SIMULATIONS OF MULTI-PHASE PARTICLE DEPOSITION ON
FILM-COOLED TURBINE SECTIONS

A Dissertation in
Mechanical Engineering

by
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ABSTRACT

The demand for clean, efficient energy has driven the motivation for improving the performance standards for gas turbines. Increasing the combustion temperature is one way to achieve the best possible performance from a gas turbine. One problem associated with increased combustion temperatures is that impurities ingested in the fuel and air become more prone to deposition with an increase in turbine inlet temperature. Deposition on aero-engine and land based turbine components caused by particle ingestion can impair turbine cooling methods and lead to reduced component life. It is necessary to understand the extent to which particle deposition affects turbine cooling in regions where heat transfer from the hot gases to the cooled turbine components is most critical.

For this study, a novel approach was developed to dynamically simulate particle deposition on turbine component models such that the effects of deposition on film-cooling were quantified. To simulate both solid and molten particulate matter in the hot gas path, low melting temperature wax was injected into the mainstream gas path of a low speed wind tunnel used to test film-cooling effectiveness on turbine cascade models. Results showed deposition could reduce film-cooling effectiveness by as much as 30% depending on cooling condition, particle phase, and location of cooling holes. Cooling effectiveness after deposition decreased with an increase in blowing ratio on the airfoil pressure side and leading edge. For all endwall cooling configurations tested, effectiveness after deposition increased with an increase in blowing ratio.

Experiments were also conducted to determine if film-cooling geometries could be modified to mitigate the negative effects of deposition. Various configurations were tested and results showed that embedding cooling rows in transverse trenches could reduce the negative impact of deposition on cooling effectiveness. With the use of trenches, the maximum effectiveness reduction caused by deposition was 15% as compared to 30% with no geometric modification. Deposition was simulated in a film-cooled turbine cascade model with and without endwall contouring. Experimental results showed that strategic placement of cooling holes on a contoured endwall could prevent deposition around film-cooling holes and thus mitigate the negative effects of deposition on cooling-effectiveness.


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NOMENCLATURE

a speed of sound
A_p particle surface area
C chord length
C_{ax} axial chord length
C_D discharge coefficient
C_p specific heat; static pressure coefficient
\Delta h_{fus} specific latent heat of fusion
\Delta P pressure drop
\Delta q_r net heat flux reduction
d film-cooling hole diameter
d_p Rosin-Rammler size constant, where Y_d = 0.368
D cylinder diameter
D_h duct hydraulic diameter
DR density ratio, DR = \rho_c / \rho_{\infty}

E_c Eckert number, E_c = \frac{U_{\infty}}{C_p \left(T_w - T_{\infty}\right)}
h heat transfer coefficient; trench depth
I momentum flux ratio, I = \rho_c U_c^2 / \rho_{\infty} U_{\infty}^2
k thermal conductivity
k_s equivalent sand grain roughness
L film-cooling hole length
L_{\infty} particle travel distance
L_c characteristic length for Stokes number
M blowing ratio, M = \rho_c U_c / \rho_{\infty} U_{\infty}
M_{ideal} inviscid, local blowing ratio, M_{ideal} = \left[\frac{\rho_c \cdot \frac{P_{\infty} - P_{c}}{P_{\infty} - P_{\infty}^{\infty}}}{\rho_{\infty} \cdot \frac{P_{\infty} - P_{c}}{P_{\infty} - P_{\infty}^{\infty}}}\right]^{1/2}
M_{in} blowing ratio based on inlet mainstream velocity, M = \rho_c U_c / \rho_{\infty} U_{\infty}
M_L local blowing ratio, M_L = \rho_c C_D \sqrt{2(p_c - p_{\infty}) / \rho_c} / \rho_{\infty} U_{\infty}
m_L mass loading parameter
m_f continuum fluid mass flowrate
m_p particulate mass flowrate
Ma_{in} Inlet Mach Number, Ma_{in} = \frac{U_{in}}{a}
Ma_{ex} Exit Mach Number, Ma_{ex} = \frac{U_{ex}}{a}
n Rosin-Rammler size distribution parameter
N_c number of cooling holes
p static pressure
P film-cooling hole pitch; airfoil cascade pitch
\( P_{\text{air}} \) atomizing air pressure
\( P_o \) total pressure
\( P_w \) liquid wax pressure
\( Pr \) Prandtl number, \( Pr = \sqrt{\frac{\nu}{\alpha}} \)
\( PR \) pressure ratio
\( Re_d \) coolant jet Reynolds number, \( Re_d = \frac{\rho_c U_c d}{\mu_c} \)
\( Re_D \) leading edge Reynolds number, \( Re_D = \frac{\rho_w U_m D}{\mu_w} \)
\( Re_{in} \) inlet Reynolds number, \( Re_{in} = \frac{\rho_w U_{in} C}{\mu_w} \)
\( s \) circumferential distance from cylinder stagnation
\( S \) nozzle guide vane span
\( St \) Stanton number, \( St = h/pC_p U \)
\( Stk \) Stokes number, \( Stk = \frac{\rho_d d^3 U_p}{18\mu L_c} \)
\( t \) time
\( t_s \) solidification time
\( T_{aw} \) adiabatic wall temperature
\( T_c \) coolant jet temperature
\( T_p \) particle temperature
\( T_{p,i} \) particle initial temperature
\( T_{p,s} \) particle solidification temperature
\( T_w \) wall temperature
\( T_{\infty} \) mainstream temperature
\( TSP \) thermal scaling parameter
\( Tu \) turbulence intensity percent, \( Tu = u_{rms}/U_\infty \)
\( U_c \) coolant jet velocity
\( U_p \) particle velocity
\( U_{\infty} \) mainstream velocity
\( V_p \) particle volume
\( VR \) velocity ratio, \( VR = U_c/U_{\infty} \)
\( W \) duct width
\( x, y, z \) spatial coordinates
\( Y_d \) Rosin-Rammler particle size mass fraction, \( Y_d = e^{-\left(\frac{d_p}{d_{50}}\right)^n} \)

**Greek**
\( \alpha \) film-cooling hole incidence angle; thermal diffusivity
\( \delta \) boundary layer thickness
\( \eta \) adiabatic effectiveness, \( \eta = \frac{\eta_{\text{meas}} - \eta_{\text{corr}}}{1 - \eta_{\text{corr}}} \)
\( \bar{\eta} \) laterally averaged effectiveness
$\bar{\eta}$ area-averaged effectiveness
$\eta_{cl}$ centerline effectiveness
$\eta_{corr}$ effectiveness correction
$\eta_{max}$ maximum effectiveness for a tested film-cooling region
$\eta_{meas}$ measured adiabatic effectiveness, $\eta_{meas} = (T_w - T_{aw})/(T_w - T_c)$
$\bar{\eta}_o$ baseline area-averaged effectiveness (no deposition)

$\theta$ angular location from cylinder stagnation
$\mu$ gas dynamic viscosity
$\nu$ gas kinematic viscosity
$\phi$ overall effectiveness
$\rho_c$ coolant jet density
$\rho_p$ particle density
$\rho_\infty$ mainstream density
$\tau_f$ time characteristic of fluid
$\tau_p$ particle relaxation time
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CHAPTER 1:  
Introduction

A major challenge associated with gas turbine design is developing methods to cool turbine components so that higher combustion temperatures and thus better performance can be achieved. Sophisticated methods must be developed to provide maximum cooling to the turbine while minimize the coolant air demand from the compressor. To make matters worse, particles in the hot gas path can deposit on turbine components and reduce the effectiveness of turbine cooling techniques such as film-cooling. The particulate matter can come from impurities in the fuel or from airborne particulates such as sand or volcanic ash. Following the April 14, 2010 eruption of the Eyjafjallajökull volcano in Iceland, ash clouds over northern Europe caused the largest shut-down of travel in controlled airspace since World War II. Figure 1.1 shows a photograph taken from space of the volcanic ash cloud that put intermittent stops to air travel in northern Europe for weeks [1]. The closure of air travel from the volcanic ash cloud had widespread economic and political effects and brought international attention to the problems associated with particle ingestion in gas turbines.

Ideally, particles in the air and fuel supply would be filtered prior to entering an engine. For aero engine applications, no such filters exist and sand particles on the order of $10 \, \mu \text{m}$ have been found to deposit on turbine components directly downstream of the combustor [11]. The resulting particle deposition can increase surface roughness and in some cases block film-cooling holes on first stage vane and endwall surfaces where cooling is most critical. Figure 1.2 shows a photograph of particle deposition that collected on a first stage vane cascade in an engine after encountering a volcanic ash cloud [2].

Particle deposition is not only a problem in aero engine applications but in land based power generation gas turbines as well. The increasing demand for clean energy on a planet with decreasing natural gas resources has attracted recent attention to developing power generation turbines for operation with alternative fuels. A common alternative fuel is coal derived syngas that contains varying amounts of hydrogen and carbon monoxide. Prior to combustion in a gas turbine, syngas is commonly filtered to strain out any unwanted particulate matter; however, particles as large as $10 \, \mu \text{m}$ still exist [9]. Deposition can increase surface roughness and in some cases block film-cooling holes on first stage vane and
endwall surfaces where cooling is most critical. Figure 1.3 shows an example of an extreme case of deposition that occurred over several days in a power generation gas turbine [3].

In most cases, the region of the airfoil with the highest heat transfer rates from the hot gas path is the stagnation region at the leading edge of the nozzle guide vane [12]. A dense series of cooling holes arranged at different incidence angles around the circumference of the vane leading edge is often utilized to ensure proper cooling of the high heat transfer region. The complex cooling geometry used around the leading edge region is often referred to as the showerhead. A better understanding of the effects of particle deposition in the vicinity of showerhead cooling geometries is required to ensure that cooling designs can withstand the negative effects of deposition.

One method shown to improve film-cooling effectiveness without deposition is the transverse trench method. Studies have shown that recessing a row of film-cooling holes in a transverse trench can improve cooling performance [13, 14]. Whether these improvements exist in severe environments such as when particle deposition occurs is a question yet to be answered.

Secondary flow structures form because of velocity gradients that exist near the endwalls in turbine cascades and are of great concern because they can lead to decreased performance of individual turbine stages and an overall reduction in turbine efficiency. Sophisticated endwall contour designs have been shown to reduce the negative effects of secondary flows [15]; however, it is unknown how endwall contouring affects particle deposition.

Little is known about the effects of particle deposition on turbine cooling because deposition is a complex process involving three phases of flow: solid and molten particulate entrained in the hot gas path through the turbine. The complex flow physics make deposition a challenging process to simulate in a laboratory environment. Film-cooling is a complex phenomenon, involving turbulent mixing between coolant jets and the mainstream crossflow. The direct effects of deposition on film-cooling performance are still unknown. Because full scale engine tests are costly and do not allow for a complete physical understanding of deposition phenomena, it is necessary to develop a method for simulating deposition. By simulating deposition in a laboratory environment, a better understanding of particle interaction with film-cooled surfaces can be acquired. However, meaningful experiments
require matching relevant engine conditions. For this study a novel approach was developed to dynamically simulate particle deposition to determine its effects on film-cooling. Low melt wax was used as the particulate to simulate deposition on turbine component models. Surface temperatures were measured to quantify cooling effectiveness in different regions of a turbine airfoil where cooling is critical.

The following document describes the experiments that were conducted to simulate deposition and determine the effects of deposition on film-cooling in different regions of a turbine airfoil cascade. Chapter 2 provides an in-depth review of the literature outlining studies related to film-cooling, secondary flows, deposition mechanisms, experimental methods used to simulate deposition, and experimental methods used in multiphase flow research. Chapter 2 concludes with a discussion on the hypothesis and uniqueness of the current study.

Chapter 3 describes the parameters of interest to not only simulate deposition but also to scale gas turbine operating conditions. A thermal scaling parameter (TSP) was developed for this study to scale the particle phase change process from engine to laboratory conditions. The TSP is discussed in detail in Chapter 3. The experimental facilities are described in Chapter 4, and the methods used to dynamically simulate deposition on large scale turbine components are described in Chapter 5.

The effects of deposition on film-cooling were determined in three regions of the airfoil where film-cooling is critical. First, Chapter 6 describes the work that was conducted to simulate deposition and quantify the effects of deposition on a flat plate to simulate airfoil pressure side film-cooling. Second, Chapter 7 describes the effects of deposition development, blowing ratio, and particle phase on deposition and its effects on vane leading edge film-cooling. Finally, Chapter 8 describes the effects of deposition development, momentum flux ratio, and particle phase on endwall film-cooling. The endwall region was the main focus of the study where various cooling geometries were tested with and without transverse trenches and with and without endwall contouring. Following the presentation of results in Chapters 6, 7, and 8, an overview of the results and how they compare between different sections of the airfoil are presented in Chapter 9. Conclusions of the study and recommendations for future related work are presented in Chapter 10.
Figure 1.1. Photograph taken from space of the Eyjafjallajökull volcanic ash cloud stretching from Iceland to the United Kingdom in April 2010 [1].

Figure 1.2. Photograph of volcanic ash deposition on aero engine turbine vanes [2].
Figure 1.3. Photograph of deposition on a turbine vane from a natural gas fired power generation turbine [3].
CHAPTER 2:  
Review of Relevant Literature

The following section discusses past work related to deposition in turbomachinery as well as the specific effects of deposition on film-cooling. First, a review of the fundamentals of film-cooling and the relevant work that has been conducted to determine the effects of different operating conditions is presented. Second, the work that has been conducted to mitigate secondary flows in turbine vane cascades using contoured endwalls is discussed. Third, the work involving deposition mechanisms and the effects of deposition on turbomachinery are presented. Fourth, the studies that explored the effects of idealized as well as dynamically simulated deposition on cooling effectiveness are presented. Finally, a review of the relevant work that utilized particle image velocimetry (PIV) and particle tracking velocimetry (PTV) to measure multiphase flows is discussed. The uniqueness of the current study is discussed following the review of relevant literature.

2.1 Fundamentals of Film-Cooling Performance

Film-cooling is a method essential to the life of turbine components that involves the injection of a relatively cool jet into a hot thermal boundary layer crossflow. Film-cooling is typically used on endwall and airfoil surfaces as shown in Figure 2.1. Ideally, the coolant creates a protective film along turbine component surfaces to reduce heat transfer; however, the mixing between the coolant jet and the boundary layer creates highly complex flow that is dependent on many parameters and difficult to predict. Figure 2.2 illustrates a row of film-cooling holes injecting coolant into a hot thermal boundary layer crossflow. Film-cooling performance is highly dependent on many operating and geometric parameters; however, the review in this section will focus on the effects of density ratio (DR), velocity ratio (VR), blowing ratio (M), and momentum flux ratio (I). Recent studies have been conducted that showed potential improvement to cooling by embedding film-cooling rows in shallow transverse trenches. In addition to reviewing the operating parameters listed above, this section will review the effects of embedding film-cooling holes in a transverse trench.

