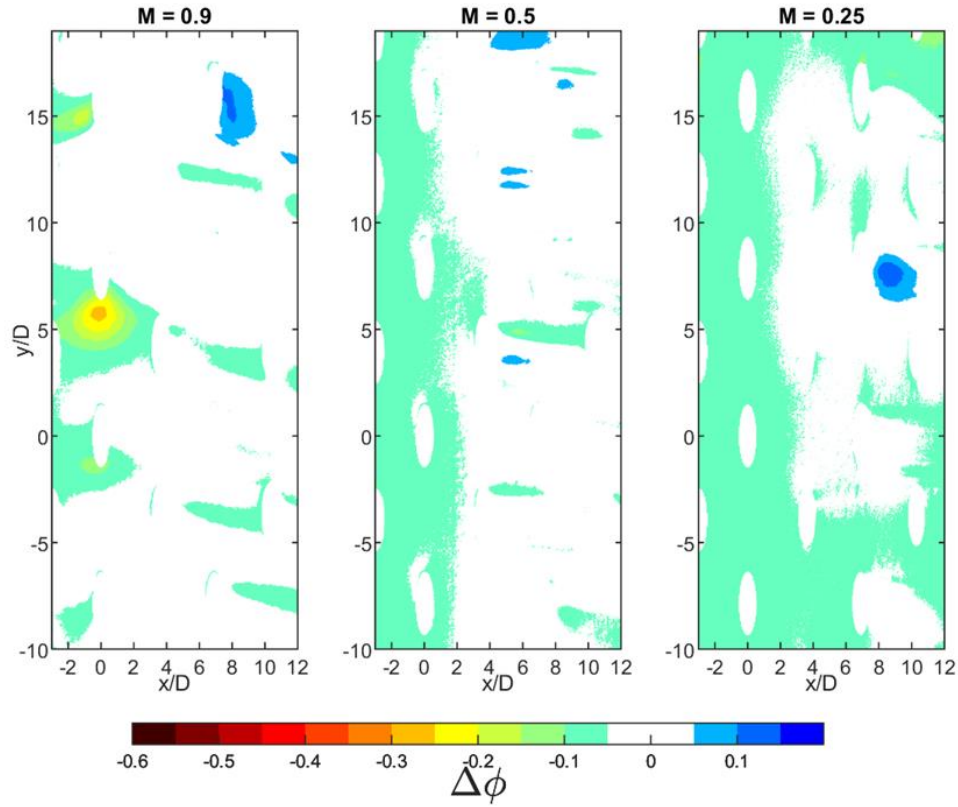


Figure 45: $\Delta\phi$ contours using slots impingement plate

Figure 46 shows the trends in average $\Delta\phi$ for the four impingement plates. For three of the four plates, increasing the blowing ratio increases $\Delta\phi$. However, none of the plates had an overall positive $\Delta\phi$. Therefore, the net effect of impingement for the plates tested was to decrease the overall cooling effectiveness.. Impingement is only beneficial at the far outside, away from the cooling holes.

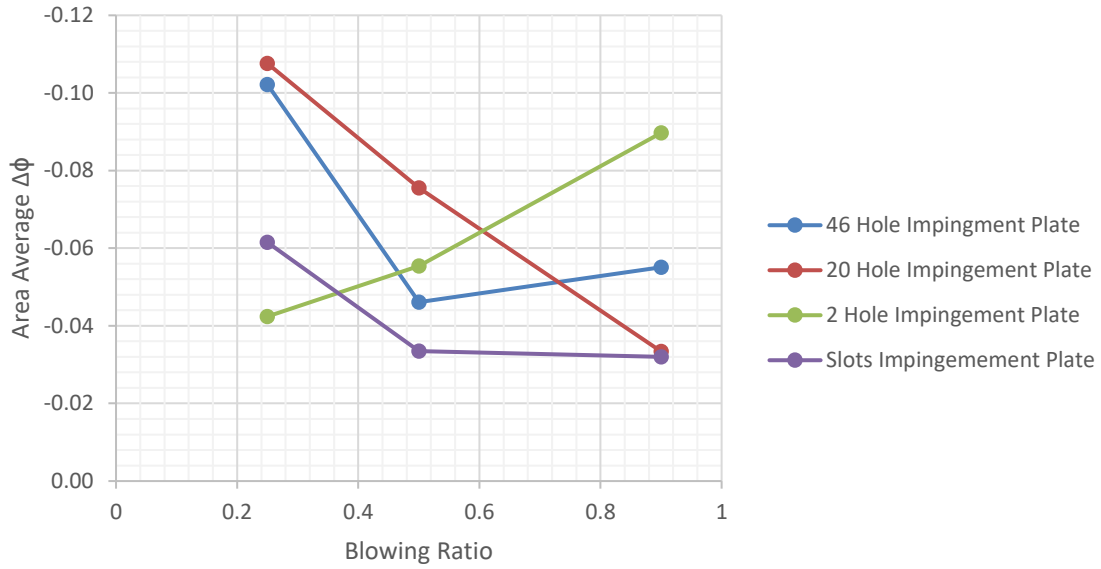


Figure 46: Average $\Delta\phi$ for the impingement plates

This is contradictory to previous literature. In most cases, impingement is helpful. This is always true in previous literature away from cooling holes. One previous paper by Ravelli et al. [22] found impingement was not helpful in the direct vicinity of the coolant holes. The contours of overall effectiveness from that paper are shown in Figure 47. Plot (a) shows the leading edge without impingement at $M=0.1$. Plot (b) shows the same blowing ratio without impingement. Film effectiveness has decreased downstream of each coolant hole, but the centerline effectiveness has increased. Plot (c) shows no impingement at $M=2$ and Plot (d) shows the same case with impingement. At this blowing ratio, centerline and spanwise effectiveness is decreased due to impingement.

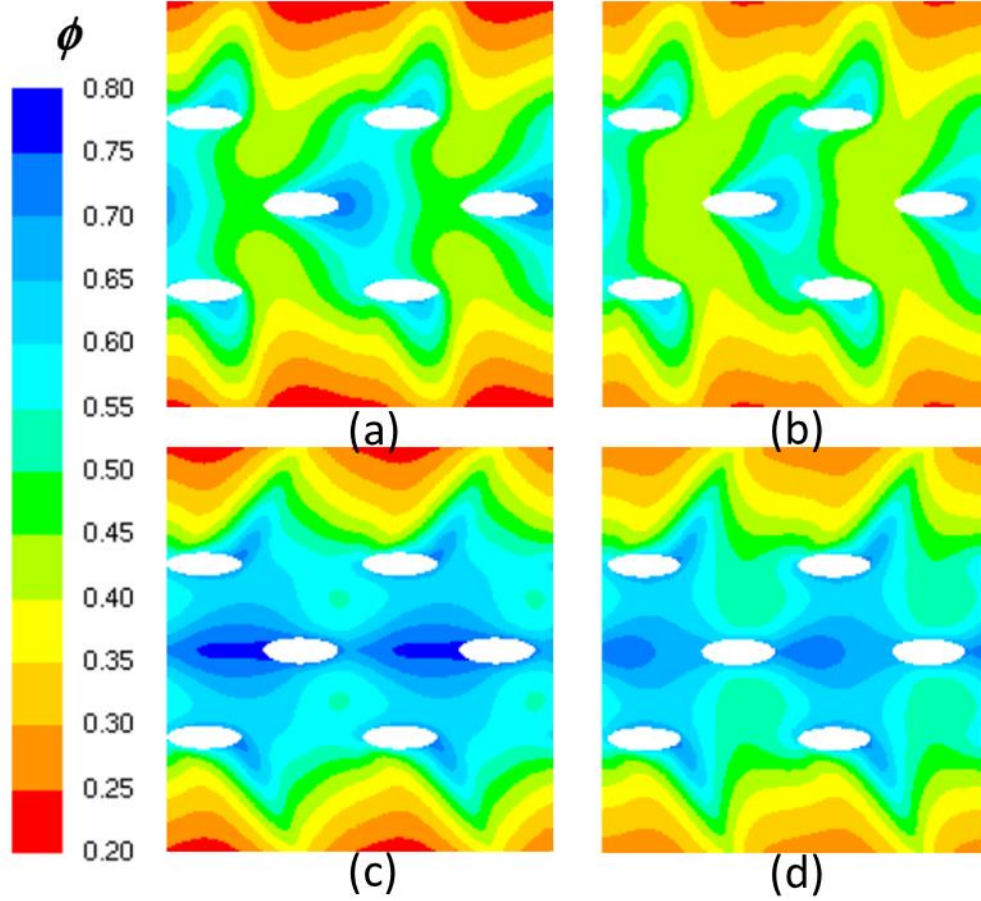


Figure 47: Contours of overall effectiveness from [22]

4.4 Coolant Warming Factor

The coolant warming factor, χ , is defined as

$$\chi = \frac{T_{\infty} - T_{ce}}{T_{\infty} - T_{cp}} \quad (5.1)$$

Where T_{cp} is the coolant temperature where the coolant enters the plenum and T_{ce} is the coolant temperature where it exits the cooling hole. χ is the amount the coolant warms up between when it enters the plenum and exits the cooling hole. To better understand where the coolant is warmed, the coolant warming factor was split into two components. χ_p is the

coolant plenum warming factor which describes the temperature difference between the coolant entering the plenum and the inlet to the coolant holes. χ_h is the coolant hole warming factor which describes the temperature difference between the coolant hole inlet and exit. These two parameters are defined as

$$\chi_p = \frac{T_\infty - T_{ci}}{T_\infty - T_{cp}} \quad (5.2)$$

$$\chi_h = \frac{T_\infty - T_{ce}}{T_\infty - T_{ci}} \quad (5.3)$$

An additional property of these parameters is they can be multiplied together to equal the coolant warming factor for the whole model, χ .

$$\chi = \chi_h \chi_p \quad (5.3)$$

The closer χ is to unity, the less warming occurs in the plenum and holes. If the hole or plenum warming factor is close to zero than the other, then that is where the warming is primarily occurring.

The coolant warming factors were computed using thermocouple data. The thermocouple locations are shown in Figure 15 and Figure 16. The thermocouples for measuring coolant temperature were placed on the third vertical row of holes from the stagnation line, so no ingestion ever occurred through the instrumented holes. Figure 48 shows the average of the coolant warming factors for the three baseline cases taken for repeatability each test day.

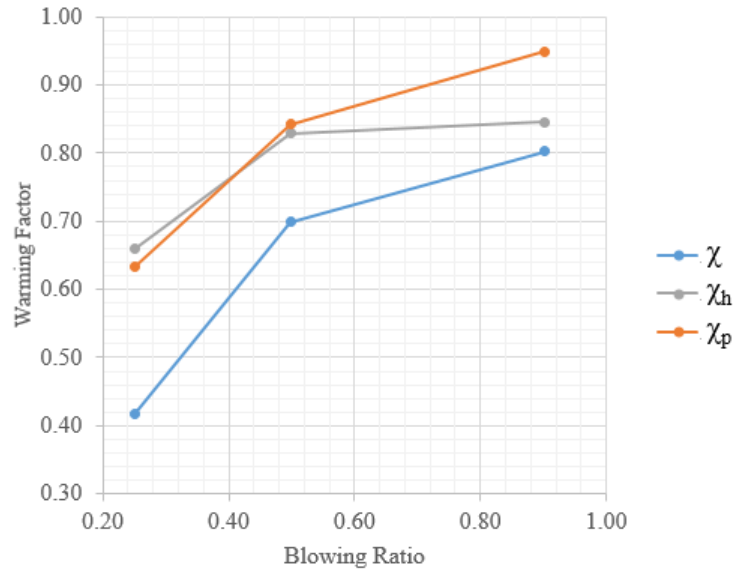


Figure 48: Average coolant warming factors for the repeatability cases

At $M=0.9$, where no ingestion occurred, most of the coolant warming occurred in the film holes. The hole warming factor was 0.85, whereas the plenum warming factor was 0.95. At $M=0.5$, freestream was ingested in the stagnation row of holes, which likely caused increased coolant warming in the plenum. The warming factors were nearly equal, 0.83 for warming in the holes and 0.84 for warming in the plenum. Finally, at $M=0.25$, freestream was ingested in the front three coolant holes, resulting in significant coolant warming in the plenum. The coolant is slightly heated more in the plenum than in the holes, evidenced by 0.63 plenum warming factor and a 0.66 hole warming factor. Ingestion is not expected to normally occur in real engine designs. Therefore, the highest blowing ratio is the only representative case of typical conditions. The blowing ratio without ingestion shows most of the warming occurs in the holes, rather than the plenum.

The first research objective was to separate the three cooling forms. Blocking cooling holes removed the effect of hole and film cooling from those certain holes. By examining the $\Delta\phi$ plots, it was possible to assess the combined cooling from the two factors. The second research objective was to investigate the impact of different impingement plates. All four plates tested resulted in negative $\Delta\phi$, but the slots plate resulted in the least negative $\Delta\phi$. The third research objective was to find the areas where each cooling form was most effective. It was found film cooling was most effective downstream of each cooling hole exit and hole cooling was most effective directly above the hole itself, due to conduction through the material.

Blocking coolant holes did not satisfactorily answer the first objective because it didn't separate the effects of film and hole cooling. To separate the cooling forms, tubes were inserted into the exit of the film cooling holes, shown in Figure 49. The idea was the coolant would still flow into the plenum and through the coolant holes, but would flow into the tubes at the exit, instead of forming a film over the outside surface. This would show the effect of hole cooling without the effect of film cooling.



Figure 49: Experimental model with tubes installed to remove film cooling

Unfortunately, this method failed because the tubes disrupted the freestream flow over the entire model and completely changed the boundary layer and therefore the formation of the film on the rest of the model. Because the problem could not be addressed physically, computational fluid dynamics was used to answer the remaining research objectives.

V. Computational Methodology

A computational analysis was conducted to separate the effects of cooling within film holes from external film cooling. CFD allows for nonphysical boundary conditions which make the separation possible. This chapter describes the computational set up and methodology.

5.1 Computational Domain

The computational domain used was a simple flat plate model with both internal and external cooling. The model was a conjugate heat transfer model, so both the fluid and solid domains were meshed and solved. The geometry was chosen so that it could easily be experimentally verified by a similar flat plate experiment conducted by Natsui et al. [35], as well as more easily computationally solved by limited computational resources. The model consists of a flat impingement plate below a solid with a staggered array of circular holes and a matched Biot material with five film cooling holes at 20 degrees inclined to the flat surface. The model is intentionally simple to test novel techniques of separating the effects of different parts of cooling, which directly answers the first research objective.

Figure 50 shows a schematic of the computational set up, the upper right is a cross section of the side view with the solid material shown in blue and the film cooling holes shown in white. The bottom right shows a top view with the impingement condition underneath where the air inlets are shown in red. Finally, the left side shows a cross section of the width of the domain.

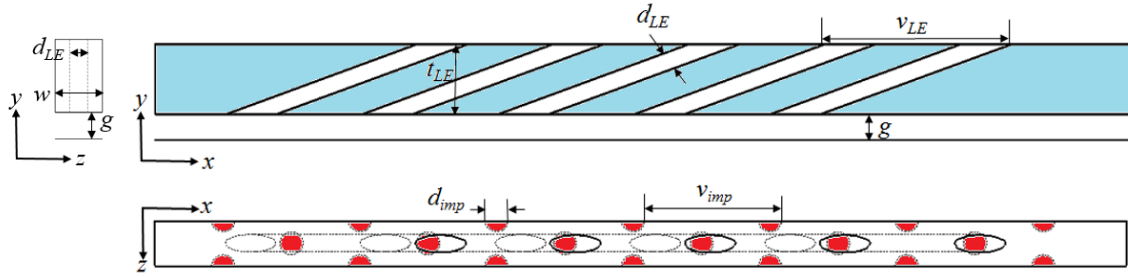


Figure 50: Diagram of CFD model

Unlike the experimental model, the flow direction was in the x directions, rather than the y direction as in the experimental tests. The width of the model was 1.27 cm with symmetrical boundary conditions along the long sides for the height of the model. The diameter of the film cooling holes, d_{LE} , and impingement holes, d_{imp} , were 0.48 cm and 0.33 cm, respectively, which is identical to the experimental model. The gap, g , between the impingement condition and the conducting material was 0.71 cm. The conducting material was modelled as DuPont Corian® with the same prescribed physical properties as previously discussed. The thickness of the material was 1.93 cm, identical to the physical model. The model dimensions are shown in Table 3.