The objective of film-cooling is to reduce heat transfer from the hot surrounding fluid to the turbine component by creating a cool boundary on the surface. The injection of coolant creates a cool layer of fluid on the surface; however, the injection of a jet into a
crossflow can actually increase heat transfer to the turbine component because of the turbulent mixing that takes place during the injection process. Total coolant performance can be quantified by calculating the net heat flux reduction that results from coolant injection. The net heat flux reduction is the amount that heat flux is reduced by adding coolant relative to the heat flux to the surface with no coolant as shown by Equation 2.1

\[
\Delta q_f = \frac{q_o - q_f}{q_o} = 1 - \frac{h_f}{h_o} \left( T_{aw} - T_w \right) = 1 - \frac{h_f}{h_o} \left( 1 - \frac{\eta}{\varphi} \right)
\]  

(2.1)

where \( q_o \) is the heat flux to the surface with no coolant, \( q_f \) is the heat flux to the surface with coolant, and \( h_f/h_o \) is the ratio of the heat transfer coefficients with and without coolant respectively. The \( h_f/h_o \) ratio is often referred to as heat transfer augmentation. The parameter \( \varphi \) is the non-dimensional metal temperature for the operational turbine airfoil and is defined by Equation 2.2

\[
\varphi = \frac{T_w - T_w}{T_w - T_c}
\]  

(2.2)

where \( T_w \) is the mainstream temperature, \( T_w \) is the operational wall temperature, and \( T_c \) is the coolant temperature. Often referred to as the overall effectiveness, \( \varphi \) is generally assumed to be a typical value of 0.6 which is the desired design value for turbine applications [16]. The adiabatic effectiveness, \( \eta \), is the non-dimensional form of the adiabatic wall temperature and is shown in Equation 2.3

\[
\eta = \frac{T_c - T_{aw}}{T_c - T_c}
\]  

(2.3)

where \( T_c \) is the coolant temperature at the exit of a film-cooling hole and \( T_{aw} \) is the adiabatic wall temperature which is the near-wall fluid temperature that drives heat transfer into the wall when coolant is being injected. Adiabatic effectiveness measurements are typically conducted using a film-cooled wall constructed from low thermal conductivity material to create an adiabatic wall condition. Adiabatic wall temperatures are typically measured using either thermocouples or other spatially resolved methods such as IR or liquid crystal thermography. The adiabatic effectiveness is the most common parameter used to quantify film-cooling performance and is used throughout this document.
effectiveness, $\eta_{cl}$, and laterally averaged effectiveness, $\bar{\eta}$, are typically used to evaluate the performance of film-cooling holes.

Many studies have been conducted to determine the independent effects of density ratio, velocity ratio, blowing ratio, and momentum flux ratio on flat plate film-cooling. Pederson et al. [17] conducted an extensive study to determine the effects of DR, VR, M, and I. At low blowing ratios when coolant jets were attached to the wall, they found that laterally averaged effectiveness was independent of DR. At high blowing ratios when the jet was separated, Pederson et al. [17] found that effectiveness increased with increasing DR.

Sinha et al. [18] found that, in general, the effect of DR could not be scaled with VR, M, or I; however, depending on whether the jet is attached or separated, effectiveness scaled with either M or DR. They found that effectiveness increased with an increase in M when coolant jets were attached to the wall and that the detachment-reattachment of the jets scaled with I. For a given I, a decrease in DR resulted in an increase in M resulting in increased $\eta_{cl}$. Sinha et al. [18] also concluded that lateral spreading of the coolant increased with an increase in DR and a decrease in I. In a parallel study, Thole et al. [19] measured mean temperature profiles downstream of a row of film-cooling holes for a range of density ratios between 1.2 and 2.0. They observed the effects of VR, M, and I and determined that the momentum flux ratio was the best scaling parameter for the thermal field of the coolant jets. They determined that coolant jets either remained attached, detached and re-attached, or detached and remained detached from the surface depending on I. When I < 0.4, coolant jets are likely to provide adequate cooling to the surface without separating. At 0.4 < I < 0.8 coolant jets typically separate from the wall and re-attach downstream resulting in decreased cooling performance. At I > 0.8 coolant jets separate from the wall and provide minimal cooling and maximum mixing, which increases heat transfer to the turbine surface [16]. The studies by Pederson et al. [17], Sinha et al. [18], and Thole et al. [19] showed that although cooling performance depends on multiple operating parameters, the most important scaling parameter for film-cooling conditions is momentum flux ratio. The momentum flux ratio and blowing ratio were used to characterize film-cooling operating conditions in the current study.

The leading edge of the inlet guide vane experiences high heat transfer and requires extensive film-cooling with multiple rows of holes surrounding stagnation in the form of a
showerhead. Although the geometry and fluid behavior associated with film-cooling jets is more complicated on the leading edge of the airfoil, the effects of VR, DR, M, and I on cooling effectiveness are similar at the leading edge as compared to a flat plate or an endwall. Polanka et al. [20] were the first to perform highly spatially-resolved adiabatic effectiveness measurements in a showerhead cooling configuration on a nozzle guide vane model. They concluded that the showerhead performed best at high blowing ratios when cooling jets merged with adjacent jets resulting in uniform distribution of coolant around stagnation. Ou and Rivir [21] and Gao and Han [22] experimentally modeled the showerhead region using symmetric cylinder models. Both studies confirmed the conclusions by Polanka et al. [20] that cooling effectiveness improved with an increase in blowing ratio; however, Gao and Han [22] found that effectiveness decreased in the downstream region at high blowing ratios due to high mixing with the mainstream. In the current study, the effects of blowing ratio on deposition and the resulting cooling effectiveness are quantified. The results for the leading edge deposition study are presented in Chapter 7.

Various geometric modifications can be made to film-cooling holes to improve adiabatic effectiveness. One of the most commonly studied geometric modifications is achieved by recessing a row of film-cooling holes into a transverse trench. The concept of embedding coolant holes in a transverse trench was introduced as a method that could be easily manufactured with a slight modification to the thermal barrier coating process [23]. Waye and Bogard [24] measured adiabatic effectiveness for a row of cooling holes embedded in various trench configurations on the suction side of a turbine vane. They tested nine different trench configurations all with a depth of half a hole diameter with varying widths and compared them to a baseline case with no trench. They found that the ideal geometry had a vertical wall at the downstream edge of the cooling row which increased the lateral spreading of the coolant and improved adiabatic effectiveness by as much as 100% at the hole trailing edge and as much as 40% downstream of the holes.

Adiabatic effectiveness and heat transfer augmentation for a row of cylindrical holes embedded in a transverse trench with a depth of one hole diameter were measured by Harrison et al. [13]. The tests showed that net heat flux reduction was significantly higher for the holes embedded in the trench compared to a row of holes with no trench. Harrison et al. [13] measured net heat flux reduction at M = 0.6, M = 1.0, and M = 1.4. With no trench,
they measured negative net heat flux reduction at $M = 1.0$ and $M = 1.4$ meaning that the addition of coolant actually increased heat transfer from the hot mainstream to the turbine. With the trench installed, there was a decrease in heat transfer with an increase in $M$. They concluded that the trench prevented jet separation and improved lateral spreading of the coolant.

Sundaram and Thole [14] measured adiabatic effectiveness for a row of film-cooling holes embedded in a narrow transverse trench. They tested a row of cooling holes in trenches at depths of 0.4, 0.8, and 1.2 hole diameters with a vertical wall at the hole trailing edge. The cooling row was located on the endwall near the leading edge stagnation region of the vane. Sundaram and Thole [14] found that the trench with a depth of 0.8 hole diameters produced the highest adiabatic effectiveness levels. They also determined that the sensitivity of adiabatic effectiveness to blowing ratio was dependent on trench depth. At 0.4$d$, effectiveness increased with increasing $M$ until it reached a peak of $M = 2.5$. At 0.8$d$, effectiveness increased with an increase in $M$ and leveled off between $M = 2.5$ and $M = 3.0$. At 1.2$d$, effectiveness increased with increasing $M$ with the greatest increase occurring between $M = 2.5$ and $M = 3.0$. Sundaram and Thole [14] and Harrison et al. [13] found that the trench allows for increased coolant flow without the increased risk of jet separation at high blowing ratios.

Sundaram and Thole [5] made flowfield measurements along the leading edge of a stator vane. They used laser Doppler velocimetry to measure the flowfield in the leading edge region for a case with no cooling, with film-cooling, and with film-cooling holes embedded in a transverse trench. Figure 2.3 illustrates the comparison of leading edge flowfields for the three cases tested. For the film-cooling cases in Figures 2.3b and 2.3c, the blowing ratio was set to $M = 2.5$ which led to separation for the normal film-cooling case but not for the trench case which supports the findings of Harrison et al. [13] and Sundaram and Thole [14]. Sundaram and Thole [5] found that injecting film-coolant at $M = 2.5$ (Figure 2.3b) led to the formation of “dual vortices” located upstream and downstream of the cooling holes. By embedding cooling holes in the transverse trench, the coolant remained attached to the surface, as shown in Figure 2.3c, and eliminated the formation of the downstream vortex. Although the upstream vortex still existed when film-cooling holes were embedded in the trench, the vorticity was lower than it was for the horseshoe vortex in the baseline case. This
means that use of the transverse trench not only improved adiabatic effectiveness of the coolant but also mitigated the formation of the leading edge “horseshoe” vortex. The “horseshoe” vortex and secondary flows associated with it are discussed in Section 2.2.

2.2 Effects of Endwall Contouring on Secondary Flows

Viscous flow effects where airfoils mate with endwalls in a turbine cascade result in highly three dimensional flow structures referred to as secondary flows. Secondary flow structures increase the total pressure loss through a turbine stage and ultimately lead to reduced turbine efficiency. Secondary flows also enhance mixing with the hot mainstream gases increasing heat transfer to turbine components. Figure 2.4 illustrates the secondary flow structures that exist near the vane endwall junction. These structures are created by the formation of a “horseshoe” vortex at the leading edge stagnation line of each turbine airfoil in a cascade. The horseshoe vortex is formed when the endwall boundary layer approaches the stagnation line at the leading edge of the airfoil. The velocity gradient in the boundary layer creates a static pressure gradient along the stagnation span. The static pressure gradient drives the flow toward the endwall forming a vortex as illustrated in Figure 2.3a. The horseshoe vortex then divides into a passage vortex which propagates through the vane passage on the pressure side and a counter vortex that wraps around the suction side of the vane. The counter vortex and the passage vortex from the neighboring vane then merge to form a counter-rotating vortex pair. In addition to decreased aerodynamic performance and increased heat transfer, these highly complex secondary flows create a non-ideal situation for endwall film-cooling performance.

A sophisticated method used to mitigate secondary flow effects is endwall contouring. By modifying the endwall profile, the local pressure distribution along the endwall can be manipulated to reduce the scale of secondary flow structures. Endwall contouring can be classified as either axisymmetric or non-axisymmetric. Axisymmetric contours are symmetric about one axis, typically the streamwise. Non-axisymmetric contours are not symmetric about any axis and are sometimes referred to as three-dimensional contours.

Deich et al. [25], Ewen et al. [26], and Morris and Hoare [27] were among the first to study the effects of axisymmetric endwall contouring. Deich et al. [25] and Ewen et al. [26]
reported that turbine efficiency improvements of up to 3.5% can be achieved with the use of “endwall profiling”. Morris and Hoare [27] were the first to test the effectiveness of endwall contouring at different inlet flow conditions. They tested three different contour designs with various inlet boundary layer thicknesses and compared results with a flat endwall baseline case. They tested axisymmetric and non-axisymmetric endwall profiles. One profile actually increased losses and they concluded that great care must be taken in designing a contoured endwall to ensure a reduction rather than an increase in secondary losses.

Harvey et al. [28] performed computational tests on multiple non-axisymmetric endwall contour geometries to determine an optimal design to reduce secondary flows. Hartland et al. [29] conducted experiments on the geometries designed by the computational work by Harvey et al. [28] and observed a 30% reduction in secondary flow losses by adding the contoured 27. Ingram et al. [30] conducted an experimental and computational study to compare secondary loss results. Similar to the findings by Hartland et al. [29], Ingram et al. [30] found that the computational tests under-predicted the reduction in secondary losses achieved by adding a non-axisymmetric endwall contour. The findings by Hartland et al. [29] and Ingram et al. [30] show that further development of computational methods is necessary to obtain the degree of accuracy of experimental research in secondary flows.

Praisner et al. [7], Knezevici et al. [31], Lynch et al. [15], and Lynch et al. [10] all studied the effects of non-axisymmetric endwall contouring on secondary flows using a Pack-B airfoil. The Pack-B is a Pratt & Whitney airfoil design that has been extensively used in low pressure turbine research. Praisner et al. [7] used a computational design technique to develop a non-axisymmetric contoured endwall for the Pack-B passages as shown in Figure 2.5. The contour designed by Praisner et al. [7] was effective at reducing losses and turbulent kinetic energy as compared to the flat endwall with measured and predicted reductions of 10% and 4% respectively. The contour designed by Praisner et al. [7] was used later by Knezevici et al. [31] who performed pneumatic probe pressure measurements and oil flow visualization. They determined that the contour reduced losses and caused a reduction in secondary kinetic energy as compared to the flat endwall. Lynch et al. [15] made measurements of heat transfer and performed oil flow visualization using the Pack-B with and without endwall contouring. They concluded that the contour reduced heat transfer by up to 20% in regions of high heat transfer.
Lynch et al. [10] placed a row of film-cooling holes in the region of high heat transfer in the Pack-B passage with and without endwall contouring. They compared measured and computationally predicted values of adiabatic effectiveness in the passage at film-cooling blowing ratios of 1.0 and 2.0. Although laterally averaged effectiveness was over-predicted by the SST k-ω model, effectiveness simulations replicated the trends observed by experiments. Measurements and predictions showed that the endwall contour reduced coolant spreading across the endwall. They attributed the reduction in spreading to the reduction of cross passage flow with the contoured endwall. Although the contoured endwall effectively reduced the secondary losses, it actually had a negative effect on film-cooling.

The studies presented in Section 2.2 represent a sample of the work performed to mitigate secondary flow effects. Even with the vast amount of research conducted related to secondary flows, there has been no study performed to determine the effects of secondary flows on deposition. The film-cooled flat and contoured endwalls used by Lynch et al. [10] were used in the current study to determine if the contour can mitigate the negative effects of deposition on cooling. The results from the deposition study with and without endwall contouring are presented in Section 8.3.