Table 3: CFD model dimensions

Leading Edge				Impingement			
Corian height	t	0.76	in	hole spacing	v	1.5	in
Corian width	w	0.5	in	hole diameter	d	0.13	in
Corian length	l	10.66	in	gap	g	0.28	in
hole diameter	d	0.19	in				
hole spacing	v	1.48	in				
hole angle	γ	20	deg				

The model included an unheated no-slip wall upstream of the conducting material. The distance between the start of the grid and the exit of the first film cooling hole is $67.5d$.

This length allowed the boundary layer to form before the film from the coolant holes was injected. The top of the computational domain was $21d$ above the top edge of the conducting material and was prescribed as a slip wall. The end of the domain was a pressure outlet at $7d$ past the exit of the fifth cooling hole. The ends of the conducting material and impingement zone were set to be walls. The sides of the entire domain were set as symmetry conditions. The impingement holes were set as uniform velocity inlets with a mass flow rate prescribed for each test.

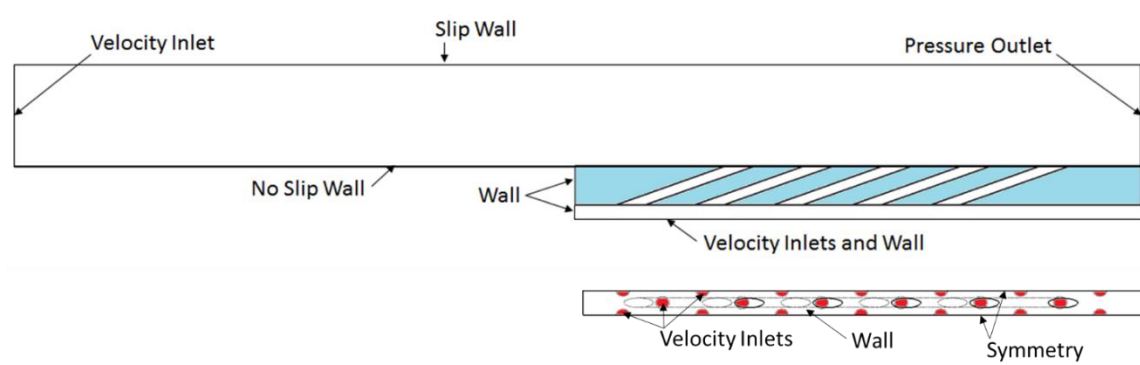


Figure 51: Boundary conditions used on the CFD model

5.2 Grid Generation

The grid used was generated using Pointwise[®] grid generation software. The entire model was meshed using unstructured cells. Eight distinct zones were created, seven for air and one for the conducting material. Each cooling hole was a zone, the freestream block on top was a zone, the impingement area beneath the conducting material was a zone, and the conducting material was a separate zone. The nodes between the zones were kept common to aid in solution convergence. Views of the grid used is shown in Figure 52 and Figure 53. The boundary and zone conditions were prescribed within Pointwise.

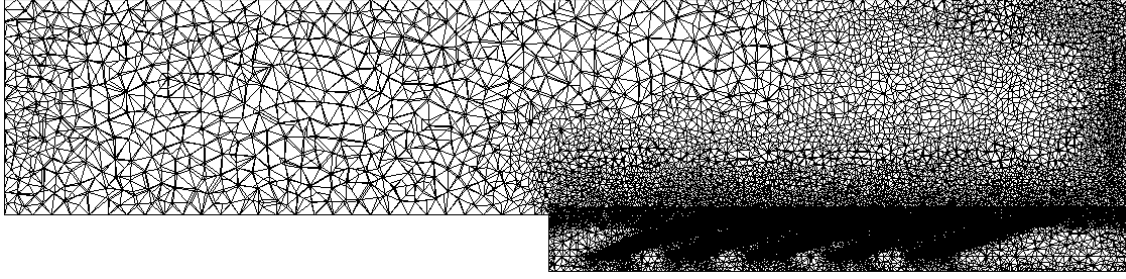


Figure 52: Mesh used for all cases

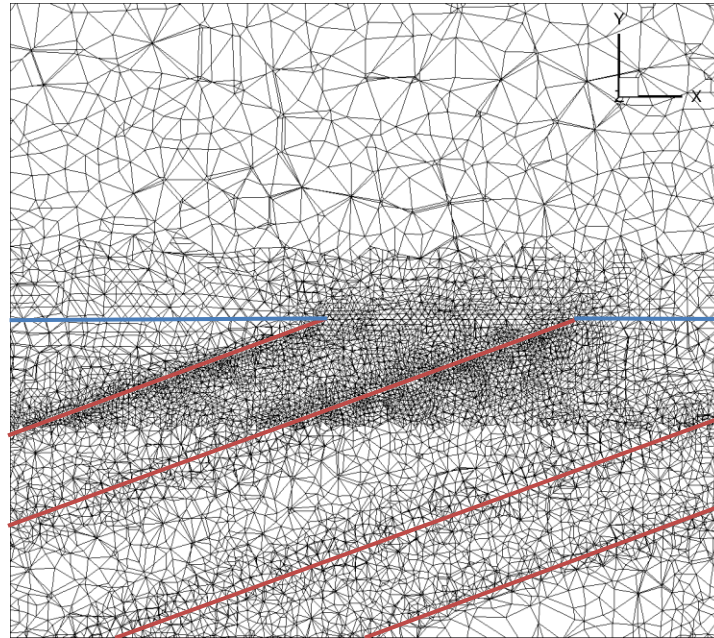


Figure 53: Close view of grid around a film cooling hole exit

Grid convergence was achieved by a grid sensitivity study, where the solution was found for a set of successively refined grids until the resulting solutions stopped changing. The aim of the study was to find a grid independent solution with the least number of cells to reach a solution in a timely manner.

Three grids were used for the study (see Table 4). They were refined using Fluent's built in grid refinement ability. All the grids were run at a Reynolds number based on cooling hole diameter of 3,400 and a coolant blowing ratio of 0.5. The coarsest grid was

generated based on calculations for correct wall boundary spacing and common CFD practices. The medium grid was created by using a region adaptation on the converged result from the coarse grid. The region specified for adaption was 0.2 inches above and below the interface between the conducting surface and freestream flow. The fine grid was made by taking the converged medium grid and applying a gradient adaption based on the velocity gradient. The criteria for adaption was any cells where the velocity gradient was within 10% of the maximum gradient throughout the entire mesh. For both adaptions, after the new cells had been created, the entire mesh was reordered using 4 Laplacian smoothing iterations.

Table 4: Grids used for grid refinement study

Name	Cells	Refinement Technique
Coarse	512,302	N/A
Medium	1,740,113	Region Adaptation
Fine	1,761,274	Velocity gradient adaptation

After each mesh had a converged solution, the surface temperature on the top of the conducting material along the center line was extracted and nondimensionalized to overall effectiveness, as in Equation 2.3. The data was plotted (Figure 54) and comparison showed little change between the medium and fine meshes. The medium mesh was chosen because it gave nearly identical result as the fine mesh and it converged in 6 hours, rather than 10 in the case of the fine mesh. This allowed more simulations to be run concurrently and more quickly. Note, in Figure 59, the red dashed lines indicate where the cooling hole exits are.

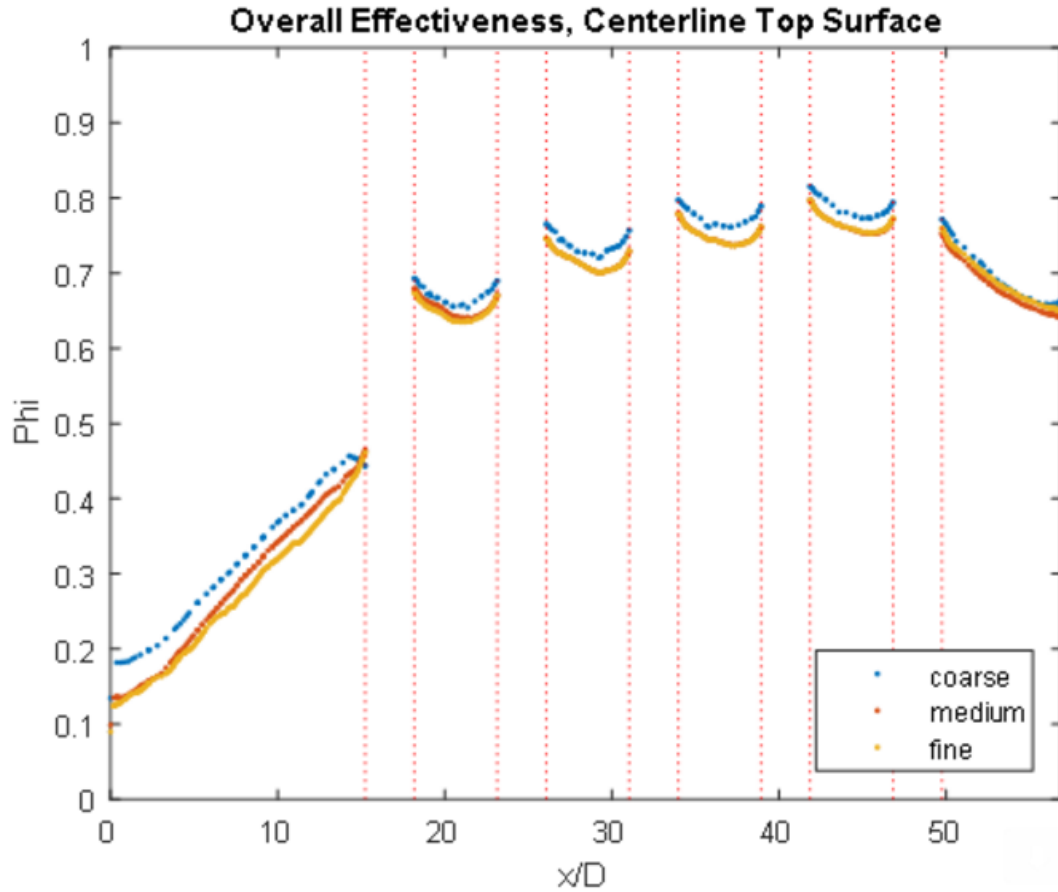


Figure 54: Overall effectiveness along the centerline for all the grids

An additional parameter monitored for each case was y^+ , the cell spacing closest to the wall. To ensure the smallest vortices were captured, y^+ along the entire length of the wall had to be less than 5. Figure 55 shows the values along the centerline of the top of the conducting surface in the medium mesh. At all points y^+ is less than five, so all the vortices should be captured. Again, note the vertical dashed red lines show where the hole exits are.

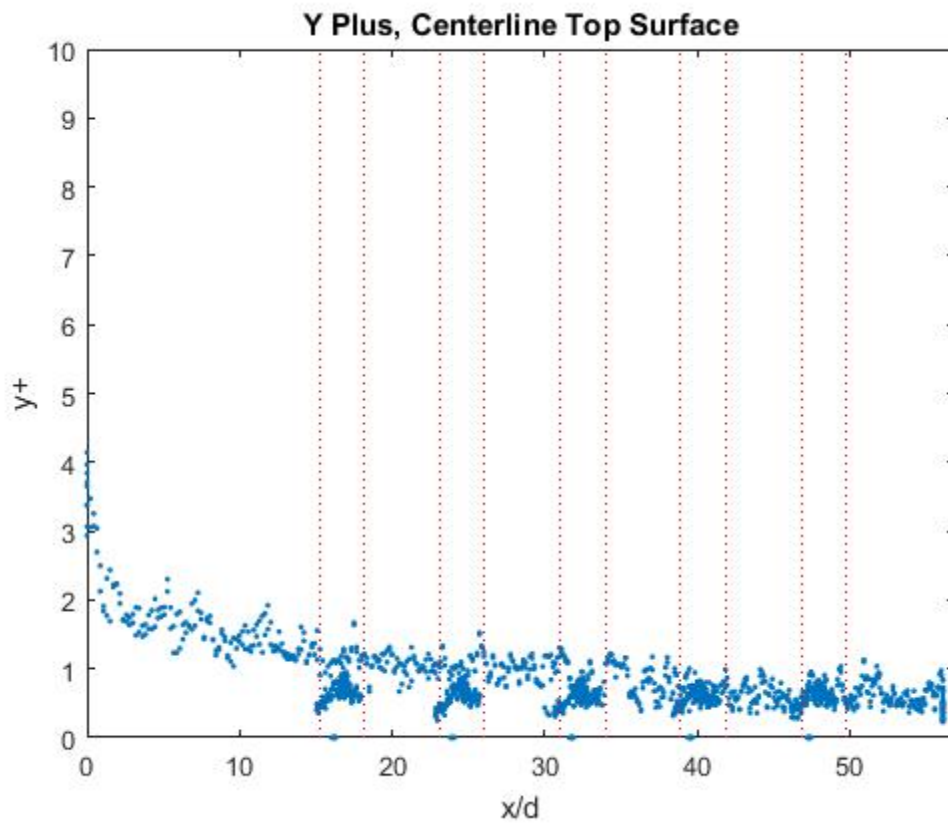


Figure 55: y^+ for the length of the top of the conducting surface

5.3 Solver

Simulations were performed using ANSYS Fluent[®] computational fluid dynamics solver. This software package can model both fluid flow and heat transfer through solid materials. The turbulence model used was the realizable k - ϵ model with enhanced wall treatment.

Most previous studies on turbine cooling using CFD use either the k - ω SST or the realizable k - ϵ turbulence model. Figure 56 shows a comparison of turbulence models for an identical cross section along the top of the conducting surface with overall effectiveness contours. The coolant and freestream velocities and inlet temperatures are identical for both cases. However, there is a large difference in the effectiveness distributions between the turbulence models. The k - ϵ model shows more cooling in the material between the coolant holes. It also shows more film effectiveness downstream of each hole.

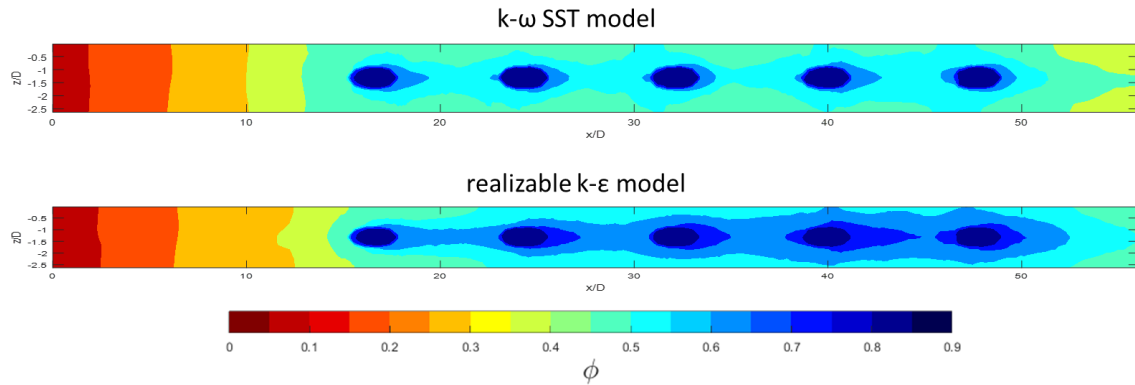


Figure 56: Comparison of turbulence models

To determine which model was more appropriate, it was decided to run an adiabatic effectiveness study and compare to available experimental data from a similar study. The study of interest was conducted by Natsui [35] on a flat plate with a vertical row of film

cooling holes inclined at 20 degrees to the free stream. Comparison of a single hole in the array was expected to have nearly identical flow characteristics to the first film cooling hole in the CFD model. The experimental data collected was on adiabatic effectiveness using pressure-sensitive paint (PSP) for a range of blowing ratios, as shown in Figure 57. To directly compare the results of the CFD, the mesh for the conducting material was removed, so the non-dimensional surface temperature was the adiabatic effectiveness, instead of overall.

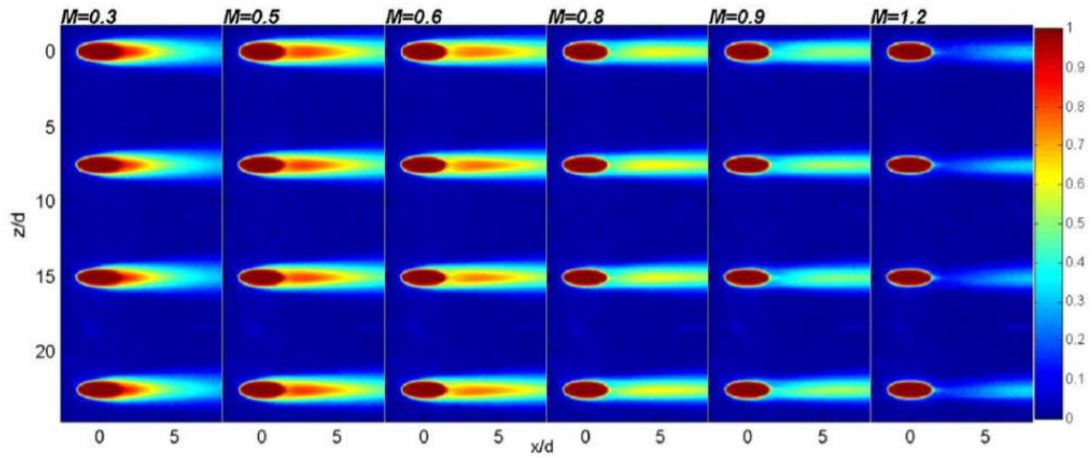


Figure 57: Adiabatic film cooling effectiveness profiles for a single row for $M=0.3-1.2$. Freestream and coolant flow is from left to right. From ref. [35]

The adiabatic CFD model was tested at three blowing ratios, $M=0.3$, 0.5, and 0.9. These ratios were chosen because they are nearly identical to the eventual test points at $M=0.25$, 0.5, and 0.9. The results for both models and the corresponding experimental data are shown in Figure 58. At a blowing ratio of 0.3, the $k-\omega$ model predicts greater adiabatic effectiveness than the $k-\epsilon$ model. The $k-\epsilon$ model most closely predicts the experimental results. At $M=0.5$, the $k-\omega$ model switches, now underpredicting the experimental results.

The $k-\varepsilon$ model, however, predicts more elevated effectiveness at $M=0.5$ than $M=0.3$, like the experiment. At $M=0.9$, $k-\omega$ predicts very little effectiveness, while the $k-\varepsilon$ model predicts higher effectiveness. For all three blowing ratios, the realizable $k-\varepsilon$ model most closely matched the experimental data. Realizable $k-\varepsilon$ has also been used successfully by Harrison and Bogard [36], among other researchers. Therefore, for all test points, it was the turbulence model used.

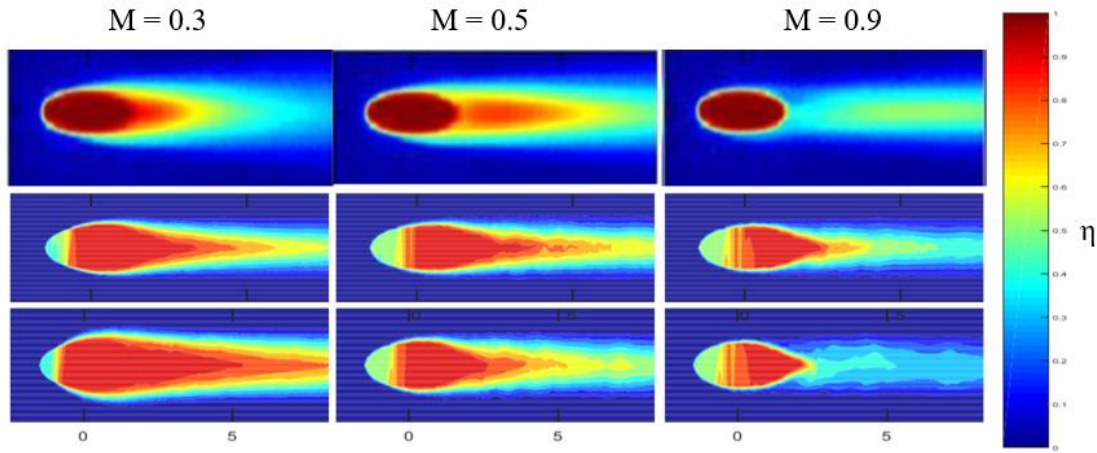


Figure 58: Comparison of adiabatic film cooling effectiveness. Top row: experimental data. Second row: realizable $k-\varepsilon$ model. Third row: SST $k-\omega$ model.

The realizable $k-\varepsilon$ model has several disadvantages. First, it assumes the flow is fully turbulent. The computational model does not have a boundary layer trip on the top surface, so it is uncertain whether the flow has transitioned in any location, without post processing the data and examining it. Second, the model poorly predicts strong separation or swirling flows, such as can occur at the exit of a film cooling hole. Finally, the turbulent viscosity is modelled, rather than solved in the underlying equation set. [37] Even with

these underlying limitations, this model has significant use in previous literature and typically produces quick reliable results.

After the model was read into Fluent's 3D double precision solver, it was correctly scaled and the mesh reordered. The solver used was a pressure based steady solver. Next the energy solver was turned on and the viscous solver was set to realizable $k-\varepsilon$ with enhanced wall treatment including pressure gradient and thermal effects. The conducting material had to be defined within Fluent using the physical properties for Corian. These physical properties were then set for the solid zone mesh previously defined in Pointwise. The boundary conditions were the next thing set. The inlet velocities and mass flow rates for each test case are shown in Table 5. Additionally, the freestream temperature was set to 320 K and the coolant was set to 300 K.

Table 5: Boundary conditions for each test case

Freestream Reynolds number	Freestream velocity (m/s)	Blowing Ratio	Coolant Mass flow rate (kg/s)
3,400	12	$M = 0.25$	0.0003
		$M = 0.5$	0.0006
		$M = 0.9$	0.0011
25,000	111.5	$M = 0.25$	0.0027
		$M = 0.5$	0.0055
		$M = 0.9$	0.0100

The next step in setting up the flow solver was to specify the solution methods. The setting which produced the most converged solution was coupled pressure-velocity coupling, the Green-Gauss node based gradient spatial discretization, standard pressure discretization, second order upwinding for momentum and energy, and first order

upwinding for turbulent kinetic energy and turbulent dissipation rate. The reference condition was set at the freestream inlet. All settings not explicitly mentioned were left as the standard Fluent preset.

The solution was initialized using Fluent's standard initialization algorithm and the solution was iterated until the residuals either reached 1×10^{-6} or stopped changing, as shown in Figure 59.

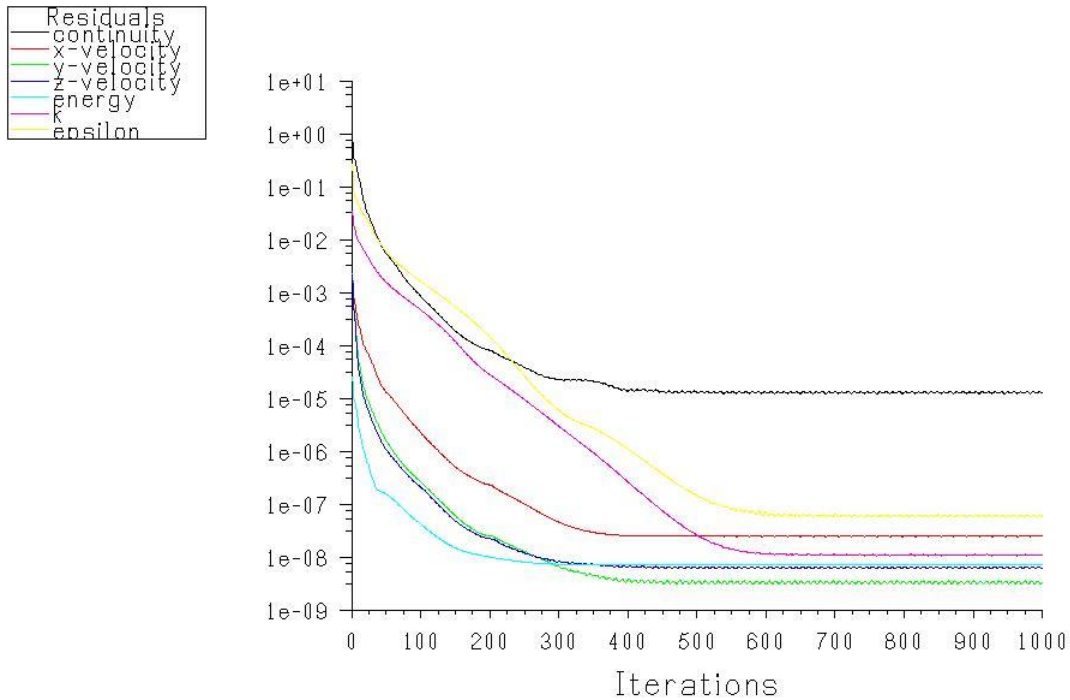


Figure 59: Residuals for a converged solution

VI. Computational Results

All the computational fluid dynamics results were processed using *Tecplot 360* and MathWorks *MATLAB*. The surface temperature and grid node locations were extracted for the top surface of the conducting material. The temperature was nondimensionalized to overall effectiveness, ϕ , using Equation (2.4).

6.1 Test Cases

The test cases were chosen to give the most complete picture of conjugate heat transfer within the model. Two Reynolds numbers were tested: the low Reynolds number is 3,400 based on coolant hole diameter. The flow velocity for this case is identical to the experimental Reynolds number based on hole diameter. The second Reynolds number is 25,000, a significantly higher Reynolds number, chosen to provide a contrast to the low Reynolds number cases. Three blowing ratios were tested, $M=0.25$, $M=0.5$, $M=0.9$ which are identical to the blowing ratios tested experimentally. Because the temperature difference was so small, the density ratio for all the cases tested was 1.0-1.1. Three nonphysical modifications were made to the computational model to distinguish the effects of each type of cooling, providing information that would be impossible to be obtained physically.

The first modification was made in order to isolate the effects of impingement: the bottom of the conducting surface was made adiabatic. The flow path was not affected, the flow still impinged on the bottom surface, flowed through the cooling holes, and formed a film over the top of the surface. The only modification was to set the convection coefficient at the bottom row of cells to zero. The second modification was made to isolate cooling

within the holes, the walls each coolant hole were made adiabatic. Like the cold wall modification, this only affected the grid cells on the boundary of the coolant holes.

The third modification was made to isolate the effect of film cooling. In this case, the cells at the exit of each film cooling hole were set to be a pressure outlet. The corresponding cells on the other side, in the freestream, were set as a no slip, adiabatic wall. In this case, every surface transferred heat like the baseline, only the flow was affected. The flow path and velocities were unchanged for the impingement zone and as it flowed through the film cooling holes. The sink at the hole exit only affected the coolant flow. The wall condition in the opposing direction kept the freestream boundary layer as if it were only a flat plate.

Figure 60 shows each of these modifications on a small section of the conducting material. Figure 60(a) shows a baseline model where the coolant flow is shown in blue, the freestream is shown in green, and the conducting walls are black. Figure 60(b) shows the case where the bottom wall is now adiabatic, shown in red. Figure 60(c) shows adiabatic coolant hole walls. Finally, Figure 60(d) shows the outlet in yellow at the exit of the film cooling holes with a no slip adiabatic wall, in red, above it. Note that no film exits the cooling holes in Figure 60(d).

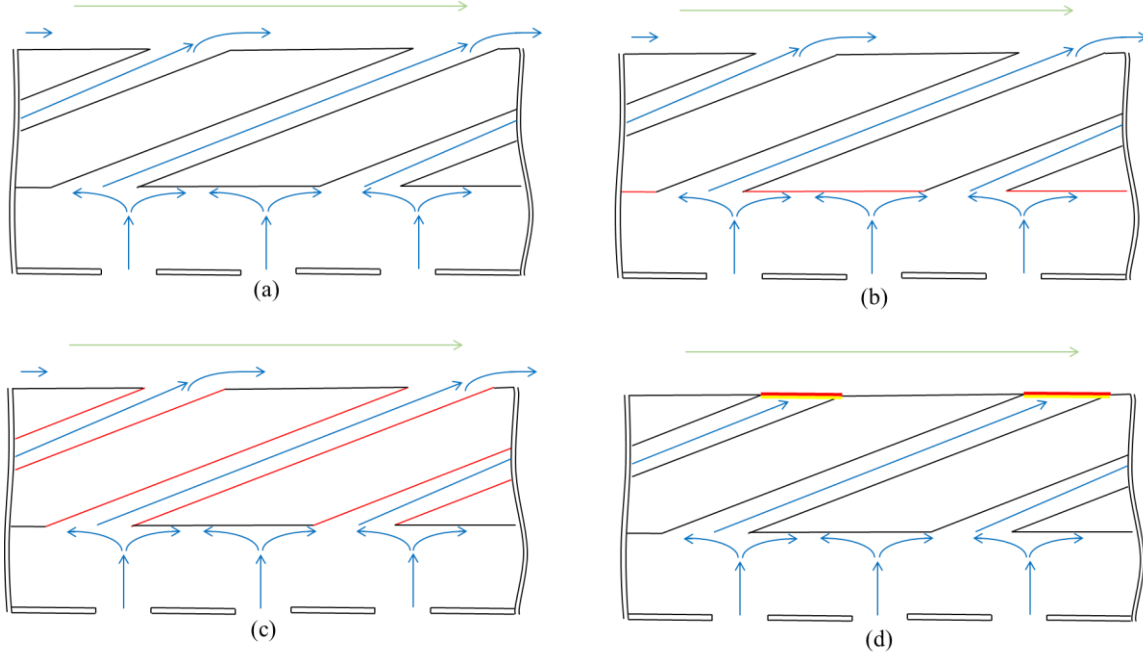


Figure 60: Modified boundary conditions in the CFD model. Diagram (a) shows the baseline condition, (b) shows impingement removed, (c) shows hole cooling removed, and (d) shows film cooling removed.

6.2 Low Reynolds Number Simulations

The first simulations were performed at $Re=3,400$, which matched the experimental Reynolds number, based on hole diameter. Figure 61 shows the adiabatic effectiveness at the low Reynolds number condition. The top figure is at $M=0.25$, next is 0.5, and the last is 0.9. The film has an increased effect as it travels downstream, to the right. $M=0.5$ is the most effective blowing ratio for film coverage. This is most easily seen by looking between the first and second cooling holes. The entire width past the last cooling hole is cooler with increasing blowing ratio. Overall, this geometry is very good for film cooling because the holes are tightly packed and exit in the same direction as the freestream.