2.3 Deposition Mechanisms and Effects of Deposition on Turbine Cooling

Following the oil embargo of 1973, research interests were focused on developing alternative fuels for gas turbines. The foundation was laid for research related to deposition, erosion, and corrosion in turbomachinery between the early 1970s and the early 1990s before the focus shifted toward optimizing natural gas fueled plants. Research conducted during this time period explored the dominant mechanisms of particle delivery to turbine surfaces that include inertial impaction, turbulent diffusion/eddy impaction, Brownian diffusion, and thermophoresis [2].

Numerical calculations by Rosner and Fernandez de la mora [32] and Gökuğlu and Rosner [33] showed that thermophoresis enhances transport of small particles through nonisothermal turbulent boundary layers to deposit on cooled surfaces. Slater et al. [34] performed a computational study to evaluate the independent and combined effects of diffusion, inertia, and thermophoresis on deposition fraction in a turbine cascade. They determined that the independent effects of diffusion and inertia were small amounting to
deposition fractions of 0.08% and 2.05% respectively; however, the combined effects of diffusion and inertia amounted to a deposition fraction of 7.33%. The combined effects of deposition, inertia, and thermophoresis amounted to a deposition fraction of 8.95%. Slater et al. [34] concluded that thermophoresis produced only a modest increase in deposition due to the dominant effect of streamline curvature and only had an effect on deposition rates for particles with small inertial response times. The introduction of film-cooling would increase the influence of streamline curvature on deposition and thus make inertial deposition mechanisms even more dominant over small particle deposition mechanisms such as diffusion and thermophoresis.

Dring et al. [35] conducted an analytical study to determine the nature of particle trajectories in turbine airfoils based on applied drag forces alone. They found that Stokes number, Stk, was the most important dimensionless parameter that scaled a particle’s trajectory. Dring et al. [35] determined that particles having Stokes numbers less than 0.1 follow fluid streamlines, while particles with Stokes numbers greater than 10 impact on airfoil surfaces driven by their own inertia. In an earlier study unrelated to gas turbine applications, Friedlander and Johnstone [36] proposed that particles with Stk << 1 deposit because of eddy impaction and turbulent diffusion in near wall regions of the surrounding flow. Stokes number is defined and discussed in detail in Section 3.3.

Multiple studies have been conducted to explore the effects of gas temperatures and surface temperatures on particle deposition. Wenglarz and Fox [37], Walsh et al. [38], Richards et al. [39], and Wenglarz and Wright [40] conducted studies relating deposition sticking mechanisms to gas and surface temperatures.

Wenglarz and Fox [35] conducted deposition, erosion, and corrosion experiments in a small scale turbine combustor. They observed that deposition rates increased with an increase in gas temperature. They attributed these increased deposition rates to the increased fraction of molten particulate that existed at elevated gas temperatures. Walsh et al. [38] studied the deposition of bituminous coal ash on a heat exchanger tube and observed the sticking process over time. They found that, initially, only molten particles would stick to the clean surface solidifying soon after deposition. As the thickness of the deposition built up, the temperature of the deposition surface exposed to the hot gas path became high enough that it would remain sticky allowing not only molten particles to deposit but solid particles to
deposit as well. Walsh et al. [38] found that, eventually, the rate that incoming particles deposited on the surface equaled the rate that particle shedding would occur because of erosion.

Richards et al. [39] studied the deposition of bituminous coal ash in a reactor specifically designed to measure deposition at different gas and surface temperatures. They determined that a decrease in gas temperature corresponded to an increase in ash particle size, which resulted in an increase in sticking fraction. Richards et al. [39] also determined that increased surface cooling resulted in a decrease in sticking fraction when particles were small enough to be sufficiently cooled by the thermal boundary layer prior to impaction. Although large deposits made a greater contribution to sticking fraction, they were more easily removed from the surface than the small deposits. In their study evaluating alternative fuels for gas turbines, Wenglarz and Wright [40] concluded that the gas temperature relative to the melting temperature of an entrained particulate was the relationship that best determined the sticking probability of particles that deposited because of inertial impaction. If a particle is in molten form when it impacts a surface, it is more likely to stick to that surface than a particle that is in solid form.

2.4 Experimental Methods for Simulating Deposition

Shifting to alternative fuels in gas turbines has led to numerous studies related to the effects of deposition on turbine cooling effectiveness. These studies can be categorized based on the methods used to mimic surface deposition. The first method used to mimic surface deposition was to condition an experimental model with deposits based on surface measurements from actual turbine hardware. Cardwell et al. [41] and Sundaram et al. [42] conducted studies related to the effects of roughness on airfoil platform cooling relevant to gas turbines for which roughness simulations were based on turbine hardware measurements made by Bons et al. [8].

From the surface roughness measurements on actual engine turbine components made by Bons et al. [8], they generated detailed three-dimensional maps of in-service turbine hardware. They found that rather than being made up by high-angled formations, the surfaces were composed of random combinations of peaks, valleys, and plateaus caused by deposition. Regions near film-cooling holes showed great variation in surface roughness as
shown in Figure 2.6. They observed a periodic distribution of furrows ranging from 80 \( \mu \)m at five hole diameters downstream to 500 \( \mu \)m at 15 hole diameters downstream. The existence of these furrows suggests film-cooling jets prevent deposition in the regions immediately downstream of cooling holes.

Cardwell et al. [41] simulated roughness on a film-cooled endwall using sandpaper in a large scale turbine airfoil cascade to model the roughness measured by Bons et al. [8]. The endwall root-mean-square (RMS) roughness height measured by Bons et al. [8] was 28 \( \mu \)m which corresponded to a sand grain roughness of 227 \( \mu \)m in the large scale tunnel used by Cardwell et al. [41]. The equivalent sand grain roughness, \( k_s \), is the roughness height that would yield the same skin friction for a sand grain surface as the surface in question. By measuring adiabatic effectiveness at various blowing ratios, Cardwell et al. [41] determined that roughness caused a decrease in cooling effectiveness at high blowing ratios; however, at low blowing ratios roughness had little-to-no effect on cooling. They attributed the decreased effectiveness at the high blowing ratios to increased coolant jet separation caused by a roughness induced thicker boundary layer.

An extensive study was conducted by Sundaram et al. [42] to determine the effects of deposition on the endwall film-cooling near the leading edge and the pressure side of a nozzle guide vane. Similar to Cardwell et al. [41], Sundaram et al. [42] modeled roughness in a uniform manner using sandpaper. In addition to the sandpaper, Sundaram et al. [42] manufactured deposits of ideal two-dimensional shapes and sizes based on measurements made by Bons et al. [8]. Sundaram et al. [42] placed idealized deposits of varying heights in different locations relative to leading edge and pressure side endwall cooling holes. They found that when placed downstream of leading edge cooling holes, small deposits actually enhanced cooling effectiveness by 25% for a deposit height of 0.5d and a blowing ratio of 1.5. When placed in the vicinity of pressure side cooling holes, deposits lowered effectiveness between cooling holes, but enhanced effectiveness along the vane endwall junction.

Somawardhana and Bogard [43] conducted a study to determine the effects of varying surface roughness and near-hole obstructions on adiabatic effectiveness. Similar to the methods used by Sundaram et al. [42], Somawardhana and Bogard [43] placed idealized obstructions of different shapes and sizes upstream and downstream of film-cooling holes to
simulate deposits caused by large agglomerations of particles that could deposit randomly. They found that roughness caused as much as a 30% reduction in adiabatic effectiveness at low blowing ratios and a 30% improvement in adiabatic effectiveness at high blowing ratios. The studies by Sundaram et al. [42] and Somawardhana and Bogard [43] showed that obstructions placed upstream of film-cooling holes reduced adiabatic effectiveness while obstructions placed downstream of cooling holes actually improved effectiveness.

The effects of idealized roughness and deposition on the cooling effectiveness with and without a transverse trench were studied by Somawardhana and Bogard [44]. Their results showed embedding cooling holes in a transverse trench improved cooling effectiveness and eliminated the negative effects of roughness and deposition. Sundaram and Thole [14] conducted a study to determine the effects of deposition near a row of film-cooling holes embedded in a transverse trench. They placed “bumps” with varying heights downstream of a cooling row embedded in a trench. They observed no major effect on effectiveness for bumps with heights of 0.5d and 0.8d; however, the bump with a height of 1.2d enhanced effectiveness by approximately 20%. Studies by Sundaram et al. [42], Somawardhana and Bogard [43], and Sundaram and Thole [42] show promise that roughness and deposition might actually improve cooling effectiveness in special cases.

Another method used to mimic surface deposition is to dynamically simulate the deposition process in a laboratory environment. Jensen et al. [45], Bons et al. [9], Wammack et al. [46], Bons et al. [47], Crosby et al. [48], and Ai et al. [3] conducted studies in a Turbine Accelerated Deposition Facility (TADF) designed to simulate deposition on a test coupon in a laboratory environment. These studies were conducted specifically to simulate particle deposition that would result from the combustion of various alternative fuels in a land based gas turbine.

Jensen et al. [45] operated the Turbine Accelerated Deposition Facility (TADF) that was designed to simulate deposition in an accelerated manner. By increasing the concentration of particulate matter in the hot gas path they could simulate 10,000 hours of turbine operation in a four hour test. The deposition simulation was carried out by feeding particles into a natural gas combustion process. The particle-laden exhaust gases, which simulated gas temperatures and velocities that exist in modern first-stage turbines, were then impinged onto a removable coupon that could be analyzed following testing. The simulation
methodology was validated using scanning electron microscope and x-ray spectroscopy analyses, which showed that deposit microstructures and chemical compositions were similar to deposits found on actual turbine hardware.

The TADF was also used by Bons et al. [9] to analyze deposition that resulted from inertial impaction of materials found in four alternative synfuels. They found that injecting coal and petroleum coke (pet coke) ash resulted in larger deposits than biomass deposits produced by sawdust ash and straw ash. Similar to the findings of Richards et al. [39], Bons et al. [9] found that large coal and pet coke deposits were easily separated from the surface while the small biomass deposits were more tenacious.

The evolution of deposition over time was studied by Wammack et al. [46] in the TADF. Their test procedure consisted of injecting particulate in different phases referred to as “burns”. They measured surface topographies after every burn cycle to observe how the deposition evolved over time. They found that roughness increased substantially as a result of the first two burn cycles. Following the initial increase, they observed a delay in the deposit growth through the third burn cycle. The delay was followed by a further increase in roughness after the fourth burn cycle. They suggested that the type of evolutionary process described above could lead to varied heat load over time.

Bons et al. [47] also studied the evolution of deposition over time in the TADF by injecting particulate matter in different burn cycles. They measured the surface topography after every burn cycle and recreated roughness models out of acrylic. Heat transfer coefficients were measured on each model to determine how deposition affected heat transfer as it evolved. They observed a 15% increase in heat transfer coefficients after the first burn cycle and an additional 9% increase after the second burn cycle. Heat transfer then leveled off through the third burn cycle before increasing an additional 3% after the fourth burn cycle. Through four burn cycles, the heat transfer coefficients increased by a total of 27% relative to the smooth baseline. Crosby et al. [48] studied the independent effects of particle size, gas temperature, and surface temperature on deposition in the TADF. They found that deposition rate increased with an increase in particle size, an increase in gas temperature, and an increase in surface temperature.

The TADF was used by Ai et al. [3] to investigate the effects of deposition on a film-cooled surface. They designed a new test coupon holder that could provide coolant to a
coupon with various film-cooling configurations. The film-cooled coupon was set up so that particle laden exhaust flow impacted the surface at a 45° angle. In addition to measuring surface topographies, they measured surface temperatures using an RGB camera and were able to conclude that increased deposit height resulted in increased surface temperatures. By observing deposition behavior over time they concluded that increased surface temperatures accelerated deposition resulting in a non-linear deposit growth rate with time. They also observed the effects of blowing ratio for film-cooling on deposition growth and surface temperature. They found that increased blowing ratio resulted in decreased surface temperatures and therefore reduced deposition in cooling hole wakes.

Ai et al. [49] used the TADF to observe deposition and its effects on trenched film-cooling holes. They tested the trench configuration with deposition at various flow impingement angles. Without deposition, the trench improved cooling at shallow impingement angles; however, during deposition simulation, particles accumulated inside the downstream lip of the trench which led to hole blockage and reduced cooling. Although the tests by Ai et al. [3] and Ai et al. [49] simulated high temperature deposition mechanisms, the experiment lacked proper simulation of film-cooling conditions. Their measurement techniques were also limited because of the testing environment and provided little insight into the particle interaction with the film-cooled surface.

Smith et al. [50] used a similar accelerated technique to those in the TADF to simulate deposition in a true scale nozzle guide vane cascade using the Turbine Reacting Flow Rig (TuRFR). They found that mainstream temperature and film-cooling operating conditions had a significant effect on deposition, observing that surface deposition increased significantly with an increase in mainstream temperature. This finding supports the other results in the literature that show deposition is highly dependent on the relationship between particle melting temperature and mainstream gas temperature [37-40].

Few studies have been conducted to simulate deposition on a vane leading edge. Sreedharan and Tafti [51] performed a computational study to model deposition on a showerhead film-cooling geometry. They modeled the deposition and erosive behavior of 5 µm and 7 µm ash particles. They assumed that particles above their softening temperature would stick to the surface upon impaction and particles below their softening temperature would not stick. They found that showerhead cooling holes were effective at preventing
deposition, because they effectively solidified particles and diverted their trajectories. Results showed that an increase in blowing ratio increased deposition of 5 µm particles by 4% but decreased deposition of 7 µm by 5%. Albert et al. [52] used wax particle injection to experimentally simulate deposition on a showerhead film-cooling geometry. They found that deposition thickness increased with time until reaching a quasi-steady state. Albert et al. [52] concluded that blowing ratio and the relationship between gas temperature and particle solidification temperature had large effects on deposition. Although Sreedharan and Tafti [51] showed that coolant jets can reduce deposition, Albert et al. [52] concluded that secondary vortices associated with coolant jets can actually enhance particle transport to some regions of the surface.

2.5 Experimental Methods in Multiphase Flow Research

Particle image velocimetry (PIV) and particle tracking velocimetry (PTV) are techniques that have been developed to characterize flows and track particles in multiphase flows in a non-intrusive manner. The origins of PIV and PTV can be traced back to early flow visualization techniques used by Ludwig Prandtl in 1904. Prandtl suspended mica particles in a water tunnel which could be used to qualitatively visualize steady and unsteady flow patterns around simple two-dimensional obstacles [53]. Today, PIV and PTV have evolved into sophisticated digital measurement techniques that utilize double pulsed lasers and high speed cameras to quantify 2-D and 3-D flows accurately in fractions of a second. This section provides some specific examples of how PIV has been used to characterize film-cooling flows and how PTV has been used to characterize particle paths in multiphase flows.