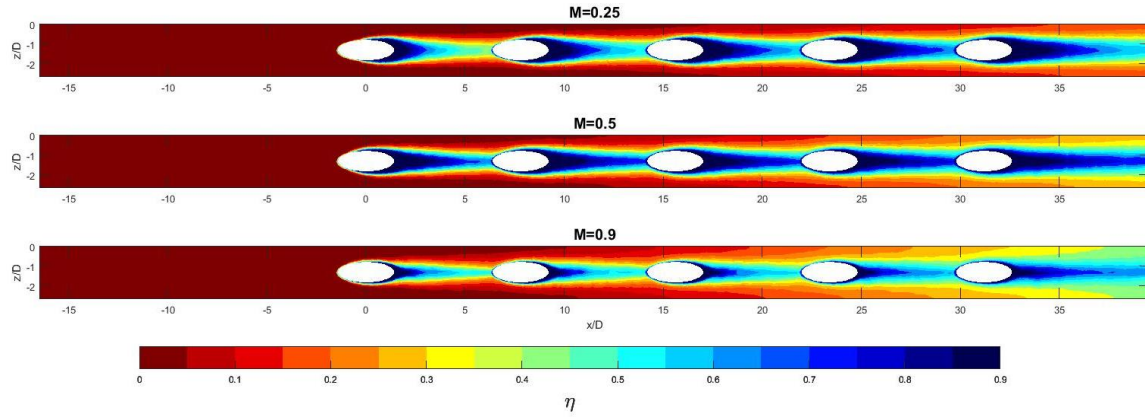


Figure 61: Adiabatic effectiveness for $Re = 3,400$

Figure 62 shows overall effectiveness for the low Reynolds number at the lowest blowing ratio. It shows four figures: the top is the baseline condition where the all surfaces are diabatic and the flow impinges on the bottom of the conducting material, goes through the film holes, and forms a film over the top of the surface. The next is the ‘impingement removed’ condition which shows the effect of making the bottom surface adiabatic. It is labelled as the impingement removed case because it shows the effect of removing heat transfer due to impingement on the bottom surface, the flow still flows through the impingement holes like in the baseline case. The next case is ‘hole cooling removed’ where the walls of the film cooling holes are adiabatic. The final case, ‘film cooling removed’ is the case where the coolant disappears through an outlet at the exit of each cooling hole leaving the top surface affected by the freestream only.

There is very little change in overall effectiveness between the impingement removed and baseline cases. The area past the last coolant hole shows a slight reduction in effectiveness, by 0.05. Hole cooling removed shows much more reduction in overall effectiveness. There is a significant reduction in ϕ upstream of the first cooling hole. After

each coolant hole, there is cooling from the film, increasing effectiveness with each successive hole exit. The final plot shows the effect of removing the film. The most obvious effect is visible past the last cooling hole exit, where the film has not built up and there is no hole underneath to provide cooling.

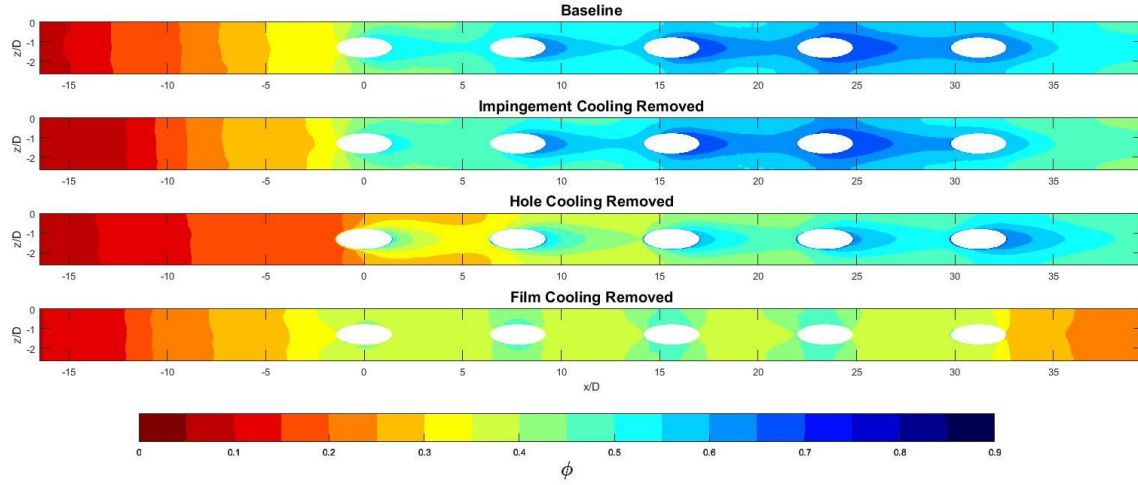


Figure 62: Overall effectiveness for $Re = 3,400$ $M=0.25$

To make analysis simpler, the difference between the baseline and the modified condition, or the $\Delta\phi$, is shown in Figure 63. The plot was generated by subtracting the baseline case from the modified case. A negative $\Delta\phi$ shows where the modified case is now hotter than the baseline case. The places where the modification results in the top surface becoming cooler cause a decrease in $\Delta\phi$. For example, in the top plot in Figure 63, the only region of non-zero $\Delta\phi$ is past the last cooling hole exit, where there is a slightly negative $\Delta\phi$. Therefore, this area is slightly cooled by the impinging jets hitting the bottom surface.

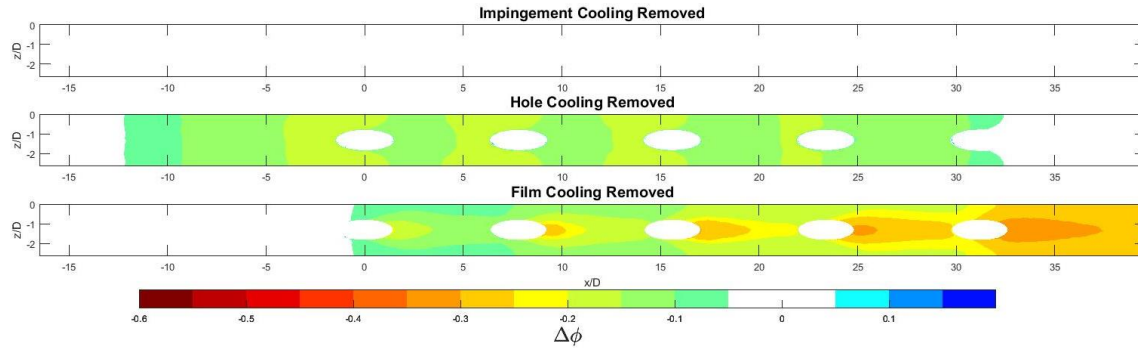


Figure 63: $\Delta\phi$ for $Re = 3,400$ $M=0.25$

The effect of cooling within the holes is highest directly upstream of each coolant hole exit. Figure 64 shows the same $\Delta\phi$ plot as above in Figure 63, but with a cross sectioned side view underneath with vertical lines drawn at critical points. The first line is at the start to the inlet of the first cooling hole. This also almost identically lines up with where $\Delta\phi$ begins to decrease on the top surface. $\Delta\phi$ steadily decreases from there to the coolant hole exit. As the coolant flows through the cooling holes, it cools the surrounding wall via convection. The top surface is then cooled via conduction from the cooling hole walls. The less material between the top surface and the cooling holes, the cooler the top surface.

Past the first coolant hole exit, $\Delta\phi$ slightly increases, then at the exit to the second coolant hole decreases back to the same value at the exit to the first hole. This trend continues down to the fourth hole exit. The slight increase occurs because the next coolant hole is only halfway through the height of the material, so there is more material between the hole and the surface, so less cooling is observed. Directly before the exit to the last cooling hole, ϕ does not decrease again because there is no coolant hole beneath. The

previous holes had another coolant hole inlet directly below the exit of the hole before. The coolant going into the inlet cooled the hole walls, which was conducted to the cooling hole above. Via convection, the coolant in the previous hole was slightly cooled which then cooled the top surface even more. Because the last coolant hole does not have any cooling holes directly beneath it, there is no extra cooling the coolant, so the top surface is warmer. After the last cooling hole, there is a slightly negative $\Delta\phi$ due to 3D conduction from the coolant hole.

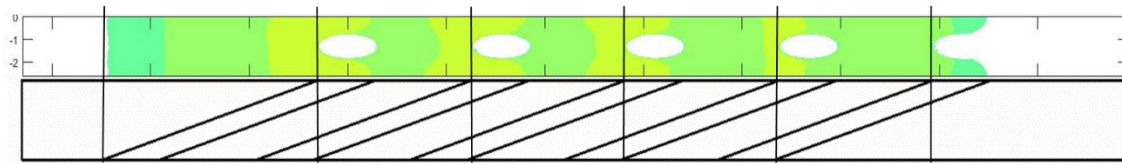


Figure 64: $\Delta\phi$ for hole cooling, from Figure 63, and side view of holes for comparison

The final plot in Figure 63 shows the effect of removing film cooling. ϕ is reduced following each film cooling hole, with increasing effectiveness the further downstream. This finding agrees with the adiabatic film cooling effectiveness in Figure 61. After each hole exit, the film grows, increasing the thermal and viscous boundary layers. As the film grows, the less the layer closest to the top surface is warmed by the freestream and the more it cools the surface via convection.

The $\Delta\phi$ distributions for each plot were averaged to find the average effect of each modification. The coolant holes exits were not included in the average, only the solid top surface. The average $\Delta\phi$ for impingement removed was -0.02, -0.10 for hole cooling

removed, and -0.13 for film cooling removed. This indicates film cooling was the most effective at cooling the surface for this case.

The next case tested was at $M=0.5$, the most effective blowing ratio for film cooling. Overall effectiveness and $\Delta\phi$ plots are shown in Figure 65 and Figure 66, respectively. The trends in this case closely match the trends at $M=0.25$. One notable difference in $\Delta\phi$ for hole cooling removed is less negative $\Delta\phi$ with increasing downstream distance. Hole cooling is most effective at cooling the surface in the areas with less film cooling. Downstream the film builds up and cools the surface just as much, if not more so, than hole cooling. Upstream of the first cooling hole and between the first few holes, the film is nonexistent or very thin and hole cooling is the most dominant part of cooling.

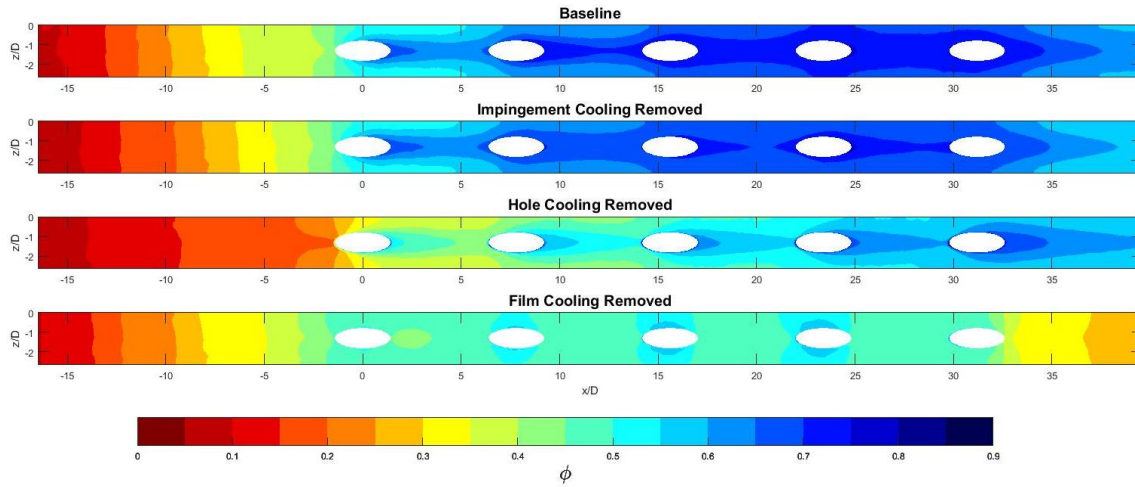


Figure 65: Overall effectiveness for $Re = 3,400$ $M=0.5$

The average $\Delta\phi$ s for this case are -0.02 for impingement removed, -0.12 for hole cooling removed, and -0.14 for film cooling removed. Hole cooling and film cooling both have increased in effectiveness, while impingement remains approximately the same. Film cooling, again, is the dominant type of cooling. The impingement $\Delta\phi$ plot does not show

any contours because the values are between -0.05 and 0.05 for the entire top surface. Impingement provides hardly any cooling to the top surface.

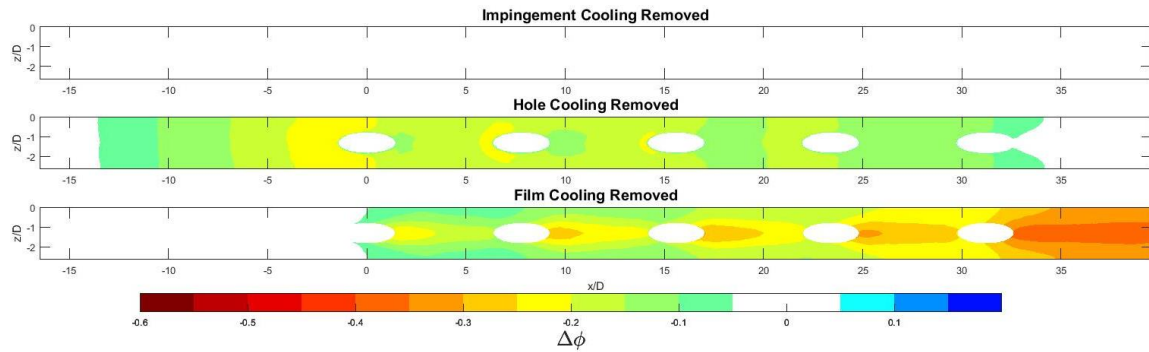


Figure 66: $\Delta\phi$ for $Re = 3,400$ $M=0.5$

The final case tested at this Reynolds number was at $M=0.9$. Overall effectiveness and $\Delta\phi$ plots are Figure 67 and Figure 68. This case shows that hole cooling is more effective than film cooling. The area averaged $\Delta\phi$ is -0.02 for impingement removed, -0.15 for hole cooling, and -0.10 for film cooling. The hole cooling is increased likely because the flow rate has increased, which keeps the bulk average temperature of the coolant within the holes lower. The increased blowing ratio negatively impacts film cooling because the coolant forms a jet which lifts off the top surface, allowing the hot freestream to remain in contact with the surface. Impingement again has limited effectiveness compared to hole and film cooling.