In recent years, it has become possible to conduct studies utilizing 2-D and 3-D PIV to characterize the complex flows associated with film-cooling. Bernsdorf et al. [54] conducted a 3-D PIV study to measure the flowfield in the vicinity of a film-cooling hole for the calibration and validation of a 3-D CFD model. They tested multiple blowing ratios for film-cooling geometries having angles of 30° and 50° relative to the mainstream crossflow. Similar to Thole et al. [19], Bernsdorf et al. [54] concluded that the centerline trajectory of the coolant scaled with coolant jet momentum flux ratio. The 3-D PIV also allowed for the visualization of the complex flow physics. Jessen et al. [55] conducted a 2-D PIV study to determine the effects of density ratio and blowing ratio on flow characteristics and to validate
numerical findings. They found that increased blowing ratio enlarged the size of the recirculation region downstream of the coolant jet. In agreement with the findings by Sinha et al. [18], Jessen et al. [55] concluded that lateral spreading of the coolant increased with an increase in density ratio. Renze et al. [56] performed numerical calculations predicting film-cooling behavior using large-eddy simulation (LES) and compared the results with experimental PIV results. They found that the PIV results were in excellent agreement with the LES results.

The fundamental difference between PIV and PTV depends on the relationship between the distance a particle travels between two successive image acquisitions and the smallest spatial scale of the velocity field structure. If the travel distance of a particle between successive acquisitions is smaller than the spatial scale of the velocity field then PIV is performed by cross-correlation between successive images. If the travel distance of the particle is larger than the spatial scale of the velocity field then velocities can be determined by tracking specific particle trajectories through successive images. PTV can be used to obtain these particle trajectories and measure velocities using a Lagrangian frame of reference. Cenedese et al. [57] utilized PTV to track pollutant dispersion in a lab simulation of an atmospheric convective boundary layer. They concluded that PTV could be used to determine the turbulent time scales associated with hot updrafts and cold downdrafts. Romano [58] measured particle trajectories and Lagrangian statistics using PTV at the outlet of a circular jet. With the use of PTV, Romano [58] was able to determine how particles from different regions of the jet dispersed. Cardwell et al. [59] developed a time-resolved PTV method to identify the motion of foreign ingested particles within ribbed channels used for internal cooling of turbine components. Using PTV they were able to observe where particles impacted ribbed surfaces and how rib impactions affected particle trajectories. They found that large agglomerations of particles either broke up into groups of small particles or rebounded and travelled opposite to the flow direction following rib impaction.

The use of sophisticated flow measurement methods can reveal a lot about particle interaction with complex flow structures. Although PIV was not conducted in the current study, a time resolved digital PIV system was used to conduct flow visualization studies to qualitatively observe how particles interact with endwall film-cooling jets. The flow visualization technique is explained in detail in Section 5.3.
2.6 Hypothesis and Uniqueness of Current Study

The literature reviewed in Sections 2.1 through 2.5 covered the fundamentals of film-cooling performance and secondary flows as well as the methods used to improve film-cooling and reduce losses caused by secondary flows. The studies described above revealed much about the deposition process as well as different techniques that have been used to simulate and quantify the effects of deposition and resulting cooling effectiveness on the airfoil surface and on the endwall. The literature showed that embedding cooling holes in transverse trenches can improve cooling effectiveness; however, the effects of deposition on trenched hole cooling effectiveness have never been quantified. The literature also showed that endwall contouring can be used to reduce secondary flow losses; however, the effects of deposition on a contoured endwall with film-cooling have never been quantified. The hypothesis for the current study is that deposition induced roughness decreases cooling effectiveness, but the negative effects of deposition can be mitigated with the use of geometric modifications such as transverse trenches and endwall contouring.

There has never been a study versatile enough to dynamically simulate deposition while closely observing the deposition in the vicinity of film-cooling holes in a large scale laboratory environment. The major objective of the current study was to develop a method to dynamically simulate particle deposition and determine the effects of deposition on pressure side, leading edge, and endwall film-cooling. These three regions were modeled using a flat plate test section, a symmetric cylinder airfoil model, and sophisticated airfoil cascade test sections respectively. The study utilized digital image processing with high speed flow visualization of particle interaction with film-cooling jets as well as IR thermography to gain a physical understanding of the effects of simulated deposition on pressure side cooling, showerhead cooling, endwall cooling with and without trenches, and endwall cooling with and without contours.

Although a similar wax injection method was used by Albert et al. [52] to simulate deposition on a showerhead, the emphasis of their study was to determine the effects of deposition on local overall effectiveness rather than surface cooling effectiveness. For the current study, highly detailed surface temperatures were measured to quantify the effects of deposition development, blowing ratio, and particle phase on cooling effectiveness applicable to different regions in a turbine airfoil cascade.
Figure 2.1. Film-cooled high pressure turbine stage [4].

Figure 2.2. Film-cooling parameters for a row of cylindrical cooling holes.
Figure 2.3. Leading edge flowfield superimposed with streamwise velocity for (a) baseline, (b) film-cooling holes without a trench at $M = 2.5$, and (c) film-cooling holes with a trench at $M = 2.5$ [5].
Figure 2.4 Secondary flow structures for a turbine airfoil passage [6].

Figure 2.5. Non-axisymmetric contoured endwall designed by Praisner et al. [7] and used in the current study for deposition experiments.
Figure 2.6. Surface deposition downstream of film-cooling holes measured by Bons et al. [8].
CHAPTER 3: Scaling Turbine Conditions to Laboratory Conditions

A method was developed to realistically model the physics of film-cooling and particle deposition in a low speed wind tunnel operating near standard temperature and pressure (STP). Gas turbines operate at turbine inlet temperatures exceeding 1500 K and velocities on the order of 100 m/s. To gain a physics-based understanding of the fluid dynamics involved in film-cooling and particle deposition, experiments were conducted in a low speed laboratory with velocities on the order of 10 m/s and temperatures on the order of 330 K. This chapter is devoted to discussing the parameters used to scale the relevant gas turbine conditions in the experimental facilities.

3.1 Mainstream Scaling Parameters

The parameters of importance for scaling mainstream flow conditions in turbine cascades are Reynolds number (Re), Mach number (Ma), Eckert number (Ec), Prandtl number (Pr), and turbulence intensity (Tu). This section describes each of these parameters for use in scaling mainstream flow conditions between engine and laboratory settings. The relevant mainstream scaling parameters discussed in this section are presented for each experimental facility described in Chapter 4.

The mainstream flow regime through a turbine cascade can be scaled using the Reynolds number which is dependent on the fluid velocity, fluid kinematic viscosity, and airfoil length scale. The most appropriate length scale and that which is used most commonly in airfoil applications is the chord length. The experimental models used in the current study were scaled up to account for the lower velocities in the wind tunnel relative to an engine. For this study, turbine operating conditions and geometric specifications were provided by industry [10, 11, 60].

The Mach number is another parameter typically used to characterize flow through a turbine cascade. The Mach number is the ratio of fluid velocity to the speed of sound through the fluid and is the parameter used to scale the compressibility effects of the flow. In turbine applications, flow is usually characterized by Mach number defined by the conditions at the inlet and the exit of an airfoil row. Mach number conditions are typically not matched in large scale wind tunnel cascades; however, it has been shown that local heat transfer
coefficients on a turbine vane are sensitive to Reynolds number and have a weak dependence on Mach number [61]. In a computational study to determine the effects of Mach number on secondary flows, Hermanson and Thole [62] found that flow patterns for high and low Mach number cases were similar and that low speed wind tunnel experiments were applicable.

The Eckert number, which is derived from the non-dimensional form of the energy equation, scales the temperature increase through adiabatic compression and is dependent on Mach number, freestream-to-wall temperature ratio, and specific heat ratio. The Eckert number and thus the frictional heat are only significant when the Mach number is high such as near the throat of an airfoil passage. In the leading edge region of the cascade, however, the Eckert number is not significant. Most of the experiments in the current study focused on endwall and airfoil regions near the leading edge.

Also derived from the energy equation is the Prandtl number which is the ratio of momentum to thermal diffusivity. The mainstream fluid in the current study is air which has a Prandtl number around 0.7 and is relatively insensitive to temperature and can therefore be accurately matched between engine and laboratory conditions.

Turbulence intensity is also an important parameter particularly when modeling deposition and film-cooling for turbine applications. The turbulence intensity is the ratio of the root-mean-square (RMS) of the fluctuating velocity to the axial mean component of the mainstream velocity, $U_\infty$. Measurements as low as 6% have been made by Goebel et al. [63] in engine conditions while measurements as high as 35% have been made by Goldstein et al. [64]. For the current study, turbulence intensity was relatively low around 4%. Although the 4% turbulence intensity is on the low end of the scale observed in actual engines, Radomsky and Thole [65] showed that turbulence intensity does not have a significant effect on secondary flows.

As discussed in Chapter 2, secondary flow behavior is a parameter of great importance to film-cooling as well as particle deposition; however, because it is determined by the mainstream, it is discussed in this section. The size of the secondary flow structures that exist in the vane passages near the wall are on the order of the boundary layer thickness, which depends on the boundary layer development along the upstream endwall. For the endwall experiments in the current study, the boundary layer thickness was approximately 10% of the span which is similar to that in the engine [60].
3.2 Film-Cooling Scaling Parameters

A detailed discussion of the literature relevant to the most commonly used film-cooling parameters was presented in Section 2.1. As previously stated, density ratio, velocity ratio, blowing ratio, and momentum flux ratio are not necessarily independent operating parameters; however, numerous studies have been conducted to determine which of these parameters scale cooling effectiveness in different ranges of operating conditions.

Recall that studies by Pederson at al. [17] and Sinha et al. [18] revealed that cooling effectiveness scales with blowing ratio and is independent of density ratio when coolant jets remain attached to the wall (I < 0.4). When film-cooling jets are separated from the wall (I > 0.8), density ratio does not have a large effect on centerline effectiveness, but does affect the lateral spreading of the coolant. These findings are similar to the findings by Thole et al. [19] who concluded that momentum flux ratio was the parameter that best scaled the separation tendencies of a film-cooling jet. Based on results by Pederson et al. [17], Thole et al. [19], and Sinha et al. [18] the behavior of a film cooling jet and its ability to cool a surface is most dependent on momentum flux ratio.

In addition to the operating parameters described above, film-cooling is dependent on boundary layer thickness and coolant jet Reynolds number. The boundary layer thickness was discussed in Section 3.1 and is approximately 10% of the airfoil span and is matched in experiments. The coolant jet Reynolds numbers in the engine range from Re_d ~ 1000 at low blowing ratios to Re_d ~ 5000 at high blowing ratios. The coolant jet Reynolds numbers in lab experimental conditions range from Re_d = 800 at low blowing ratios to Re_d ~ 3000 at high blowing ratios. Cooling hole geometric parameters including hole spacing (P/d), hole length (L/d), and hole incidence angle (α) are the simplest parameters to match and are defined in Figure 2.2. For the current study, adiabatic effectiveness was measured at various operating conditions for different film-cooling geometries with dynamically simulated deposition. The film-cooling geometries and operating conditions tested in each facility for the current study are described in Chapter 4.
3.3 Particle Scaling Parameters

The scaling parameters necessary to model particle flow physics are described in this section. The dimensionless parameter used to scale particle trajectories which was briefly discussed in Chapter 2 is the Stokes number. The Stokes number is defined by the ratio of the particle relaxation time, $\tau_p$, to the time characteristic of the fluid, $\tau_f$, shown in Equation 3.1.

$$\text{Stk} = \frac{\tau_p}{\tau_f}; \tau_p = \frac{\rho_p d_p^2}{18 \mu}; \tau_f = \frac{L_c}{U_p}$$ (3.1)

As the Stokes number increases, the particle relaxation time relative to the time characteristic of the flow increases. When Stokes number is less than one, the particle relaxation time is lower than the time characteristic of the flow and particles tend to follow fluid streamlines. When Stokes number is greater than one, the particle inertia is too great to respond to changes in the flow. Particle deposition is highly dependent on the trajectories of a size range of particles that exist in the mainstream. Because the experiments for this study were conducted in laboratory conditions near standard temperature and pressure, it was necessary to set the size distribution of particles to match Stokes numbers that exist in actual engine conditions. Because the focus of this study is on the interaction between particles and film-cooling jets, the appropriate length scale for calculating Stokes number is the cooling hole diameter. It is important to note that separate analyses were performed for the flat plate, leading edge, and endwall deposition studies. As an example, the parameters necessary for performing the Stokes analysis for the endwall deposition studies in the vane cascade model are listed in Table 3.1. Figure 3.1 shows that the wax particle diameter required to match the engine condition Stokes number is approximately 10 times the diameter of a fly ash particle in actual engine conditions. The details for the Stokes analyses performed for each part of the current study are provided in Section 5.2.

In addition to scaling particle trajectories, it is also important to simulate the phase and thermal properties of particles that exist in engine conditions. Studies in the literature have shown that particle phase (solid or liquid) is of particular importance to deposition; therefore it is necessary to scale the thermal properties of particles as they exist in engine conditions. The time it takes a particle to solidify downstream of the combustor is an appropriate parameter for scaling the phase change process. A lumped mass approximation
was used to calculate temperature as a function of time for fly ash, sand, and wax particles. Biot numbers for fly ash particles, sand particles, and wax particles were calculated to be 0.2, 0.05, and 0.04 respectively. Because the Biot number for each case was much less than one, the lumped mass approximation was applicable and could be used to calculate temperature as a function of time for particles. The solidification of a particle immersed in a fluid with constant temperature takes place in two separate processes. During the first process, the temperature drops exponentially with time as shown by Equation 3.2 until it reaches the material solidification temperature.

$$T_p = \left(T_{p,i} - T_\infty\right) e^{-\frac{hA_p}{\rho_p C_p V_p t}} + T_\infty$$

(3.2)

The time it takes the particle to reach the material solidification temperature, $t_1$, can then be expressed by Equation 3.3.

$$t_1 = -\frac{\rho_p C_p V_p}{hA_p} \ln \left[\frac{T_{p,s} - T_\infty}{T_{p,i} - T_\infty}\right]$$

(3.3)

During the second process, the temperature remains at the solidification temperature until the particle loses the equivalent of the latent heat of fusion to the surrounding gases. The time it takes for the particle to lose the heat necessary to change from liquid to solid, $t_2$, is shown in Equation 3.4.

$$t_2 = \frac{\Delta h_{\text{fus}} \rho_p V_p}{hA_p \left(T_{p,s} - T_\infty\right)}$$

(3.4)

The total time it takes a particle to solidify from the time it is immersed in the hot surroundings is $t_s = t_1 + t_2$. To scale the solidification time from engine conditions to laboratory conditions, it is normalized by the amount of time it takes the particle to travel from the injection location to the turbine surface of interest. The expression for the thermal scaling parameter (TSP) is shown in Equation 3.5.

$$\text{TSP} = \frac{t_s}{L_c/\overline{U}_e}$$

(3.5)

where $L_c$ is the distance a particle travels while immersed in the surrounding hot gases. A particle with a TSP < 1 solidifies prior to reaching the turbine while a particle with a TSP > 1 is in molten form as it encounters the turbine surfaces. Although particles are subject to
different surroundings in the laboratory than in the engine, the TSP can be matched to scale the particle solidification time between engine and laboratory conditions. As an example of how TSP can be used to scale solidification time, the necessary operating conditions and thermophysical material properties for calculating TSP for fly ash and wax particles are listed in Table 3.2. To illustrate how TSP can be used to scale particle solidification, a plot of particle temperatures with respect to TSP for a 10µm fly ash particle in engine conditions and a 100 µm wax particle in laboratory conditions is shown in Figure 3.2. Particle solidification times, \( t_1 \) and \( t_2 \), from Equations 3.3 and 3.4 respectively are illustrated in Figure 3.2. Recall from the example Stokes analysis that the wax particle size necessary to match the particle trajectory of a 10 µm fly ash particle is 100 µm. Figure 3.2 shows that the 100 µm wax particle in the lab has the same TSP as the 10 µm fly ash particle in the engine. The initial wax temperature and surrounding gas temperature were determined iteratively for the given material properties to match Stokes number and TSP simultaneously for a given size range of particles. These two parameters were used to scale particle trajectories and solidification times for the leading edge study and the endwall studies. The particle scaling analyses for each set of experiments in the current study are described in detail in Section 5.2. The TSP had not been developed prior to conducting the flat plate studies; however, a similar technique, described in Section 5.2, was used to match particle solidification times between experiments.