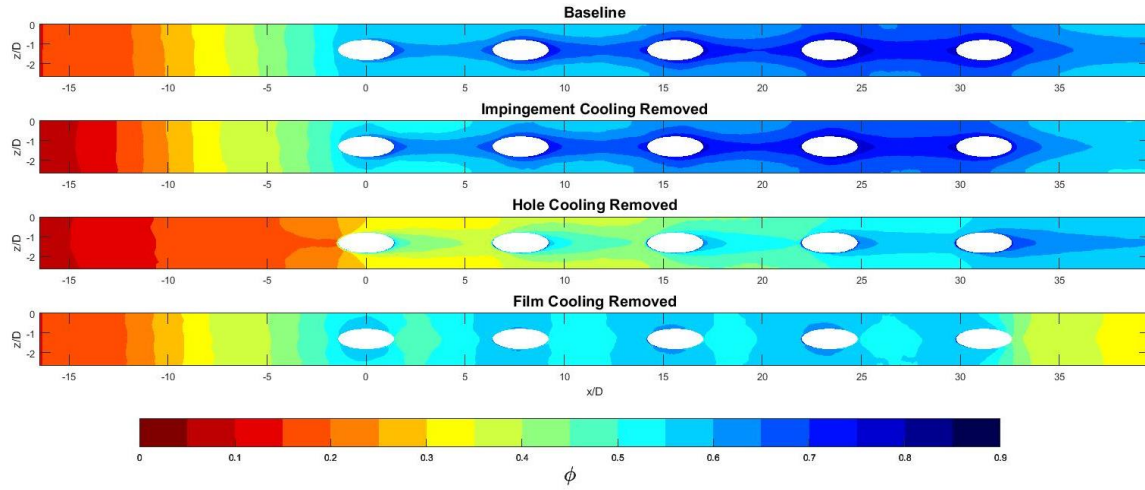


Figure 67: Overall effectiveness for $Re = 3,400$ $M = 0.9$

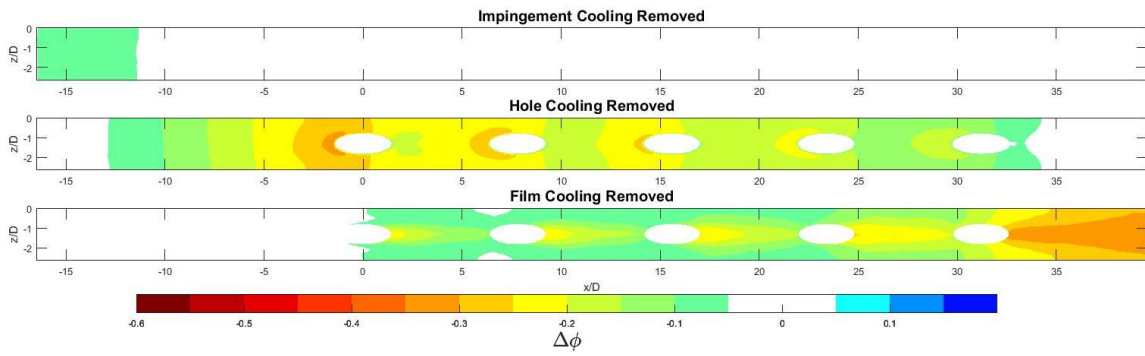


Figure 68: $\Delta\phi$ for $Re = 3,400$ $M = 0.9$

The final figure covered in this section, Figure 69, is a more detailed look at the effect of removing impingement. This is the same data plotted as in the previous three cases, but with a legend $1/10^{\text{th}}$ of the scale previously used. The previous legend was an appropriate scale for the hole and film cooling removed cases, but was not appropriate to show the effect of removing impingement. The primary areas effected by impingement are upstream of the first cooling hole exit and downstream of the last cooling hole. The contours in Figure 69 show increasing effect of impingement with increasing blowing ratio in the area upstream of the cooling hole exits. The opposite is true for the area

around the cooling hole exits and the areas downstream of the cooling holes. Here, effectiveness decreases as cooling from the holes and film increasing with increasing blowing ratio.

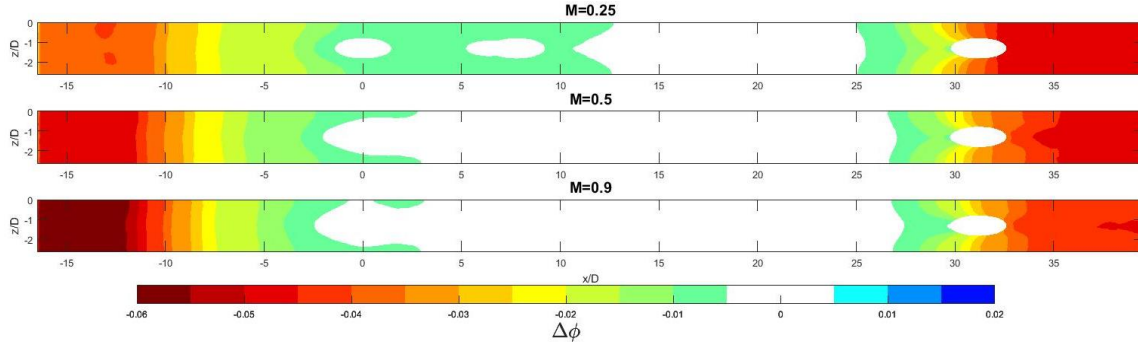


Figure 69: Impingement removed $\Delta\phi$ with 1/10th legend at $Re = 3,400$

6.3 High Reynolds Number Simulations

The next group of simulations were conducted at $Re = 25,000$ for the same three blowing ratios. The adiabatic film cooling effectiveness for these cases are shown in Figure 70. For all three blowing ratios, there is less spreading across the width of the surface due to the higher Reynolds number. The jets are more effective along the centerline because they stay attached to the surface for $M=0.25$ and 0.5 . At $M=0.9$, there is far less effectiveness past the coolant hole exits because the film is not building. The coolant jets liftoff and mix with the freestream and the thermal boundary does not develop as fully as at the low Reynolds number condition.

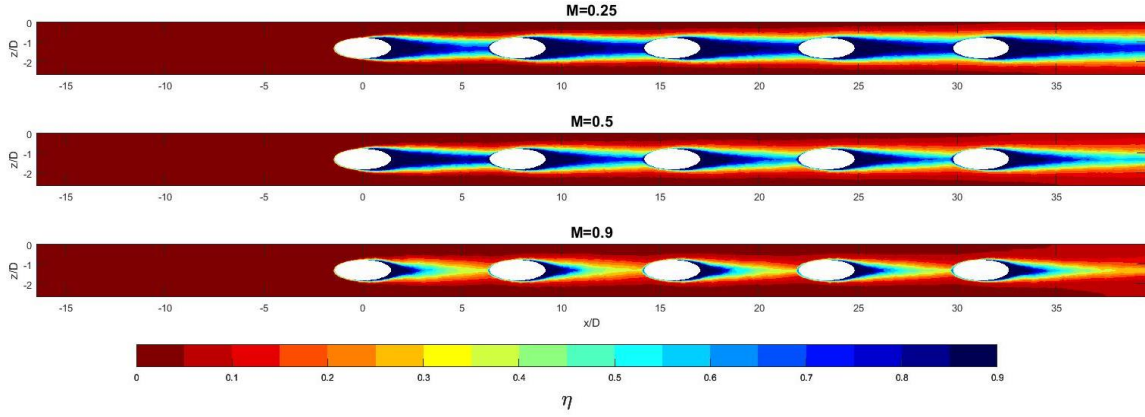


Figure 70: Adiabatic effectiveness for $Re = 25,000$

Results for $M=0.25$ are shown in Figure 71 and Figure 72 for overall effectiveness and $\Delta\phi$. The overall effectiveness contours for removing hole cooling show much less film spreading compared to the low Reynolds number condition. The area averaged $\Delta\phi$ for impingement removed is -0.01, -0.10 for hole cooling removed, and -0.16 for film cooling removed. The $\Delta\phi$ plot for hole cooling removed looks very like the same case at the low Re condition. The effects of film cooling do not travel past the midplane of the first exit hole, therefore any effect before the first hole exit is solely due to hole cooling. Film cooling has a very negative $\Delta\phi$ directly past the exit of each cooling hole exit. The dominant cooling type for this case is film cooling. Impingement has a slightly negative effect on $\Delta\phi$.

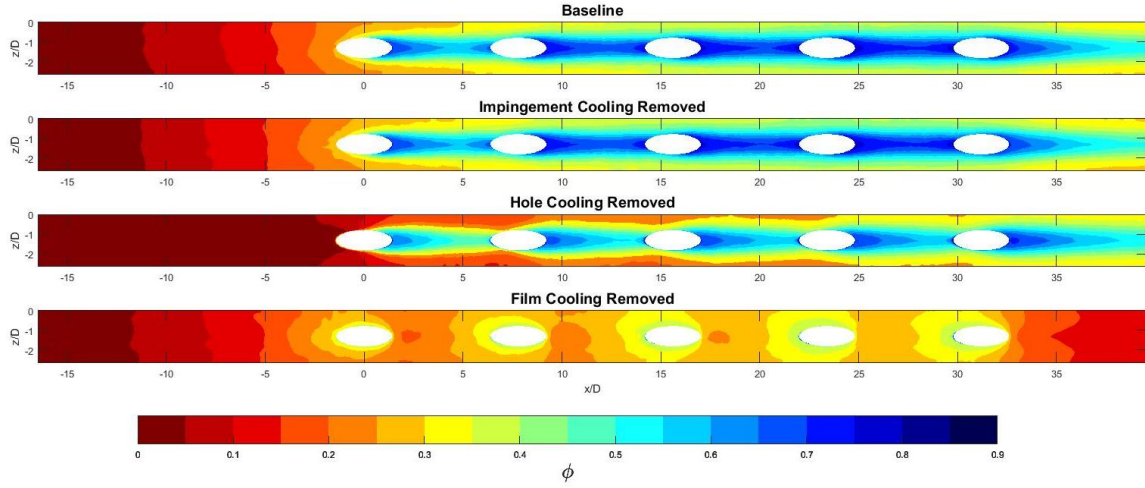


Figure 71: Overall effectiveness for $Re = 25,000$ $M = 0.25$

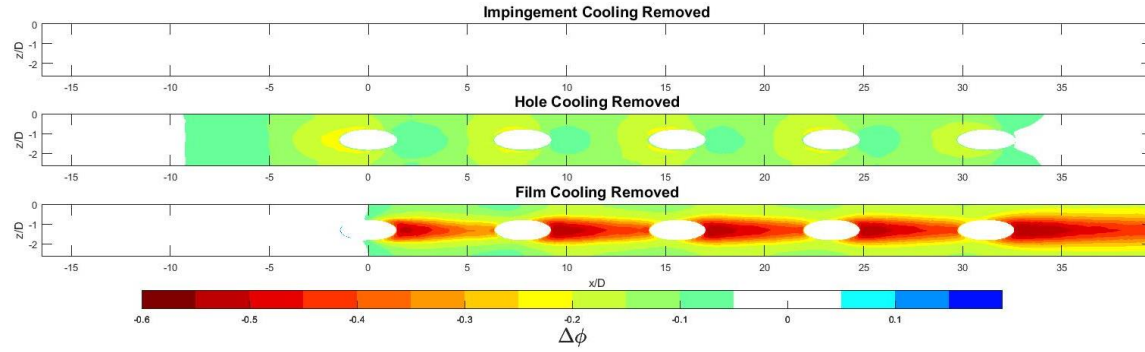


Figure 72: $\Delta\phi$ for $Re = 25,000$ $M = 0.25$

The next case was at $M=0.5$, results from which are shown in Figure 73 and Figure 74. The area average $\Delta\phi$ is -0.01 for impingement removed, -0.12 for hole cooling removed, and -0.12 for film cooling removed. Compared to $M=0.25$, the effect of film cooling has decreased and hole cooling effectiveness increased so they are nearly equal. Hole cooling effectiveness increasing with blowing ratio is also seen in the low Reynolds number simulations. Film cooling effectiveness, however, is very different at this Reynolds number. At all blowing ratios at the low Reynolds number, the film seemed to build up with downstream distance resulting in increasing effectiveness with each successive hole

exit. In this case however, $\Delta\phi$ remains the same, if not more positive with each downstream hole. The film does not build into a thicker layer with more effectiveness. The only effect of increasing the blowing ratio on film effectiveness is to cause more jet lift off, reducing the cooling. Impingement again has a slightly negative $\Delta\phi$.

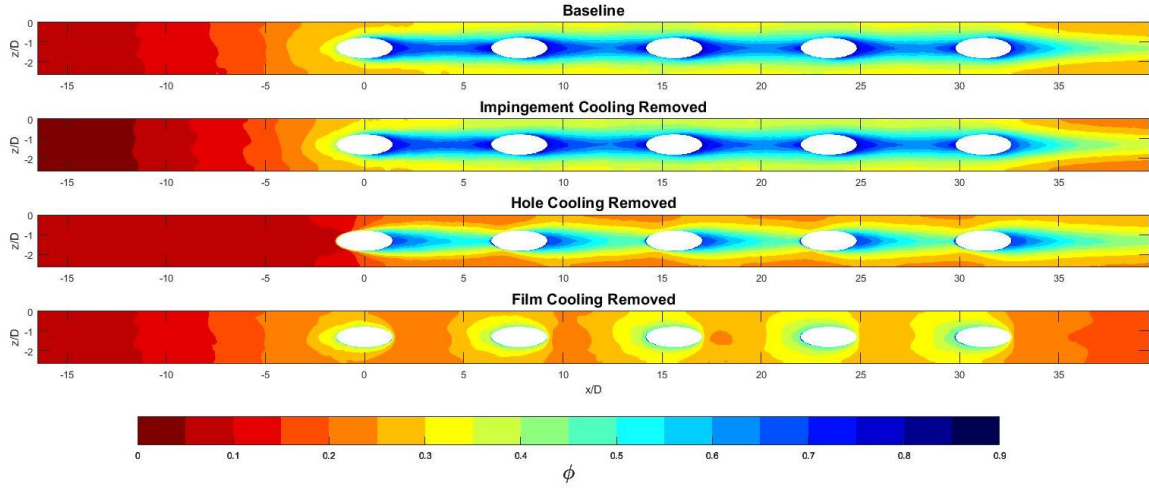


Figure 73: Overall effectiveness for $Re = 25,000$ $M = 0.5$

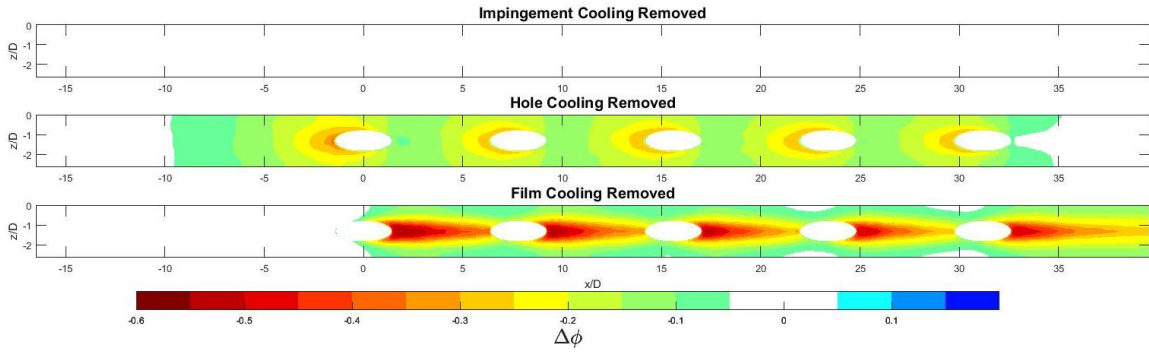


Figure 74: $\Delta\phi$ for $Re = 25,000$ $M = 0.5$

The final blowing ratio tested was $M=0.9$. The ϕ and $\Delta\phi$ plots of the top surface are shown in Figure 75 and Figure 76. The average $\Delta\phi$ is -0.01 for impingement removed, -0.13 for hole cooling removed, and -0.04 for film cooling removed. In this case hole cooling is by far the most effective at cooling the top surface. Film cooling is only effective along the

centerline directly downstream of each film cooling hole. The film is immediately lifting off the surface, preventing a cooling film from forming. Hole cooling effectiveness however increased with blowing ratio again. Impingement, again, is only very slightly effective.