Recall from Chapter 2 that Jensen et al. [45] conducted deposition studies in the Turbine Accelerated Deposition Facility (TADF). By increasing the concentration of particulate matter in the gases that simulated the turbine mainstream, they injected the same amount of particulate matter over the span of four hours that would pass through an actual gas turbine in 10000 hours. When conducting an accelerated test by increasing the particulate concentration, the limiting factor is the extent of particle-to-particle interactions. Increasing the concentration of particulate is acceptable until particle-to-particle interactions can no longer be neglected. The mass loading of the multiphase flow is the parameter that quantifies the concentration of particulate matter. The mass loading parameter is simply the ratio of particulate mass to the continuum fluid mass as shown by Equation 3.6.

\[
m_{\text{L}} = \frac{\bar{m}_{\text{p}}}{\bar{m}_{\text{f}}} \tag{3.6}
\]
In practice, if the mass loading parameter is less than 6\% then the mixture is considered dilute and particle-to-particle interactions are negligible [66]. A large turbine experiences a flow path concentration on the order of 1 ppmw in a typical 8000 hr operating year [2]. A 1 ppmw concentration satisfies the mass loading requirement to neglect particle-to-particle interactions. Deposition can be simulated at an accelerated rate in the laboratory and still satisfy the $m_L << 6\%$ requirement to insure that particle-to-particle interactions are not significant enough to have an effect on deposition results. Specific details for the scaling used for each individual study to simulate deposition are described in Section 5.2.

### 3.4 Airfoil Geometry Scaling

Large scale models are often utilized to measure adiabatic effectiveness and ultimately determine the optimum operating conditions for film-cooling. The three most critical regions of a turbine cascade where film-cooling is most necessary are the airfoil pressure side, the airfoil leading edge, and the cascade endwall where secondary flows generate high heat transfer. Rather than modeling all regions using one model, it is often more practical and enlightening to simplify the experiment by exploring one region at a time. This section is devoted to explaining how the pressure side, leading edge, and endwall can each be experimentally modeled individually. Chapter 4 provides detailed descriptions for the experimental methods used to model pressure side, leading edge, and endwall film-cooling.

The airfoil pressure side is highly sensitive to heat transfer from the surrounding hot gas path which makes film-cooling in this region critical. To understand pressure side film-cooling and its driving mechanisms, experimentalists typically model the region using a flat plate. Flat plate facilities are typically preferred over turbine models for fundamental film-cooling experiments because they allow for higher resolution measurements to be made through a larger range of operating conditions. The flat plate method is viable for modeling pressure side film-cooling because secondary flow structures associated with flow through an airfoil cascade do not play a large role in pressure side cooling. The parameters that are critical for modeling pressure side film-cooling are mainstream turbulence intensity and coolant momentum flux ratio which can be properly set using a flat plate facility.
The showerhead cooling configuration in the airfoil stagnation region can also be modeled in a simplified manner. The leading edge experiments were conducted to determine the effects of deposition on showerhead cooling specifically at the midspan of a turbine airfoil. At the midspan region secondary flow effects are negligible; therefore, the region can be modeled using a symmetric model with no neighboring airfoils. As mentioned in Section 2.1, the leading edge region of an airfoil is often modeled using a symmetric half-cylinder [21, 22]. The mainstream flowfield around the cylinder can be scaled with true turbine hardware using the Reynolds number with the leading edge diameter as the length scale. Film-cooling operating conditions can then be characterized using the blowing ratio based on the mainstream inlet velocity. One problem associated with flow around a cylinder is the wake shedding effect that exists at moderate to high Reynolds numbers. For this study, the wake shedding effect was eliminated by attaching an aft-body airfoil to the downstream side of the cylinder.

As discussed in Chapter 2, the endwall region is subject to violent secondary flow structures that lead to high heat transfer from the hot gas path to the endwall. Because secondary flow structures are highly dependent on the airfoil geometry, endwall film-cooling experiments are typically conducted in turbine cascades. It is conventional to model at least two passages to capture the effects of secondary flows on endwall cooling effectiveness. To accurately model the secondary flows, the Reynolds number and relative boundary layer thickness must be matched. To scale film-cooling operating conditions, experimentalists typically match momentum flux ratio or blowing ratio between laboratory and engine conditions.

Descriptions of each scaling parameter discussed in this section are provided in Sections 3.1 through 3.3. Experimental facility details for each section discussed in this section are provided in Chapter 4. The measurement methods used for each study are described in Chapter 5.
Table 3.1. Stokes Analysis Conditions

<table>
<thead>
<tr>
<th></th>
<th>Engine (Fly Ash)</th>
<th>Laboratory (Wax)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Particle Density, $\rho_p$ (kg/m$^3$)</td>
<td>1980 [9]</td>
<td>900</td>
</tr>
<tr>
<td>Inlet Gas Velocity, $U_\infty = U_p$ (m/s)</td>
<td>93 [60]</td>
<td>6.3</td>
</tr>
<tr>
<td>Gas Viscosity, $\mu$ (kg/m-s)</td>
<td>$5.549 \times 10^{-5}$</td>
<td>$1.852 \times 10^{-5}$</td>
</tr>
<tr>
<td>Hole Diameter, $d=L_c$ (mm)</td>
<td>0.511</td>
<td>4.6</td>
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</table>

Table 3.2. Thermal Analysis Conditions

<table>
<thead>
<tr>
<th></th>
<th>Engine (Fly Ash)</th>
<th>Laboratory (Wax)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Particle Density, $\rho$ (kg/m$^3$)</td>
<td>1980 [9]</td>
<td>900</td>
</tr>
<tr>
<td>Specific Latent Heat of Fusion, $\Delta h_{fus}$ (J/kg)</td>
<td>650000 [67]</td>
<td>150000</td>
</tr>
<tr>
<td>Specific Heat, $C_p$ (J/kg-K)</td>
<td>730 [68]</td>
<td>2890</td>
</tr>
<tr>
<td>Material Melting Temperature, $T_{p,s}$ (K)</td>
<td>1533 [69]</td>
<td>338</td>
</tr>
<tr>
<td>Turbine Inlet Temperature, $T_\infty$ (K)</td>
<td>1509 [70]</td>
<td>330</td>
</tr>
<tr>
<td>Combustion Temperature, $T_{p,i}$ (K)</td>
<td>1593 [70]</td>
<td>345</td>
</tr>
<tr>
<td>Gas Viscosity, $\mu$ (kg/m-s)</td>
<td>$5.55 \times 10^{-5}$</td>
<td>$1.82 \times 10^{-5}$</td>
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<tr>
<td>Particle Travel Distance (combustor to turbine), $L_\infty$ (m)</td>
<td>0.26</td>
<td>2.34</td>
</tr>
<tr>
<td>Particle Velocity, $U_\infty$ (m/s)</td>
<td>93 [60]</td>
<td>6.3</td>
</tr>
</tbody>
</table>

Figure 3.1. Wax particle size range necessary to match Stokes numbers of fly ash particles in engine conditions.
Figure 3.2. Fly ash and wax particle temperatures plotted with respect to TSP.
CHAPTER 4:
Experimental Facilities

Experiments to quantify deposition on cooling effectiveness were conducted using two separate facilities. For the flat plate studies, experiments were conducted in an open loop test channel designed specifically for developing a method to dynamically simulate deposition and quantify its impact on cooling effectiveness in the simplest case. The leading edge studies and all endwall studies were conducted in a large scale low speed wind tunnel; however, three different experimental configurations were used.

For the leading edge studies, a symmetric cylinder and aft body airfoil were used to model the vane leading edge with showerhead film-cooling. To determine the effects of deposition on endwall film-cooling with and without trenches, experiments were conducted using a large scale first stage vane cascade model. To determine the effects of deposition on endwall film-cooling with and without contouring, experiments were conducted using a large scale low pressure turbine blade cascade model. Details for each facility and test section used to conduct deposition experiments on a film-cooled flat plate, leading edge, and endwall are described in this chapter. Section 4.1 describes the open loop test channel used for conducting flat plate experiments, Section 4.2 describes the large scale facility used to conduct leading edge experiments, and Section 4.3 describes how the same large scale facility was used to conduct endwall experiments using two separate test sections.

4.1 Flat Plate Film-Cooling Facility

Deposition simulations and flat plate film-cooling effectiveness tests were conducted in a low speed open loop test channel shown in Figure 4.1. Room air was heated at the suction side of the blower at the inlet to the plenum. Upon entering the plenum, the heated air impinged onto a splash plate that prevented the air from propagating through to the test channel without mixing. The flow developed in a duct with a height of 6.9 cm (10.8d), a width of 41.7 cm (65.6d), and a hydraulic diameter of 11.8 cm (18.6d). A row of nine film-cooling holes with an L/d = 3, P/d = 3, and α = 30° was drilled in a removable endwall plate. Placed 130d (7Dh) upstream of the holes at the tunnel entrance was a rounded inlet, a section of honeycomb, a double layer of screens, and a 2 mm boundary layer trip wire. The endwall plate was constructed from a 2.5 cm thick low thermal conductivity polyurethane foam block.
(k = 0.033 W/m-K) that minimized conduction losses to create an adiabatic wall condition. The surface of the film-cooling plate exposed to the flow was covered by a thin layer of black contact paper. The contact paper provided a smooth surface that was easily replaceable between test cases. The mainstream flow continued 15 D_h (279d) downstream from the film-cooling plate before exiting through particle filters. Laser Doppler velocimetry and particle image velocimetry were both used to measure turbulence intensity, Tu, with and without the honeycomb and screens installed. The freestream turbulence intensity was 4.6% with the honeycomb and screens installed and 12.3% without the honeycomb and screens. All baseline and deposition tests were conducted at high freestream turbulence (Tu = 12.3%) while only the baseline that provided a benchmark was completed at low freestream turbulence.

Adiabatic effectiveness was quantified by heating the mainstream air to approximately 318 K and injecting coolant air at approximately 298 K. The mainstream velocity was set at 8 m/s while the coolant flow was manually controlled to set the momentum flux ratio for a given test. Compressed air was used as the coolant and was routed through a plenum located beneath the film-cooling plate as seen in Figure 4.1. The coolant volumetric flowrate was measured using a laminar flow element (LFE).

The methods used to simulate deposition and measure adiabatic effectiveness on the film-cooled flat plate are described in Chapter 5. Experimental results obtained for validation are presented along with the rest of the results from the flat plate studies in Chapter 6.

4.2 Leading Edge Film-Cooling Facility

Dynamic deposition simulations and adiabatic effectiveness measurements for the leading edge studies were conducted in a low-speed wind tunnel. Flow through the low speed wind tunnel, shown in Figure 4.2, was supplied by a 37kW axial fan. Downstream of the fan, the flow was cooled by a primary heat exchanger before splitting into two coolant flow paths and a mainstream flow path. The mainstream flow passed through a series of resistance heaters which heated the flow to 318K while the coolant flow paths passed through secondary heat exchangers which cooled the coolant flow to 295K. It is important to note that only the top coolant flow path was active for the current study. Downstream of the
heater bank, the mainstream flow passed through a series of flow straightening screens and honeycomb. Downstream of the flow conditioning, the flow passed through a turbulence grid to create 4% turbulence intensity with a length scale of 0.16D in the mainstream flow at stagnation [71].

For the current study, a showerhead cooling geometry was modeled using a cylindrical test piece with a symmetric aft body airfoil. The cylinder was placed in the mainstream flow channel having a width of 8.8D resulting in a blockage ratio of 0.11. For the purpose of this study, blockage effects were assumed to be negligible considering the low blockage ratio. The symmetric aft body airfoil was added to prevent wake shedding from the cylinder which could affect the flow physics in the showerhead cooling region. Figure 4.3 illustrates the showerhead cooling configuration modeled with the cylindrical test piece and aft body airfoil.

Mainstream flow conditions were scaled by matching the cylinder Reynolds number between engine and laboratory conditions. Because experiments were conducted in a low speed wind tunnel, the cylindrical test piece was scaled up to match the Reynolds number. The film-cooling operating conditions were scaled using the average blowing ratio of the coolant to mainstream flow. An auxiliary blower was used to supply coolant from the upper coolant flow path to the cylindrical test piece. An LFE located in the coolant flow path upstream of the cylindrical test piece was used to measure the coolant flowrate which could be adjusted using the variable speed blower. The local blowing ratio, $M_L$, was calculated at each cooling row location using the theoretical pressure distribution around a circular cylinder and assuming a constant discharge coefficient, $C_D = 0.6$ [72]. Although the value of $C_D$ depends on internal and external flow conditions as well as cooling hole orientation, using an average value was adequate for the analysis in this study. Table 4.1 shows the geometric specifications and operating conditions used to scale the engine conditions to the laboratory model. The showerhead geometric specifications and corresponding engine operating conditions were provided by an industry sponsor [11].