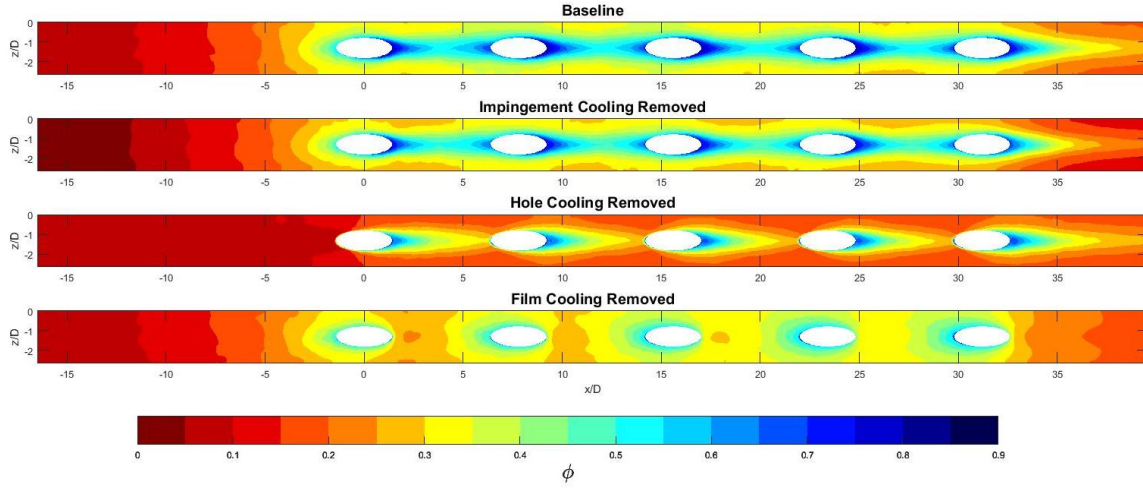


Figure 75: Overall effectiveness for $Re = 25,000$ $M = 0.9$

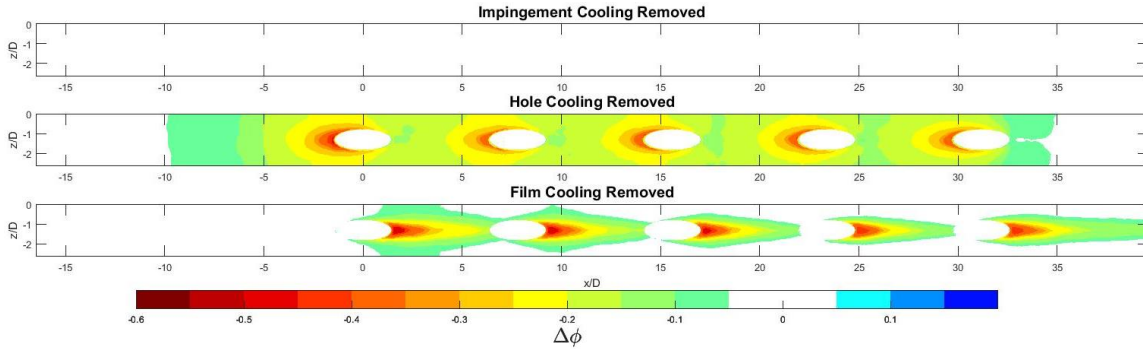


Figure 76: $\Delta\phi$ for $Re = 25,000$ $M = 0.9$

Impingement $\Delta\phi$ contours with a more appropriately scaled legend are shown below in Figure 77. The same trends as in the low Reynolds number case are evident, impingement is most effective upstream and downstream of the cooling hole exits. Increasing the blowing ratio again increasing effectiveness upstream of the cooling hole

exits. However, at the higher blowing ratio, increasing blowing ratio also increases effectiveness downstream of the cooling holes. This is likely due increased jet liftoff at the higher Reynolds number, which reduces the effect of film cooling in that area.

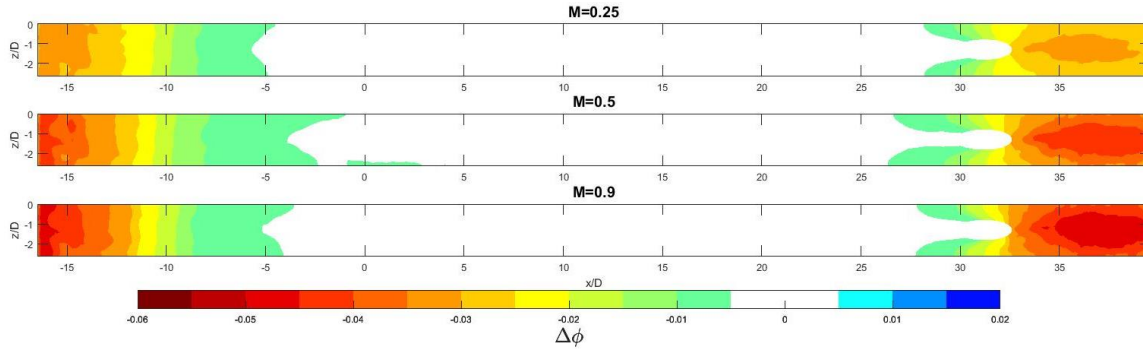


Figure 77: Impingement removed $\Delta\phi$ with 1/10th legend at $Re = 25,000$

6.4 Overall Trends

The area weighted averaged surface heat transfer coefficients from both the top and bottom surfaces were extracted using Fluent. The average internal heat transfer coefficient for the low Reynolds number was $-6.8 \text{ W/m}^2\text{-K}$, while the average external h was $-52.0 \text{ W/m}^2\text{-K}$. Therefore, the average h ratio was 7.8. The external heat transfer coefficient was used to compute the Biot number using Equation 2.12. At the low Reynolds number condition, the average Biot number was 0.25. At the high Reynolds number condition, the flowrates significantly increased, which changed the heat transfer coefficients. The internal average heat transfer coefficient was $-10.1 \text{ W/m}^2\text{-K}$ and the outer h was $-220.6 \text{ W/m}^2\text{-K}$. This significant decrease in the external h increased the ratio of heat transfer coefficients to 21.9. The Biot number also increased to 1.1.

Per [38], a Biot number range of 0.3-0.6 matches engine conditions of a CFM56 high pressure turbine. The Biot number range for these experiments was below the range at the low Reynolds number and above it at the high Reynolds number. The same paper also says a typical engine condition h ratio is 0.5. The ratio in these cases was well above 0.5 for both Reynolds numbers. Because neither h nor the Biot number match typical engine conditions, the data may lead to poorly scaled results. For all six cases, the internal h for the case where the effect of impingement was removed was extracted. The average h was 0 W/m²-K, as expected since the wall was set as adiabatic.

The coolant warming factor was found using the average temperature at the hole exit plane. The coolant warming factor for all six test cases is shown below in Figure 78. The lower Reynolds number had a lower coolant warming factor than the high Reynolds number which indicates there was more warming of the coolant within the plenum and cooling holes for the cases with lower coolant flow rate. Another trend is the coolant warming factor approaches unity with increasing blowing ratio. Again, this shows increasing coolant flow rate leads to less coolant warming. This agrees with the previous assertion that hole cooling is more effective at higher blowing ratios because the coolant bulk average temperature stays lower with increasing velocity.

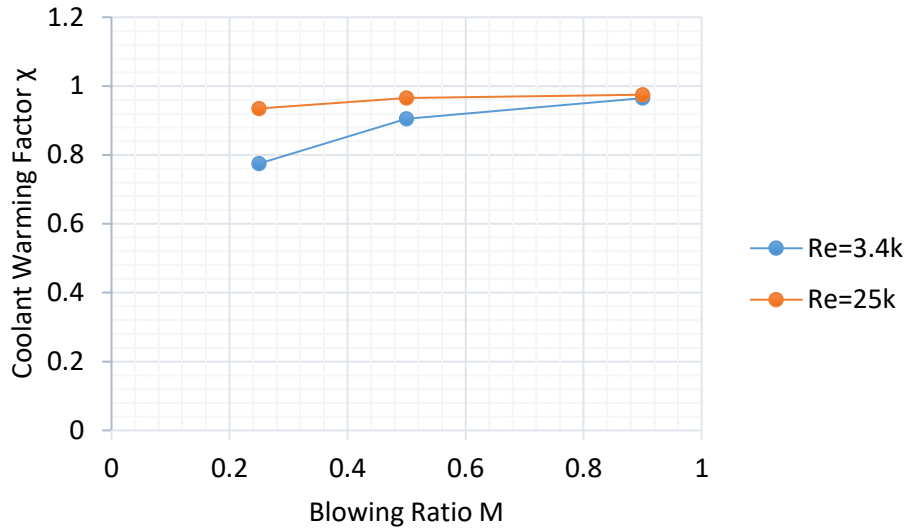


Figure 78: Coolant warming factor for CFD cases

The average blowing ratio for each hole was calculated by finding the actual blowing ratio for each hole in all six baseline cases and averaging the difference between the expected and actual blowing ratios. The average percent difference from the expected blowing ratio for each cooling hole is plotted in Figure 79. On average the first and last film cooling holes overshoot the expected blowing ratio, while the middle three holes did not, on average, reach the prescribed blowing ratio. This result makes sense because there are many impingement holes with no film cooling hole inlets nearby upstream and especially downstream of the cooling hole inlets, as shown in Figure 50. All this extra coolant flows into all the cooling holes, but it makes sense extra would flow into the holes closest to the ends, the first and fifth cooling holes.

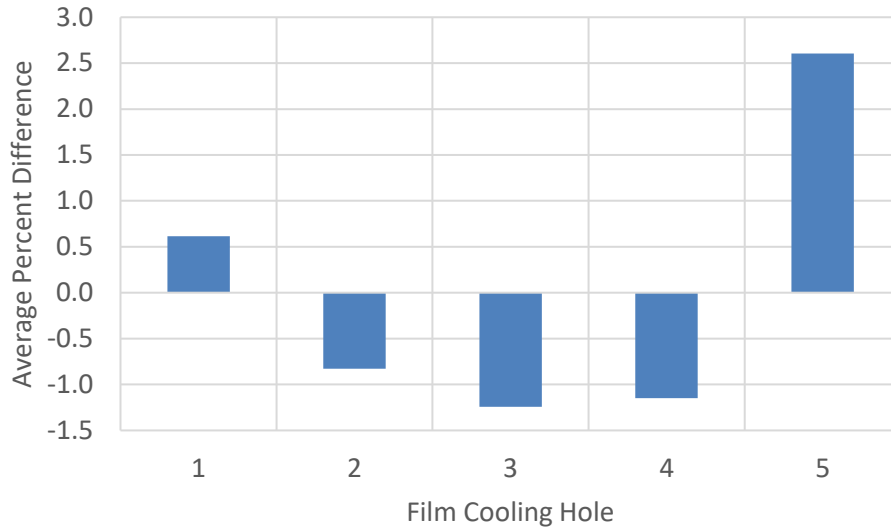


Figure 79: Average percent difference in blowing ratio for each cooling hole

Figure 80 shows the average $\Delta\phi$ values for each case for each modification. The overall trend in hole cooling is increased effectiveness with increasing blowing ratio. Convection primarily caused cooling within the holes. Increasing the flow velocity increases the heat transfer to the wall because it decreases the bulk average temperature by the walls. Therefore, this trend makes sense. Another trend is decreasing film effectiveness with increasing blowing ratio. Increasing the coolant flow can contribute to jet liftoff, thereby decreasing film effectiveness and overall cooling. This is easily seen by looking at the adiabatic film effectiveness contours in Figure 61 and Figure 70. Hole cooling surpassed film cooling and became the dominant cooling factor at $M=0.9$.

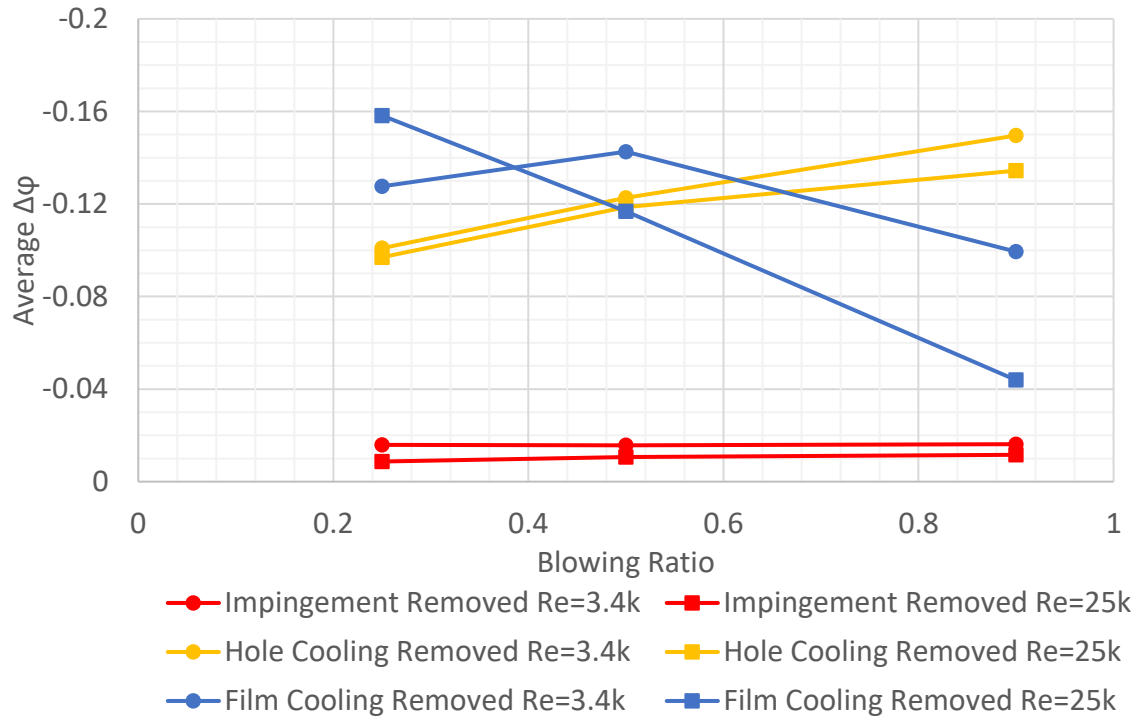


Figure 80: Comparison of $\Delta\phi$ values

Impingement effectiveness increases with coolant flow rate. The primary areas cooled by impingement are the areas outside of the cluster of cooling hole exits. Impingement was slightly more effective at the lower Reynolds number condition. Overall, impingement has much less impact on overall effectiveness than hole cooling. Impingement contributed 3-6% of overall cooling in every case tested. Hole cooling, on the other hand, contributed 35-70% of overall cooling. Hole cooling, then is over 10 times more effective than impingement at cooling a flat plate. Therefore, it makes much more sense to preform research with aims to improve hole cooling, then improving impingement. One possible way to improve hole cooling is by increasing the roughness within the cooling holes.

6.5 Effects of Roughness inside the Film Holes

To test the effect of roughness on hole cooling, three roughness conditions were applied to the CFD model using the boundary conditions settings within Fluent. The three roughnesses chosen were $R_a/D_h=0.007$, 0.010, and 0.020. A roughness of $R_a/D_h=0.007$ is equivalent to the roughness of 50 grit sandpaper when scaled according to correlations in an essay by Nanovea [39], an optical measuring device manufacturer. These roughnesses match closely to the values for typical roughnesses in aero engines outlined by Schroeder [8].

The CFD model was tested at the high Reynolds number condition at a blowing ratio of 0.5 with each of the roughnesses. The roughness condition was applied by specifying a roughness height and constant for each hole wall. The wall remained thermally coupled between the solid and air zones. The roughness height was determined by using Equation 2.11, and solving for k_s from R_a/D_h . The roughness constant was set to 0.5 for all the cases because, per the Fluent user guide [40], it reproduces Nikuradse's resistance data for pipes roughened with tightly-packed, uniform sand-grain roughness when used with k - ϵ turbulence models. The flow rate was not adjusted for any pressure drop through the roughened holes.