A top and side view of the flow channel with the cylindrical test piece location and velocity measurement points are shown in Figure 4.4. The mainstream velocity profile, shown in Figure 4.4a, was measured using a pitot-static probe at five locations across the span of the channel upstream of the cylindrical test piece. Pressure taps were installed in the
cylinder surface near stagnation to verify that flow around the leading edge was symmetric as shown by the measured dimensionless pressure distribution in Figure 4.5.

4.3 Endwall Film-Cooling Facilities

As described in the introduction to Chapter 4, endwall experiments were conducted in the low speed wind tunnel using two separate large scale test sections. The PW6000 vane cascade test section was used to determine the effects of deposition on endwall film-cooling with and without transverse trenches. A separate study was conducted using the Pack-B blade cascade test section to determine the effects of deposition on endwall film-cooling with and without endwall contouring. The geometric specifications and operating conditions for both endwall film-cooling facilities are discussed in the following subsections.

First Vane Test Section

To determine the effects of deposition on endwall film-cooling with and without transverse trenches, experiments were conducted in the same low speed wind tunnel used for the leading edge studies described in Section 4.2. A PW6000 large scale turbine cascade test section was located in the closed loop wind tunnel shown in Figure 4.6. The PW6000 is a commercial vane geometry used by Pratt & Whitney and has been used in various experimental studies in the literature [42, 43, 65]. For the experiments conducted using the vane cascade test section, the turbulence grid was located 3.6 chord lengths upstream of the vane cascade and was used to achieve 4% mainstream turbulence intensity at the entrance to the vane cascade test section [15].

The vane cascade test section consisted of two full passages with one center vane, a full neighboring vane, and a half neighboring vane. The operating conditions and geometric specifications for the vane cascade are shown in Table 4.2 and are described in detail by Radomsky and Thole [65]. The inlet Reynolds number shown in Table 4.2 was matched with the engine operating conditions to scale the mainstream flow. Radomsky [73] solved for the inviscid pressure distribution around the PW6000 vane using a two-dimensional CFD simulation.

Pressure taps were installed in the center vane to measure the static pressure distribution around the mid-span. Prior to running each set of experiments in the vane
cascade test section, care was taken to ensure that the measured non-dimensional static pressure distribution around the vane matched the theoretical solution by Radomsky [73]. The non-dimensional static pressure coefficient is defined by Equation 4.1.

\[ C_p = \frac{p - p_\infty}{\frac{1}{2} \rho U^2} \]  

(4.1)

Figure 4.7 shows the measured and predicted \( C_p \) distribution around the mid-span of the center vane.

A film-cooled endwall with a cooling hole pattern designed by Knost and Thole [74] is shown in Figure 4.8. A 45° angled slot located upstream of the cascade was present to simulate coolant leakage at the interface between the combustor and the turbine. It is important to note that the leakage coolant to mainstream mass flux ratio was 0.75% for every experiment conducted using the vane cascade test section.

The endwall was constructed out of low thermal conductivity polyurethane foam (\( k = 0.033 \text{ W/m-K} \)) and a balsawood (\( k = 0.055 \text{ W/m-K} \)) layer, as shown in Figure 4.8, was added on top of the foam in the leading edge region to build up the necessary thickness for each trench geometry tested. Adiabatic effectiveness was quantified on the leading edge region of the endwall for cooling rows with no trench and with trench depths of 0.4d, 0.8d, and 1.2d. Figure 4.8b shows that the leading edge cooling row is aligned with the flow and the passage cooling row is oriented at a compound angle of 90° relative to the flow. All film-cooling holes were drilled at 30° incidence angles and the coolant flow direction for each row is indicated by the arrows in Figure 4.8b.

Air from the top cooling passage was used to supply coolant to two separate plenums. One plenum was used to supply coolant to the 45° upstream slot to simulate coolant flow through the combustor turbine interface. Another plenum was located beneath the endwall of the turbine cascade test section and was used to supply coolant to the endwall film-cooling holes. The variable speed blower located on top of the wind tunnel controlled the airflow to the coolant plenums and could be used to set the coolant flow conditions. Film-cooling flow conditions were characterized by the momentum flux ratio derived from the ideal blowing ratio, \( M_{\text{ideal}} \), of the cooling hole located directly upstream of the stagnation point.
Blade Test Section

To determine the effects of deposition on endwall film-cooling with and without endwall contouring, experiments were conducted in the same low speed wind tunnel used for the leading edge film-cooling studies and the vane cascade studies. A Pack-B large scale blade cascade test section was located in the closed loop wind tunnel as shown in Figure 4.9. The Pack-B is a commercial low pressure turbine blade geometry used by Pratt & Whitney and has been used in various experimental studies in the literature [7, 10, 15, 31]. For the blade cascade experiments, the turbulence grid was located 9.1 chord lengths upstream of the blade cascade and was used to achieve 4% mainstream turbulence intensity at the entrance to the blade cascade test section [15].

The blade cascade consisted of seven blades with six full passages. Two of the six passages were equipped with the endwall cooling geometry tested for this study. Figure 4.10 shows the flat endwall configuration of the blade cascade with the passages of interest (passage 2 and passage 3) identified. Like the previously described facilities, the endwall was constructed out of low conductivity polyurethane foam (k = 0.033W/m-K) so that adiabatic effectiveness could be quantified using IR thermography as described in Section 5.1. The film-cooling holes in passage 2 were identical to the film-cooling holes in passage 3. Hole 1 was drilled at an inclination angle of 45° relative to the surface while holes 2-5 were drilled at an inclination angle of 40° relative to the surface. All five cooling holes in the flat endwall had L/d = 4.1.

Following the flat endwall experiments in the blade cascade, film-cooled contoured endwall passages were installed to determine the effects of deposition on cooling with a non-axisymmetric contoured endwall. Figure 4.11 shows the non-axisymmetric contoured endwall which was designed using a computational optimization technique by Praisner et al. [7]. The contoured endwall sections were cast individually in a mold using polyurethane two-part expanding foam (k = 0.033 W/m-K). The five cooling hole locations were held constant between the flat and contoured endwalls; however, because of the non-uniform surface, inclination angles of cooling holes in the contoured endwall varied from hole to hole. Holes 1 and 2 had inclination angles of 80° and 60° while holes 3-5 had inclination angles ranging from 45° to 30°. The L/d ratios for the cooling holes in the contoured endwall varied
from 7 to 8.5. All cooling holes in the flat and contoured endwalls were aligned corresponding to oil flow visualization streaklines determined by Lynch et al. [15].

The auxiliary blower shown in Figure 4.9 was used to supply coolant from the top coolant plenum in the wind tunnel to the film-cooling plenum located beneath the endwall of the cascade. Film-cooling operating conditions were characterized by the average ideal blowing ratio, \( M_{\text{ideal}} \), of all five holes for a given experiment. The ideal blowing ratio was determined using measurements of plenum pressure and local endwall static pressure. The local endwall static pressures were measured at each hole exit location using pressure taps mounted in an adjacent passage without film-cooling. The operating conditions and geometric specifications used for this study are listed in Table 4.3 and were identical to those used by Lynch et al. [10].

Similar to the experiments conducted in the vane cascade test section, the \( \text{Cp} \) distribution was measured around the center blade in the cascade to ensure accurate comparison to the theoretical \( \text{Cp} \) distribution around the midspan of the airfoil. Because there were seven blades in the cascade, pressure taps were installed in strategic locations around the mid-span of every blade to ensure periodicity. The measured and theoretical \( \text{Cp} \) distributions around the center blade along with additional \( \text{Cp} \) values measured on adjacent blades are shown in Figure 4.12.
### Table 4.1. Geometry and Operating Conditions for Leading Edge Studies

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Engine [11]</th>
<th>Lab</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density ratio, DR</td>
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<td>1.1</td>
</tr>
<tr>
<td>Blowing ratio, M</td>
<td>0.50 - 1.80</td>
<td>0.50 - 1.80</td>
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<tr>
<td>Momentum flux ratio, I</td>
<td>0.13 - 1.71</td>
<td>0.23 - 3.05</td>
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<tr>
<td>Velocity ratio, VR</td>
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<td>0.46 - 1.68</td>
</tr>
<tr>
<td>Hole incidence angle, α (deg)</td>
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<td>40.00</td>
</tr>
<tr>
<td>Hole pitch to diameter ratio, P/d</td>
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<td>3.6</td>
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<tr>
<td>Hole length to diameter ratio, L/d</td>
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<tr>
<td>Reynolds number based on D</td>
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<td>56800</td>
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### Table 4.2. Geometric and Flow Conditions for Vane Cascade Studies

<table>
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<th>Parameter</th>
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</tr>
</thead>
<tbody>
<tr>
<td>Scaling factor</td>
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</tr>
<tr>
<td>True Chord, C</td>
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</tr>
<tr>
<td>Pitch/Chord, P/C</td>
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<tr>
<td>Span/Chord, S/C</td>
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<tr>
<td>Hole, L/D</td>
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<tr>
<td>Momentum Flux Ratios, I</td>
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</tr>
<tr>
<td>Density Ratio, DR</td>
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</tr>
<tr>
<td>Slot Blowing Ratio</td>
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</tr>
<tr>
<td>Re_{in}</td>
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<tr>
<td>Inlet and exit angles</td>
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</tr>
<tr>
<td>Inlet, exit Mach number, Ma_{in}, Ma_{ex}</td>
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<tr>
<td>Inlet mainstream velocity, U_{∞}</td>
<td>6.3 m/s</td>
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</table>

### Table 4.3. Geometric and Flow Conditions for Blade Cascade Studies

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<th>Parameter</th>
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</tr>
</thead>
<tbody>
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<td>Scaling Factor</td>
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<tr>
<td>Axial Chord, C_{ax}</td>
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<tr>
<td>True Chord, C</td>
<td>0.241 m</td>
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<tr>
<td>Pitch/Chord, P/C</td>
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</tr>
<tr>
<td>Span/Chord, S/C</td>
<td>2.26</td>
</tr>
<tr>
<td>Inlet Reynolds Number, Re_{in}</td>
<td>1.25 x 10^5</td>
</tr>
<tr>
<td>Exit Reynolds Number, Re_{ex}</td>
<td>2.00 x 10^5</td>
</tr>
<tr>
<td>Inlet, exit flow angles</td>
<td>35° and 60°</td>
</tr>
<tr>
<td>Inlet, exit Mach Number, Ma_{in}, Ma_{ex}</td>
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</tr>
<tr>
<td>Average Ideal Blowing Ratios, M_{ideal}</td>
<td>1.0, 2.0</td>
</tr>
<tr>
<td>Density Ratio, DR</td>
<td>1.07</td>
</tr>
</tbody>
</table>
Figure 4.1. Schematic of the (a) open loop test channel and (b) film-cooling geometry used for the flat plate study.

Figure 4.2. Illustration of wind tunnel facility for Leading Edge Studies

Figure 4.3. Schematic illustrating the cylindrical leading edge and aft body dimensions.
Figure 4.4. Schematic showing (a) measured velocity profile, (b) test section top view, and (c) test section side view used for leading edge studies.

Figure 4.5. Measured and theoretical dimensionless pressure distributions around the cylinder leading edge.
Figure 4.6. Illustration of wind tunnel facility used for vane cascade studies.

Figure 4.7. Measured and theoretical non-dimensional local pressures around the center vane in the cascade.
Figure 4.8. Schematic of the (a) endwall film-cooling configuration, (b) leading edge region, and (c) cross-section of the stagnation region of the vane cascade endwall.

Figure 4.9. Illustration of wind tunnel facility with the blade cascade test section attached.
Figure 4.10. Schematic of the blade cascade flat endwall configuration with passages 2 and 3 expanded.

Figure 4.11. Schematic of the non-axisymmetric contoured endwall used in the blade cascade with isometric view and expanded top view.
Figure 4.12. Measured and theoretical non-dimensional local pressures around blades in the blade cascade test section.
CHAPTER 5:
Measurement Methods and Deposition Simulation Techniques

Experimental and computational techniques were developed to dynamically simulate particle deposition such that the effects of deposition on cooling could be quantified. For all experimental conditions in each study, two tests were conducted. First, deposition was dynamically simulated by injecting atomized wax into the mainstream flow. Following deposition simulation, an adiabatic effectiveness experiment was conducted to quantify the effect of deposition on cooling effectiveness. Section 5.1 describes the methods used to quantify film-cooling effectiveness experimentally by measuring adiabatic wall temperatures. Section 5.2 describes the development of various methods that were used to dynamically simulate deposition. Section 5.3 describes the flow visualization techniques that were used to gain a better understanding of how particles interact with film-cooling jets on the endwall with and without transverse trenches. Finally, Section 5.4 describes the computational methods that were used to predict particle accretion and draw comparisons with experimentally simulated deposition. The operating conditions for each deposition simulation and adiabatic effectiveness experiment are listed in the test matrices presented in Appendix B for each corresponding set of experiments.

5.1 Adiabatic Effectiveness Measurements

The following subsections describe the methods used to quantify film-cooling effectiveness by measuring adiabatic wall temperatures using IR thermography. It is important to note that all data was acquired from steady state experiments which each required between three and five hours for completion. During experiments, thermocouple measurements were monitored periodically to determine when steady state was achieved. Although the technique was similar for each set of experiments conducted, the following subsections are necessary to describe the differences between the methods used for the flat plate, leading edge, and endwall studies. The same methods described in this section for measuring adiabatic effectiveness before deposition were used to measure effectiveness after deposition simulations as well. For the leading edge and endwall deposition studies, the deposition was painted black prior to measuring adiabatic effectiveness values to account for changes in emissivity. For the flat plate deposition studies, the small amount of deposition
on the surface had a negligible effect on the surface emissivity as determined experimentally. The method for treating deposition for IR measurements is described in detail in Section 5.2.

**Flat Plate Study**

For the flat plate studies, a FLIR P20 IR camera was used to acquire a temperature map over the entire film-cooled flat plate shown in Figure 4.1b. Compared to conventional methods in which measurements could only be acquired where thermocouples were located, the spatially resolved technique used in this study allowed for adiabatic effectiveness to be calculated with 875 μm resolution everywhere on the film-cooled surface. A removable Zinc Selenide (ZnSe) window served as the top wall located above the film-cooling plate. This ZnSe window allowed for maximum transmission (~70%) to the IR camera which was mounted above the test section in a box to minimize radiation from the surroundings. A series of five IR images was taken upon reaching a steady state condition for each test. Three calibration thermocouples were located on the surface of the film-cooling plate in locations having different temperatures to cover the entire range measured by the IR camera for a given test. With the use of FLIR Systems software, image calibration was performed by adjusting the background temperature and emissivity to match image temperature values with corresponding thermocouple measurements to within the bias uncertainty of the thermocouple measurement (~0.5°C). The set of calibrated temperatures could then be used to calculate measured adiabatic effectiveness values using Equation 2.3. To account for conduction losses through the foam endwall the measured effectiveness values were corrected using the equation for η in the nomenclature where η_corr was set to the measured effectiveness value taken upstream of the cooling holes. The upstream measured effectiveness value was typically around 0.05 and was used to correct for conduction because the cooling effectiveness in that region was directly attributable to the conduction through the endwall.

**Leading Edge Study**

For the leading edge experiments, surface temperatures were measured using a 640 by 480 resolution FLIR SC620 IR camera resulting in a measurement resolution of 0.06d. Upon reaching steady state, five IR images from two separate port locations, shown in Figure
4.4, were acquired to obtain a temperature map around the leading edge circumference of the cylindrical test piece. ZnSe windows were used at each port location to prevent mainstream leakage flow through the IR ports and to allow transmission to the IR camera.

Four thermocouples used for IR image calibration were mounted in hot and cold locations on the cylindrical surface. Similar to the flat plate data, each IR image was calibrated by adjusting emissivity and background temperature until the IR measurements matched the corresponding calibration thermocouple measurements. The five calibrated IR images from each port location were then averaged to yield one data set for each port location.

Because the surface of interest was curved, each calibrated data set required spatial transformation. Prior to running experiments, a 12.7 by 12.7 mm grid was placed on the cylindrical surface and an IR image of the grid was acquired at each camera port location. The grid point locations were then identified and the data was effectively unwrapped from the curved surface so that the data around the curved surface could be represented on a plot with rectangular coordinates. The transformation algorithm developed using the grid was used to transform the data from every ensuing experiment. This IR transformation method was developed for experiments by Colban et al. [76].

Following image transformation, measured adiabatic effectiveness, $\eta_{\text{meas}}$, was calculated at every pixel location in the IR data set. After calculating the measured adiabatic effectiveness, each resulting matrix of measured effectiveness values were corrected to account for the heat loss through the foam wall ($k = 0.033$ W/m-K). Although the polyurethane foam had a low thermal conductivity it was not a purely adiabatic surface. A one-dimensional conduction correction was implemented to account for the heat loss through the foam wall to the coolant plenum inside the cylinder. To correct for the heat loss through the foam cylinder wall, the final corrected effectiveness values, $\eta$, were calculated using the equation in the nomenclature, where $\eta_{\text{corr}}$ is the equivalent effectiveness attributable to heat loss through the cylinder wall. The value for $\eta_{\text{corr}}$ was equal to the minimum $\eta_{\text{meas}}$ value for every experiment which was typically around 0.03. The minimum measured effectiveness value for each case was in a region on the cylinder surface where the only measured effectiveness was that which was attributable to conduction losses through the cylinder wall.
Vane Cascade Test Section

For the experiments conducted in the vane cascade, IR images were acquired at five port locations perpendicular to the endwall around the vane leading edge at a distance of 55 cm from the endwall surface. For the endwall experiments a FLIR P20 IR camera with a resolution of 320 by 240 pixels was used to acquire data in a viewing area of 24 cm by 18 cm resulting in 715 µm (0.16d) resolution.

Upon reaching steady state for a given experiment, IR images were taken at each of the five port locations. At each port location, five images were acquired and calibrated using thermocouples placed in discrete locations on the endwall surface. Similar to the flat plate and leading edge studies, images were calibrated by adjusting the background temperature and emissivity until the IR temperatures matched the thermocouple measurements for each corresponding thermocouple location.

Although the endwall was constructed with low thermal conductivity foam, it was still necessary to apply a one-dimensional conduction correction described by Ethridge et al. [77]. To perform the conduction correction, cool air was supplied to the coolant plenums while the upstream slot and the film-cooling holes were blocked to prevent external surface cooling. By simulating the temperature difference between the external surface with no cooling and the internal surface exposed to the coolant air, a conduction correction could be imposed on the final effectiveness results. The spatially resolved correction indicated values as high as $\eta = 0.12$ upstream of the test section and $\eta = 0.06$ on the foam endwall areas. To apply a local correction, the spatially resolved values were subtracted from the measured values and normalized.

Blade Cascade Test Section

The method used to quantify adiabatic effectiveness during the blade cascade experiments was almost identical to the method used for the vane cascade experiments. Data was acquired for the two passages illustrated in Figures 4.10 and 4.11. Five IR images were acquired through each of six port locations. The same calibration and conduction correction procedures described for the vane cascade experiments were followed for reducing the data obtained in the blade cascade test section. Conduction correction effectiveness values were approximately 0.06 for every experiment conducted in the blade cascade test section.
5.2 Experimental Deposition Simulation and Analysis

The method used to dynamically simulate deposition evolved through the course of the study. Various materials were tested in the open loop test channel used for the flat plate study before low melting temperature wax was chosen as the injectant material. The method developed in the flat plate facility was improved with the implementation of a sophisticated injection system which was used in the low speed wind tunnel. For the leading edge study, a one nozzle injection system was used and for the endwall studies a two nozzle injection system was used. Mainstream conditions were varied and waxes with various melting temperatures were used to simulate deposition for a range of turbine operating conditions. The details for the simulations conducted in each individual study are described in the following subsections.

Flat Plate Study

The objective of the flat plate studies was to develop a methodology that could be used in a laboratory setting to simulate deposition. The method was developed so that adiabatic effectiveness tests could be conducted on a surface of interest following deposition simulation. It was necessary to inject particles into a hot gas path that could simulate the deposition of both solid and molten materials. It is important to note that all flat plate deposition simulations were conducted at high freestream turbulence (Tu = 12.3%). Prior to the film-cooling studies, various types of particles were injected into a flow with a cylindrical obstacle to observe the deposition behavior. Initially, sand was injected to simulate the deposition of solid particulate. Then, multiple waxes having different melting temperatures were injected individually to observe their deposition behavior. Photos of sand deposition as well as wax deposition are shown in Figure 5.1. Both materials deposited in a manner that illustrated the horseshoe vortex and saddle point locations on the endwall. The photos in Figure 5.1 show examples of large particle deposition by inertial impaction on the cylinder and small particle deposition by turbulent diffusion and eddy impaction on the endwall. Ultimately, wax was chosen to simulate both solid and molten particles under the assumption that only the largest particles would be in molten form upon reaching the deposition test surface. A lumped mass assumption was made to calculate the solidification time of a particle immersed in the mainstream flow. For a particle with given material properties,
solidification time was directly dependent on particle size as well as initial wax temperature and the mainstream temperature. Based on the achievable test conditions in the low speed test channel, wax having a melting temperature of 328 K was chosen as the material for injection.

To simulate particle trajectories, it was necessary to match the wax Stokes number range to the Stokes number range that exists in actual engine conditions. Table 5.1 shows the engine conditions and the simulation conditions used for the flat plate studies to calculate Stokes numbers. Figure 5.2 shows the wax particle size necessary to match the Stokes number for a given engine particle size. For the flat plate studies, the wax particle diameter required to match the engine condition Stokes number was approximately 13 times the diameter of a coal ash particle in actual engine conditions. Figure 5.3 shows photographs of coal ash [9] and wax particles taken using an environmental scanning electron microscope (ESEM). The photos show that the wax particles are spherical like the coal ash particles, but are an order of magnitude larger.

The wax particle size distribution in the open loop test channel was measured using a Malvern Spraytec particle analyzer capable of characterizing aerosol droplets in the size range of 0.1 to 1000 µm. Figure 5.4 shows the size distribution and corresponding Stokes numbers for the wax particles in the open loop test channel facility. Recall from Figure 5.2 that a 130 µm wax particle in simulation conditions is scaled by Stokes number to match the trajectory of a 10 µm coal ash particle in engine conditions. In Figure 5.4 there is a vertical line at 130 µm which intersects the % Mass Passing curve at approximately 90% (as shown on the secondary y-axis) indicating that approximately 90% (by mass) of the wax particles are smaller than 130 µm in diameter.

Figure 5.5 indicates temperatures of wax particles that are and are not exposed to the film-cooling jet relative to convective distance for a 13 µm particle and a 130 µm particle. The particle temperature relationship to travel distance was calculated using the methods discussed in Section 3.3. Initially, all particles are in molten form and are losing heat to the mainstream while they cool. By assuming the particle moves at the same velocity as the mainstream, the heat transfer coefficient can be calculated from the analytical solution for conduction from a sphere [78]. The particle temperature as a function of time is shown by Equation 3.2.
When the temperature of a given particle reaches the material solidification temperature, the phase change process begins. During the phase change process, the particle continues to lose heat to the surroundings; however, the particle remains at the solidification temperature of the material. The time that the phase change process takes to complete is calculated using Equation 3.4. When the particle is exposed to the film-coolant, the governing equations remain unchanged; however, the temperature of the surroundings changes to the coolant temperature.

As seen in Figure 5.5, the 13 µm particle cools faster than the 130 µm particle and is at the same temperature as the mainstream upon reaching the film-cooling row. When the 13 µm solid particle enters the coolant, the particle temperature quickly drops to the coolant temperature further reducing its chances of depositing. The 130 µm particle does not solidify prior to reaching the coolant jets and is in the process of solidifying when it becomes entrained in the coolant flow. The coolant does increase the rate of heat loss from the particle, but does not reduce the particle temperature until the phase change process has completed. When exposed to the coolant gases, the 130 µm particle would be in solid form 370d downstream of the cooling row. If the 130 µm particle was not exposed to the coolant flow, it would not solidify until it reached a distance of 940d downstream of the cooling row. Because the 130 µm particle is in molten form upon reaching the cooling row, it has a greater chance of depositing just downstream of the coolant holes than the 13 µm particle.

Deposition tests were carried out by injecting molten wax particles using a spray gun specially designed for high viscosity fluids. The spray gun was attached to the particle injection port located on the side of the mainstream plenum downstream of the splash plate. A resistance heater was used to heat the wax reservoir that was instrumented with a thermocouple to monitor the initial wax temperature prior to injection. For each test case, 400 g of wax was injected eight times. The initial temperature of the molten wax was set so that particles greater than 50 µm in diameter would remain in molten form for a distance of 100 cm allowing them to reach the film-cooled flat plate. After each injection cycle, a series of IR images of the film-cooled endwall were taken and calibrated using the method described in the previous section. For the flat plate experiments, the surface was not painted black after deposition as it was in the other experiments described in the following sections. The IR measurement accuracy was validated by conducting an experiment with half of the
deposition painted black and half of the deposition left unchanged. The results from the unpainted side matched the results from the painted side within experimental uncertainty. It was determined that painting of the deposition was not required for the flat plate studies because the small amount of deposition on the surface had a negligible effect on the surface emissivity.

In addition to acquiring IR images, digital photographs were taken between each injection cycle. Digital photos were taken using a Nikon D40x 10.2 megapixel DSLR camera that was mounted in the same way as the IR camera. Surface lighting was provided by two fluorescent lamps mounted at opposing angles. To reduce surface glare, a polarizing filter was used with the Nikon D40x. The digital photographs were post-processed using ImageJ [79] software to determine the endwall area fraction covered by wax deposits after each injection.

Figure 5.6 illustrates surface photos at different stages in the post-processing procedure to capture the deposit sizes starting with an image of the surface without deposition and an image of the surface with deposition. For the post-processing, each digital image was converted to an 8-bit image to which each pixel was assigned a gray value between 0 and 255 corresponding to its brightness. After an image was converted to 8-bit, it could be treated as a two-dimensional matrix. The background was then subtracted by taking the difference between an image of the surface with deposition and an image of the surface without deposition. The resulting image showing only pixels representing deposition was then converted to a binary image. Conversion to binary was accomplished by setting pixels with low gray values to zero and pixels with high gray values to one resulting in a black and white picture in which black pixels represented deposits. At this point it was necessary to remove any background noise that was not eliminated by the initial background subtraction. To do this, a median filter was used which effectively removed any pixel grouping that was smaller than a user-defined pixel value. Then, the deposition area fraction was represented by the ratio of black to white pixels.
Leading Edge Study

Experiments conducted in the leading edge study were performed to determine the effects of solid and molten sand deposition on showerhead film-cooling. To simulate sand deposition, atomized wax was injected using the wax particle generator illustrated in Figure 5.7. A molten wax supply was stored in a heated reservoir and compressed air was used to pressurize the wax reservoir and supply atomizing air. Liquid wax and atomizing air were supplied to a nozzle installed in the turbulence grid located 2 m upstream of the cylindrical test piece.

The Stokes number was used to scale the particle inertial characteristics from the engine to the laboratory model. The Stokes number, defined in Section 3.3, is the non-dimensional inertial response time of a particle. Recall, particles having high Stokes numbers are ballistic and less likely to follow streamlines around an obstacle than small particles with low Stokes numbers. An analysis similar to the one conducted for the flat plate study was performed to determine that wax particles with a median diameter of 175 µm in the wind tunnel test conditions would match the Stokes numbers of sand particles with a median size of 8 µm that exist in aero engine conditions. To control the wax particle size distribution that could be generated using the system in Figure 5.7, the atomizing air pressure and liquid wax pressure could be independently adjusted. The liquid wax pressure determined the flowrate of wax while the atomizing air pressure could be adjusted independently to control the size of the generated particles.

The Malvern particle analyzer was again used to measure the size distribution of generated wax particles. To generate particles in the size range of 175 µm, an atomizing air pressure of 70 kPa (10 psi) and a liquid wax pressure of 380 kPa (55 psi) were used. Figure 5.8 shows particle size distributions generated through a range of liquid wax pressures at an atomizing air pressure of 70 kPa. Measurements showed that for a given atomizing air pressure, the size distribution of particles increased with an increase in liquid wax pressure.

Results from the literature have shown that the phase of the particle upon impacting a surface is the most important characteristic that determines whether or not a particle will deposit on that surface. To scale the thermal characteristics of sand particles in the engine to wax particles in the lab the TSP, discussed in Section 3.3, was used. Recall, the TSP is the dimensionless solidification time of a particle immersed in a flow assuming the particle is a
lumped mass and that it is traveling at the same velocity as the surrounding flow. The solidification time of the particle is normalized by the time it takes the particle to travel from the injection location to the surface of interest. In an engine, the travel time would be equivalent to the time it takes a particle to travel from the combustor to the nozzle guide vane. Recall, particles having TSP values less than one are in solid form upon reaching the nozzle guide vane while particles having TSP values greater than one are in molten form upon reaching the nozzle guide vane.

The TSP is highly sensitive to particle diameter, so the TSP of the median particle size was used to characterize the particle phase. In laboratory conditions, TSP is sensitive to particle injection temperature, particle solidification temperature, and mainstream flow temperature all of which can be adjusted independently to achieve the necessary TSP to match engine conditions. Microcrystalline wax with a solidification temperature of 351 K was used to simulate deposition. Table 5.2 shows the particle material properties, operating conditions, and particle scaling parameters for sand particles in the actual engine compared to wax particles in the laboratory wind tunnel.