The resulting overall effectiveness contours are shown below in Figure 81. The contours are nearly identical, but the average ϕ values show that increasing roughness increases overall effectiveness. The average is 0.419 for the finest roughness, 0.423 for medium roughness, and 0.424 for the coarsest roughness, compared to the average for smooth holes, 0.394. This result is contrary to Schroeder's results for adiabatic

effectiveness using shaped holes, shown in Figure 7. The fact overall effectiveness increases with roughness when adiabatic effectiveness decreases shows the importance of hole cooling.

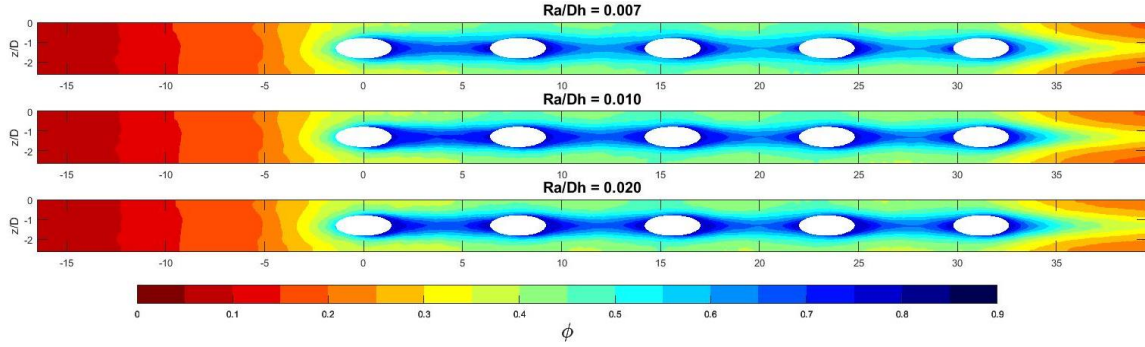


Figure 81: ϕ for three roughnesses at $Re=25,000$ $M=0.5$

The previous results for this Reynolds number and blowing ratio, shown in Figure 73 and Figure 74, were that film and hole cooling had an equal delta phi value, -0.12. So even though adiabatic effectiveness decreased due to roughness, an even greater increase in hole cooling had a similarly greater effect on overall effectiveness. Figure 82 shows $\Delta\phi$ for the three roughnesses. This is a different $\Delta\phi$ than the previously defined $\Delta\phi$. Instead of removing the effect of heat transfer or removing the film, this $\Delta\phi$ is defined as the difference between the roughened film holes and the baseline, or smooth film holes.

$$\Delta\phi = \phi_{rough} - \phi_{smooth} \quad (6.2)$$

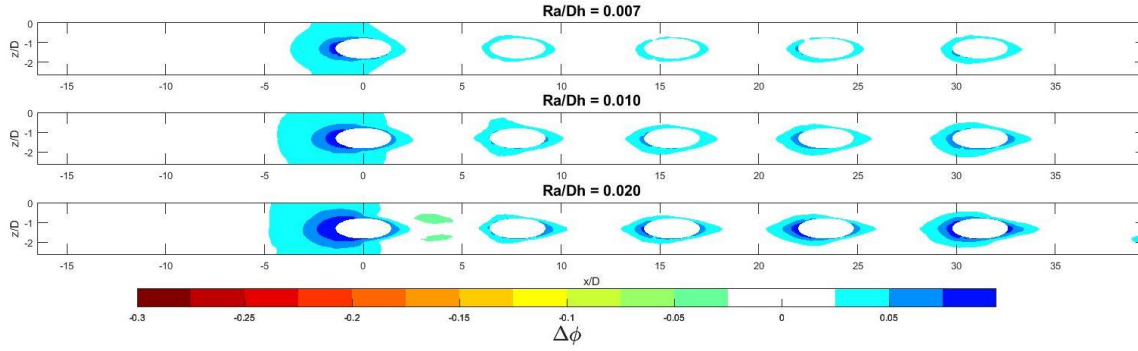


Figure 82: Roughness $\Delta\phi$ for $Re=25,000$, $M=0.5$

The contours in Figure 82 show a positive delta phi, which indicates that the modification, adding roughness, has a cooling effect on the top surface. $\Delta\phi$ is most positive upstream of the first cooling hole because this is where the conducting material is thinnest and where there is no effect from the film.

It is clear from these Schroeder's work η decreases with increasing Ra/D_h and M . However, it has also been shown hole cooling increases with roughness. The previous CFD results showed that hole cooling had a greater effect on overall effectiveness than film cooling with increasing blowing ratio (see Figure 80). This then leads to the question: will increasing the blowing ratio cause ϕ to increase with increasing Ra/D_h , even though η decreases? To answer this question, several higher blowing ratios were tested. The area average overall effectiveness for these cases is plotted in Figure 83.

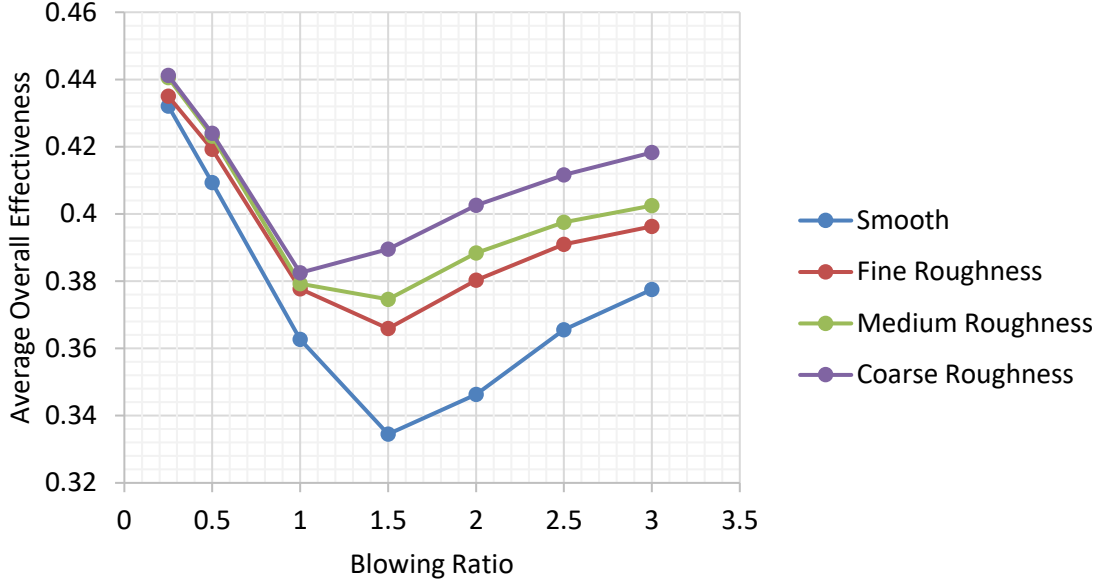


Figure 83: Area average ϕ for three roughnesses

At all blowing ratios, increasing the roughness in the cooling holes increases the average ϕ . Roughness is most effective at increasing ϕ at $M=1.5$ and 2.0 , even though Schroeder's data showed decreasing adiabatic effectiveness for $M=1.5-3.0$ with roughness. This data shows that film cooling has a diminishing effect on overall effectiveness, compared to hole cooling, due to jet liftoff. Hole cooling, however, has increased effectiveness as the velocity in the cooling holes increases, decreasing the bulk average temperature against the walls. These simulations show increasing roughness reduces the surface temperature from $M=0.25-3.0$, regardless of decreasing η .

6.6 Predicting Overall Effectiveness

The ability to predict overall effectiveness from more basic measurements is an intriguing concept. Recall Equation 2.17 on page 20, from Williams et al. [21], which predicted ϕ , based on overall effectiveness due to internal cooling only (ϕ_o), the adiabatic

effectiveness (η), and the coolant warming factor (χ). Williams found this method did not perform well which he contributed to the base assumption of 1D heat transfer, which is not the case around the film cooling holes. In the study, ϕ_0 was found experimentally by taping over the inlet to the cooling holes in the region of interest. As shown in the previous chapter on experimental results from blocking holes, this method removed the effect of hole cooling, along with film cooling. Hole cooling makes up a significant part of internal cooling, therefore removing it is likely to significantly change ϕ_p .

This research was able to remove only the effect of film cooling using the special computational boundary conditions, which is the definition of ϕ_0 . The adiabatic effectiveness distribution was also already computed for each case. The coolant warming factor from each case is shown in Figure 78. These values were input into Equation 2.17, to get a predicted overall effectiveness distribution over the top surface.

The inputs and predicted ϕ contours for $Re=3,400$, $M=0.25$ are shown in Figure 84. The top plot is contours of overall effectiveness in the baseline case; this is the distribution the predicted overall effectiveness should match. The second plot is overall effectiveness with film cooling removed, or the ϕ_0 distribution. The third plot is the adiabatic effectiveness, or η , distribution. The average coolant warming factor for this case was 0.775. These parameters yielded the final predicted ϕ distribution shown in the fourth plot.

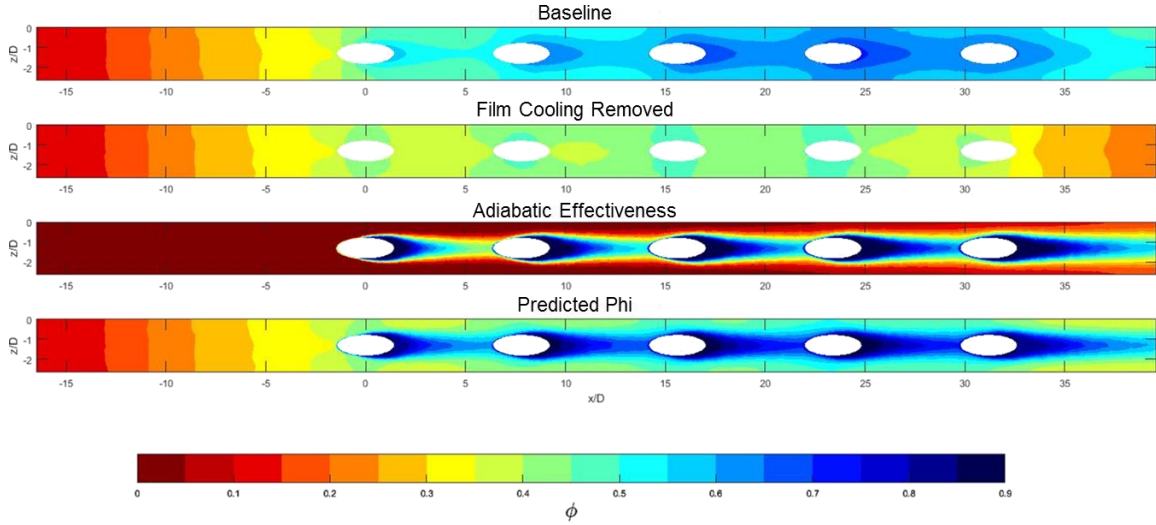


Figure 84: Predicted overall effectiveness for $Re=3,400$ $M=0.25$

The predicted ϕ distribution is a similar magnitude as the baseline case, but overpredicts effectiveness downstream of the coolant holes. Upstream of the first cooling hole, the predicted and baseline cases match nearly identically. This is where $\eta=0$, so it is exactly equal to the film cooling removed case. Downstream of the cooling holes, the η distribution quickly dominates, overpredicting the effectiveness in areas of high adiabatic film effectiveness. The η distribution also cause the prediction to under predict the spanwise effectiveness. The area average ϕ however is 0.47 for the predicted case, very close to the baseline ϕ of 0.48. The higher effectiveness directly downstream of the cooling holes is averaged out by the lower effectiveness in the spanwise direction.

Since the predictor method appears to be very sensitive to the η distribution, a case with a very different η distribution shows a much better match between the baseline and predicted cases. Figure 85 shows the same four plots for $Re=25,000$ and $M=0.9$, where the coolant warming factor was 0.975. This is the case with the smallest adiabatic effectiveness

distribution, due to jet liftoff and decreased spanwise effectiveness at the higher Reynolds number.

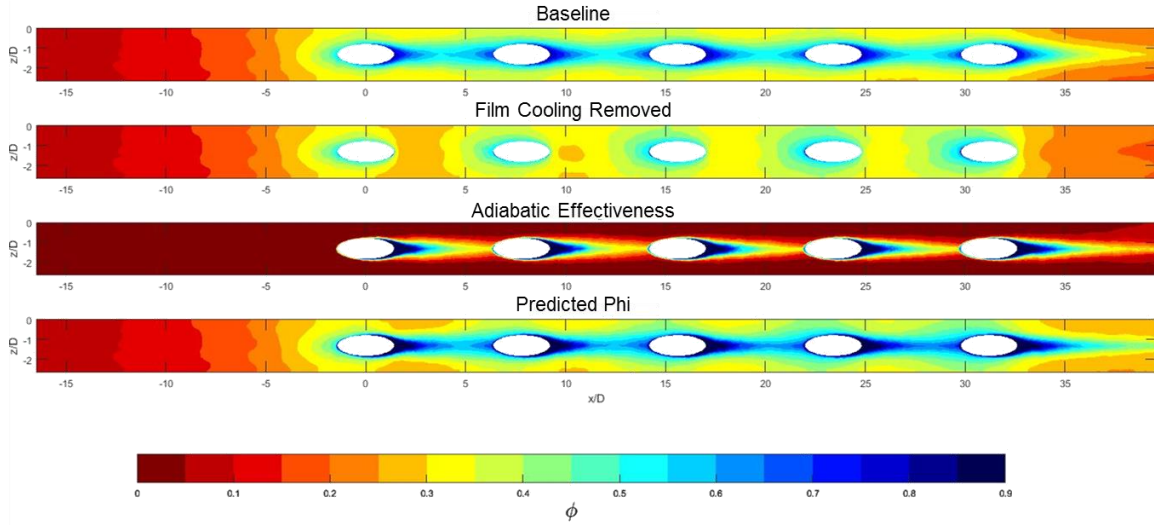


Figure 85: Predicted overall effectiveness for $Re=25,000$ $M=0.9$

The case shows a very good match between the baseline and predicted overall effectiveness. The predicted case again is nearly identical upstream of the first cooling hole. Downstream and between each cooling hole the predictor has a slightly higher effectiveness. The spanwise distribution however is nearly identical. The area average ϕ is 0.38 for the predicted case, compared to the baseline ϕ of 0.35. The predictor is higher than the baseline on average because the higher effectiveness downstream of each coolant hole is not balanced by a lower effectiveness predicted in the spanwise direction, as in the previous case.

The predicted ϕ distribution was calculated for all six computational cases. The remaining four plots can be found in Appendix A and they show identical trends to those already discussed. The area average ϕ for both the predicted and baseline cases are

compared in Figure 86. Both cases match very closely at $M=0.25$ and 0.5 for both Reynolds numbers. At $M=0.9$, the predicted ϕ is higher than the baseline case because the increased centerline effectiveness is not balanced by a decreased spanwise effectiveness prediction, as previously discussed. Overall, the predictor method does an excellent job of predicting the average overall effectiveness.

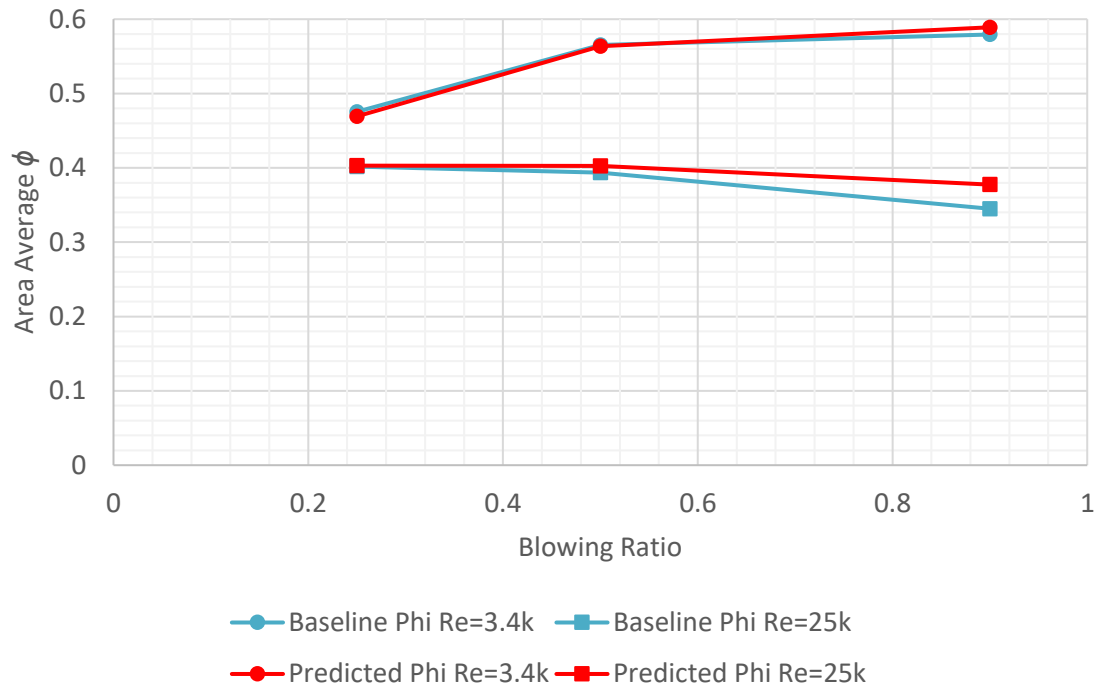


Figure 86: Average predicted ϕ vs. baseline ϕ

VII. Conclusions and Recommendations

7.1 Summary

Three forms of cooling were investigated using experimental and computational methods: internal cooling, hole cooling, and film cooling. The research was intended to discover where each type of cooling was most effective and which cooling was most effective at a variety of test conditions. Previous research has primarily investigated film and impingement cooling. Very little research has been conducted on the role of hole cooling on overall cooling effectiveness.

The experimental tests were conducted in a wind tunnel using a semi cylindrical, scaled up, matched Biot number model. The surface temperature was recorded using an IR camera and nondimensionalized to overall effectiveness. The model was modified in a variety of ways and the resulting overall effectiveness distribution was compared to the nonmodified case. The difference between the two was the $\Delta\phi$ distribution, which showed exactly where the modification made an impact on the overall effectiveness.

An unexpected result was ingestion of the freestream at low blowing ratios. The hot freestream entered the plenum through the cooling holes in the stagnation row or in the front three rows of cooling holes. The hot freestream warmed the cooling hole walls, which then conducted to the outer surface, heating it, and decreasing overall effectiveness. The hot freestream then entered the plenum where it mixed with the coolant, warming it. The mix of coolant and freestream then traveled through the cooling holes. The resulting coolant jet had a higher mass flow rate and was warmer than in a case without ingestion.

The increased mass flow rate caused increased jet lift off. In the case where the jet did stay attached, the film was warmed and therefore less effective.

Impingement resulted in decreased effectiveness in every case. The plates did not impinge on the inside surface between the coolant holes, but rather lined up with the hole inlets. This resulted in decreased heat transfer coefficient on the inner surface. The best plate was the slots and the worst was the two-hole plate.

Blocking rows of cooling holes showed film cooling is highly effective directly downstream the hole exit and hole cooling was slightly effective directly above the inlet and in the general area above the cooling holes. Blocking cooling holes where hot freestream was usually ingested resulted in an overall cooler model because the freestream was not mixing with the coolant, warming it and decreasing effectiveness. It was impossible to separate the effects of film cooling and hole cooling physically, so a computational model was used to further investigate.

The computational model was a flat plate with five overlapping cylindrical cooling holes inclined at 20 degrees to the freestream. Three nonphysical boundary conditions were applied to investigate the relative contribution of each cooling. The bottom of the conducting material was made adiabatic to show the effect of internal cooling. The walls of the cooling holes were made adiabatic to show the effect of the flow cooling the walls. A separate effect of making certain boundaries adiabatic was not only were these walls no longer cooled, the walls no longer warmed the coolant. A one-sided outlet was placed at the exit to the film cooling holes to show the effect of film cooling. The other side of the outlet remained a wall.

The $\Delta\phi$ method was again applied to show the effect on overall effectiveness. Impingement had an only slightly negative effect on $\Delta\phi$ in all cases. Film cooling was the most effective form of cooling at the lower blowing ratios where the cooling jets stayed attached. Adiabatic cooling effectiveness simulations confirmed jet liftoff contributed to decreasing cooling effectiveness at blowing ratios higher than $M=0.5$. The film was only effective past the exit to each cooling hole. At the low Reynolds number condition, the film built up with each hole exit causing increased cooling with increasing x/d . Hole cooling effectiveness increased with increasing blowing ratio. At the highest blowing ratio hole cooling was the dominant cooling type. Hole cooling cooled the top surface by conduction. Close to the hole exits, the material is thinnest and is therefore cooled most effectively by hole cooling. Hole cooling was most effective just before the hole exit where there was another cooling hole overlapping underneath.

Adding roughness to the cooling holes increased overall effectiveness at all blowing ratios. Previous experimental work showed decreasing adiabatic effectiveness with increasing roughness, however these computational results show increased overall effectiveness due to the hole cooling having a significant effect on overall effectiveness.

An equation to predict overall effectiveness, which had previously failed, was shown to give very good average predictions using the film cooling removed case, adiabatic effectiveness, and coolant warming. Cases with smaller adiabatic effectiveness values gave more accurate spanwise predictions, but cases with larger adiabatic effectiveness distributions gave more accurate average values, compared to the baseline case.

7.2 Significance of Research

This is the first research to investigate the importance of hole cooling in overall effectiveness. This research also shows novel experimental results on freestream flow ingestion and the impact of impingement. The computation technique of altering boundary conditions to reflect nonphysical properties to investigate relative effectiveness is brand new. Finally, the computational data showing increased overall effectiveness with increased in-hole roughness and blowing ratio is the first research to show the effect of roughness on overall effectiveness.

This research will inform future leading edge cooling designs and research. Cooler turbine blades with more efficient use of coolant will allow increased performance from gas turbine engines across the Air Force.

7.3 Recommendations for Future Research

This research should be used to inform future research. More research should be conducted on hole cooling and ways to improve it. It was shown increasing in-hole roughness increases the overall cooling effectiveness. This should be further investigated experimentally. Another suggestion for future research is experimentally testing impingement plates with holes that do not line up with film cooling hole inlets, using an identical leading edge model. Another suggestion is creating a conjugate CFD model of the leading edge geometry and performing simulations employing the new CFD techniques. Finally, the predictor method should be tested on different film cooling geometries.

Appendix A: Predicted Overall Effectiveness

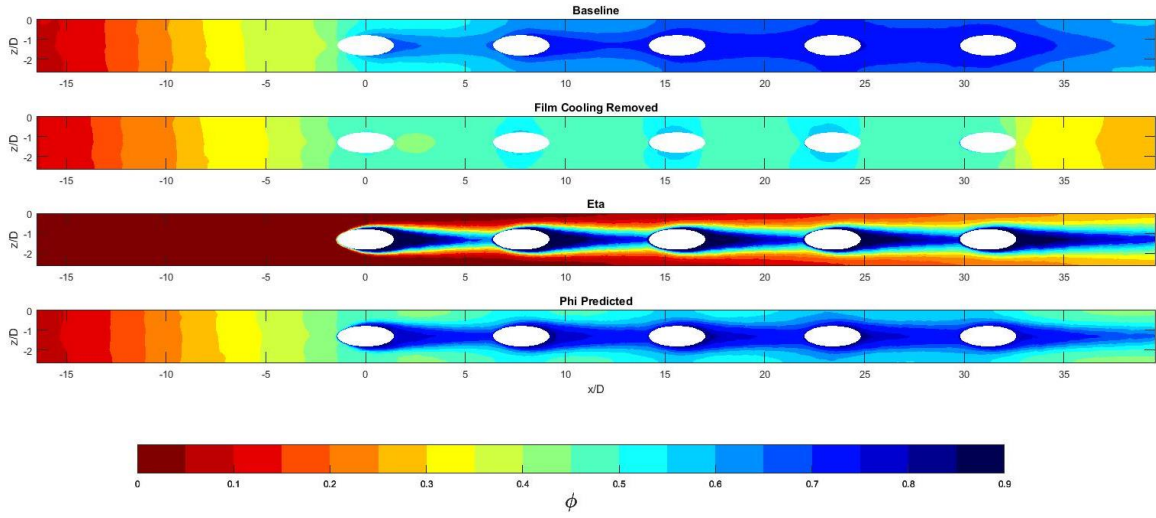


Figure 87: Predicted overall effectiveness for $Re=3,400$, $M=0.5$

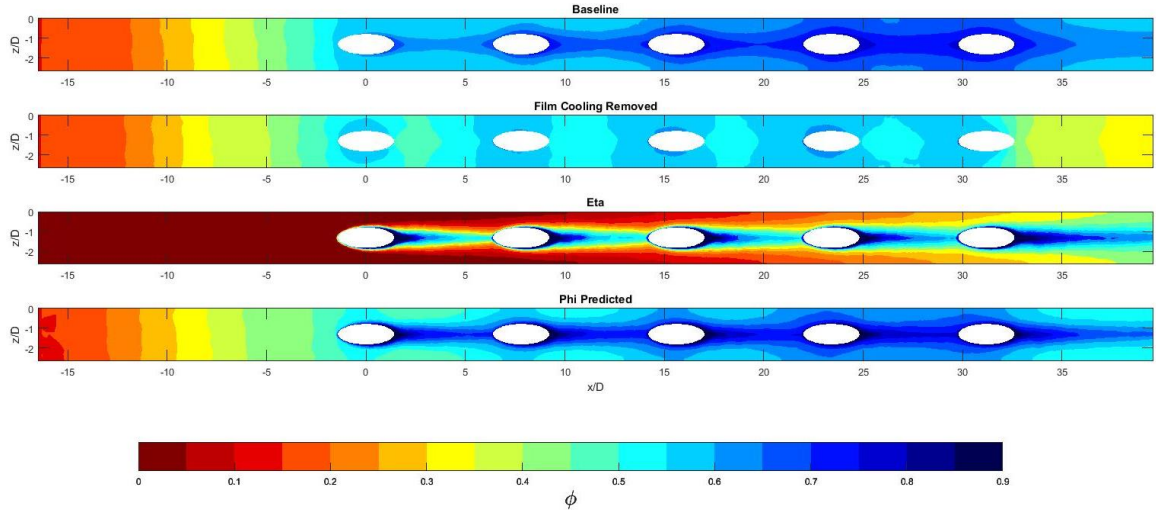


Figure 88: Predicted overall effectiveness for $Re=3,400$, $M=0.9$

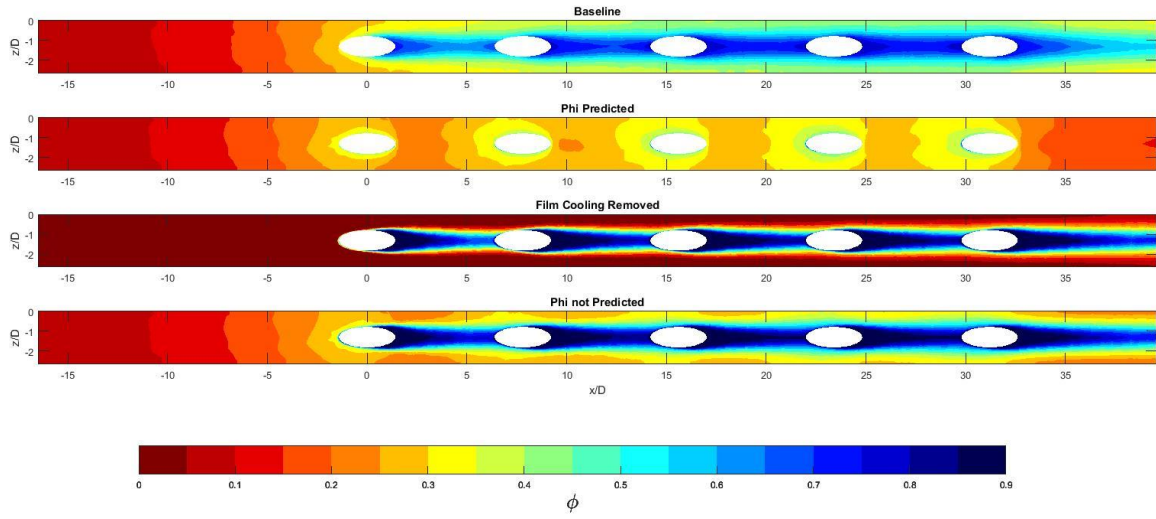


Figure 89: Predicted overall effectiveness for $Re=25,000$, $M=0.25$

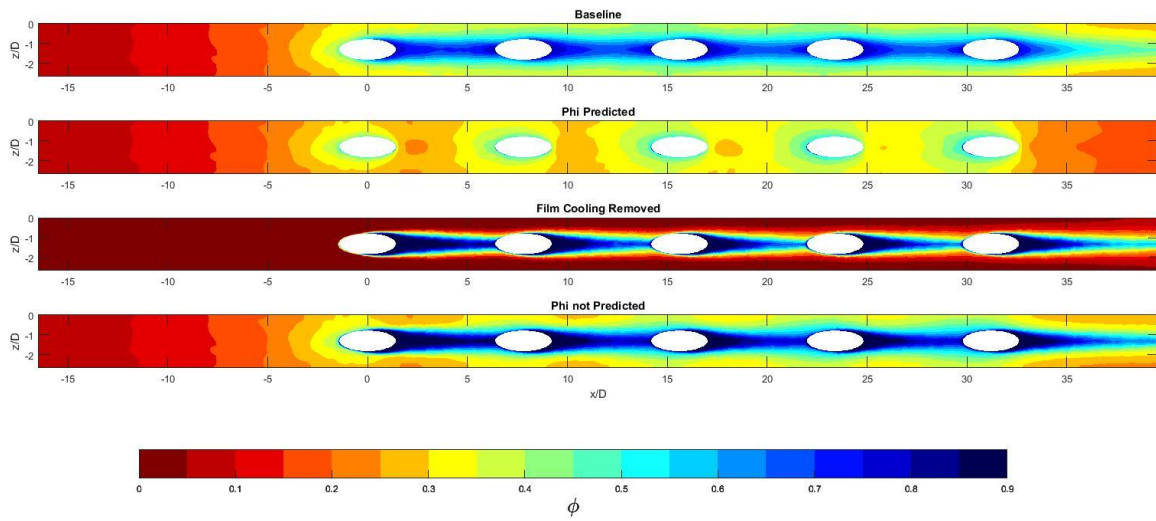


Figure 90: Predicted overall effectiveness for $Re=25,000$, $M=0.5$

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14. ABSTRACT Gas turbine engine hot gas path components are protected via coolant that travels through internal passageways before being ejected as external film cooling. Efforts aimed at improving cooling are often focused on either only the internal cooling or the film cooling; however, the common coolant flow means the internal and external cooling schemes are inextricably linked and the coolant holes themselves provide another convective path for heat transfer to the coolant. The relative influence on the overall effectiveness of internal cooling, external cooling, and convection through the film cooling holes is not well understood. Large scale, matched Biot number experiments and computational fluid dynamics simulations were performed to isolate each cooling component, and thereby determine their relative effectiveness.											
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