Following each deposition simulation, photographs were taken to further analyze the deposition area coverage. In addition to measuring surface area coverage, deposition thickness measurements were made at various spanwise and circumferential locations on the cylindrical surface. Deposition thickness measurements were made using a depth gage typically accurate to within 2 mm (0.16d). The resulting surface with deposition was painted with flat black paint to create uniform surface emissivity so that surface temperatures could be measured accurately using IR thermography as described in Section 5.1.

**Vane Cascade Study**

For the study on endwall film-cooling with and without transverse trenches, deposition was simulated dynamically using a two nozzle wax injection system with both nozzles located at 33% span as illustrated in Figure 5.9. Similar to the one-nozzle system illustrated in Figure 5.7, a stream of liquid wax was injected through the center of each nozzle head while two atomizing air jets aimed toward the wax stream served to break up the liquid wax into a mist of wax particles. The turbulence grid was located 3.6 chord lengths upstream of the vane row allowing adequate distance for the injected particles to be evenly
distributed across the width of the vane row. Deposition patterns on the endwall were periodic across the span of the test section.

To simulate the aerodynamic properties of fly ash particles in an engine, it was necessary to match the Stokes number range between the laboratory and engine environments. A method similar to that used in the flat plate and leading edge studies was used to determine that wax particles in simulation conditions must be 10 times larger than fly ash particles in engine conditions to achieve the same particle trajectory as shown in Figure 5.10. Therefore, wax particles between 1 µm and 100 µm were required to simulate particle trajectories of 0.1 µm to 10 µm fly ash particles that exist in engine conditions.

The Malvern particle analyzer was again utilized to measure the size distribution of particles generated using the wax injection system. For a liquid wax pressure of 138 kPa (20 psi) that resulted in a wax flow rate of 1.9 g/s from each nozzle, particle sizes were measured for various atomizing air pressures. Figure 5.11 shows the particle size distribution for a liquid wax flow rate of 1.9 g/s at atomizing air pressures of 69 kPa (10 psi), 207 kPa (30 psi), 276 kPa (40 psi), and 414 kPa (60 psi). Increasing atomizing air pressure increased air jet velocity which broke up the liquid stream and decreased particle sizes. To achieve particle sizes between 1 µm and 100 µm necessary to match Stokes number, the atomizing air pressure was set to 276 kPa with a liquid flowrate of 1.9 g/s from each nozzle for all deposition simulation experiments.

In addition to simulating particle trajectories, it was also desirable to simulate the thermal properties of particles that exist in engine conditions. Recall, the particle phase (solid or liquid) is of particular importance to deposition; therefore, it was necessary to scale the phase of fly ash as it exists in engine conditions. The TSP was again used for the endwall studies to scale the particle phase upon reaching the test section. For the endwall studies, the TSP of the maximum particle size, TSP\text{max}, was used to characterize the particle phase. By ensuring that the TSP\text{max} was matched between different experiments, the phase of the particle upon reaching the test section could be consistent.

Following the deposition simulation, a thin coat of flat black paint was applied to the wax-covered surface to ensure uniform surface emissivity with a value close to one. The IR data was then calibrated as described in Section 5.1 to account for any changes in emissivity.
caused by the deposition. After paint application, an adiabatic effectiveness test was conducted to determine the effects of the existing deposition on cooling.

For all the endwall studies conducted with no geometric modifications, Microcrystalline wax with a solidification temperature of 351 K was injected during steady state at either a mainstream temperature of 295 K to achieve a $TSP_{\text{max}} = 0.3$ or a mainstream temperature of 337 K to achieve a $TSP_{\text{max}} = 1.2$. For the endwall studies conducted with various trench depths, experiments were conducted at $TSP_{\text{max}}$ values of 1.2 and 2.2 to determine the effects of particle phase on deposition and the resulting cooling effectiveness. To achieve $TSP_{\text{max}} = 1.2$, a wax with a solidification temperature of 351 K was used. Wax with a solidification temperature of 333 K was used to achieve $TSP_{\text{max}} = 2.2$. As previously stated, wax was injected from two nozzles with a mass flowrate of 1.9 g/s per nozzle amounting to a total wax mass flowrate of 3.8 g/s for a duration of 240s. A wax mass flowrate of 3.8 g/s for a duration of 240s results in a particle loading of 56 ppmwh (parts per million by weight-hr) which is experienced by 8000 h of gas turbine operation with a hot gas path particulate concentration of 0.007 ppmw. Table 5.3 shows the particle material properties, operating conditions, and particle scaling parameters for fly ash particles in a gas turbine compared to wax particles in the laboratory wind tunnel for the vane cascade experiments.

Surface deposition was quantified using a two-dimensional area coverage technique similar to that described for the flat plate studies. To quantify the surface coverage, the endwall surface was photographed using a Nikon D40x 10.2 megapixel DSLR camera with a polarizing filter. For each experiment, diffuse surface lighting was created by placing a white sheet over the test section and providing upward lighting from inside the wind tunnel. By directing the lights toward the white sheet, light could be diffusely reflected evenly to prevent glare and uniformly light the surface. Care was taken to ensure that lighting conditions were consistent for every experiment.

After each experiment, photographs were taken through four port locations on the ceiling of the test section. The photos were then stitched together to create a composite image of the surface. The composite image was then cropped and converted to an eight bit image in which every pixel had a gray value between 0 and 255 representing the light intensity of that pixel. The white wax that deposited on the black surface created excellent
contrast and easy deposition identification. The 8 bit surface image was then converted to a binary image in which all black pixels represented deposition. Deposition area coverage could then be calculated by counting the ratio of black to white pixels for a given area of interest. ImageJ [79] software was utilized to perform the digital image processing described above. Figure 5.12 shows a composite endwall photo taken at the vane leading edge along with its corresponding 8 bit composite image and binary representation.

**Blade Cascade Study**

For the study on endwall film-cooling with and without a non-axisymmetric contour, deposition was simulated in the blade cascade using the same two-nozzle injection system used for the vane cascade experiments illustrated in Figure 5.9. For the experiments in the blade cascade study, the turbulence grid was located 9.1 chord lengths upstream of the blade cascade.

A Stokes analysis was performed to determine that a median wax particle size of 34 µm was necessary to match the Stokes number for fly ash particles with a median size of 5 µm in the engine. All blade cascade deposition simulations were conducted using wax with $T_{p,s} = 333$ K injected at $T_{\infty} = 320$K to achieve a $T_{\text{SP}_{\text{med}}} = 0.3$ or $T_{\infty} = 330$ K to achieve a $T_{\text{SP}_{\text{med}}} = 1.1$. For the wax with $T_{p,s} = 333$ K, a liquid wax pressure of 103 kPa (15psi) was set to achieve a wax flowrate of 1.9g/s from each nozzle. For that liquid wax pressure, an atomizing air pressure of 138 kPa (20 psi) was set to achieve a median particle size of 34 µm as measured by the Malvern particle analyzer. The particle properties and scaling parameters for the blade cascade experiments are shown in Table 5.4.

Surface photographs were taken through the port locations used to acquire IR data. The surface photographs were obtained using the same method described in the vane cascade study using a Nikon D40x 10.2 megapixel DSLR camera. These photographs were then stitched together to obtain a composite image of the endwall passage of interest. The composite images were then qualitatively compared with adiabatic effectiveness contour plots for each corresponding case.
5.3 Flow Visualization Methods

Flow visualization experiments were conducted using high speed images of wax particles immersed in the flowfield. A 2000 Hz CMOS camera with 1024 x 1024 pixel resolution was used to capture images while a Nd:YLF pulsed laser was used as a light source. The laser pulse frequency used for these experiments was approximately 6000 Hz. By pulsing the laser at three times the rate of the camera speed, multiple exposed images were acquired for improved flow visualization. The laser was mounted on top of the test section to illuminate the stagnation plane, indicated in Figure 4.8b. To capture images of particles passing through the stagnation plane, the camera was placed approximately 10° from normal to the stagnation plane at a distance of approximately 0.6 m. Successive images were then summed together to create composite images that illustrated particle trajectories of individual particles as they interacted with secondary flow structures near in the stagnation plane. Flow visualization experiments were conducted to observe particle interaction with endwall cooling near the leading edge with and without a 0.8d transverse trench.

5.4 Computational Methods

Computational simulations were performed by Lynch [80] using the discrete phase model (DPM) in the commercial computational fluid dynamics software FLUENT to predict particle tracks and accretion rates on the flat and contoured endwalls in the blade cascade. The computational domain used for the study was developed by Lynch et al. [10] and is shown in Figure 5.13. The SST k-ω model [81] was used for closure of the Reynolds-Averaged Navier-Stokes equations.

Particles were injected with the DPM using a surface injection at the inlet to the domain. A Rosin-Rammler diameter distribution was used to model the particle size distribution generated for the blade cascade experiments as measured by the Malvern particle analyzer. Figure 5.14 shows the Rosin-Rammler particle diameter distribution plotted along with the particle diameter distribution measured experimentally using the Malvern particle size analyzer.

The discrete random walk model was enabled to model the effect of turbulent velocity fluctuations on particle behavior. Lynch [80] incorporated user defined functions in the FLUENT DPM to the effectively trap any particle that passed within $y^+ = 1$ of the
endwall or blade surfaces. The trapped particles were then counted toward the accretion rate for the cell in which they collided with the surface. The accretion rate is defined as the mass flowrate per unit area of trapped particles. Figure 5.15 shows accretion rates predicted by the DPM in fluent for the flat and contoured endwalls in the blade cascade.

Figure 5.15a shows that particle accretion on the flat endwall is relatively uniform throughout the passage with the exception of areas in close proximity to cooling hole exits. In contrast to the flat endwall accretion rates, the accretion rates on the contoured endwall shown in Figure 5.15b suggest that particles are more likely to deposit in regions of high elevation on the endwall. It is interesting to note the difference in accretion rates in close proximity to film-cooling holes between the flat endwall and contoured endwall. The predicted results in Figure 5.15 suggest that the contour prevents deposition around film-cooling holes which could effectively mitigate the negative effects of deposition on cooling. The discussion in Section 8.3 describes the results from experiments conducted to determine the effects of endwall contouring on film-cooling with deposition. The comparison between experimental results and the computational results presented here is discussed in Section 8.3.

5.5 Uncertainty Analyses

Uncertainty analyses were performed for each set of experiments conducted using the propagation method described by Moffat [82]. Separate analyses were performed for the flat plate studies, the leading edge studies, the endwall studies conducted in the vane cascade test section, and the endwall studies conducted in the blade cascade test section. Detailed discussions of each uncertainty analysis along with sample calculations are provided in Appendix A. The uncertainties for each corresponding experiment are tabulated in Appendix A.
### Table 5.1. Stokes Number Conditions for Flat Plate Experiments

<table>
<thead>
<tr>
<th>Condition</th>
<th>Engine Conditions</th>
<th>Simulation Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas Temperature, $T_\infty$ (K)</td>
<td>1590 [70]</td>
<td>315</td>
</tr>
<tr>
<td>Compressor Pressure Ratio, PR</td>
<td>16 [70]</td>
<td>-</td>
</tr>
<tr>
<td>Particle Density, $\rho_p$ (kg/m$^3$)</td>
<td>1980 [8]</td>
<td>900</td>
</tr>
<tr>
<td>Gas Velocity, $U_\infty = U_p$ (m/s)</td>
<td>501 [83]</td>
<td>8</td>
</tr>
<tr>
<td>Gas Viscosity, $\mu$ (kg/m-s)</td>
<td>5.549x10$^{-5}$</td>
<td>1.852x10$^{-5}$</td>
</tr>
<tr>
<td>Chord, $C$ (cm)</td>
<td>21.8 [42]</td>
<td>-</td>
</tr>
<tr>
<td>Chord/Hole Diameter, $C/d$</td>
<td>129</td>
<td>-</td>
</tr>
<tr>
<td>Hole Diameter, $d=L_c$ (cm)</td>
<td>0.169</td>
<td>0.635</td>
</tr>
</tbody>
</table>

### Table 5.2. Particle Properties and Scaling Parameters for Leading Edge Experiments

<table>
<thead>
<tr>
<th>Property</th>
<th>Engine</th>
<th>Lab</th>
</tr>
</thead>
<tbody>
<tr>
<td>Median particle diameter, $d_p$ ($\mu$m)</td>
<td>8 [11]</td>
<td>175</td>
</tr>
<tr>
<td>Particle density, $\rho_p$ (kg/m$^3$)</td>
<td>2650 [84]</td>
<td>800</td>
</tr>
<tr>
<td>Particle initial temperature, $T_{p,i}$ (K)</td>
<td>2594 [11]</td>
<td>364</td>
</tr>
<tr>
<td>Particle specific heat, $C_p$ (J/kgK)</td>
<td>420 [84]</td>
<td>2090</td>
</tr>
<tr>
<td>Particle latent heat of fusion, $\Delta h_{fus}$ (J/kg)</td>
<td>142000 [84]</td>
<td>225600</td>
</tr>
<tr>
<td>Particle solidification temperature, $T_s$ (K)</td>
<td>1972 [84]</td>
<td>351</td>
</tr>
<tr>
<td>Particle solidification time, $t_s$ (s)</td>
<td>1.25 x 10$^{-4}$</td>
<td>0.17</td>
</tr>
<tr>
<td>Particle velocity, $U_\infty$ (m/s)</td>
<td>140 [11]</td>
<td>6.68</td>
</tr>
<tr>
<td>Particle travel distance, $L_\infty$ (m)</td>
<td>0.147 [11]</td>
<td>2.05</td>
</tr>
<tr>
<td>Thermal scaling parameter, TSP</td>
<td>0.12</td>
<td>1.0/2.0</td>
</tr>
<tr>
<td>Stokes number, $Stk$</td>
<td>4.00</td>
<td>4.00</td>
</tr>
</tbody>
</table>
### Table 5.3. Particle Properties and Scaling Parameters used for Vane Cascade Experiments

<table>
<thead>
<tr>
<th></th>
<th>Engine (Fly Ash)</th>
<th>Laboratory (Wax1/Wax2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Particle Diameter, (d_p) (µm)</td>
<td>0.1 - 10 [9]</td>
<td>1 - 100</td>
</tr>
<tr>
<td>Particle Density, (\rho_p) (kg/m³)</td>
<td>1980 [9]</td>
<td>800</td>
</tr>
<tr>
<td>Specific Latent Heat of Fusion, (\Delta h_{ fus}) (J/kg)</td>
<td>650000 [67]</td>
<td>225600</td>
</tr>
<tr>
<td>Specific Heat, (C_p) (J/kg-K)</td>
<td>730 [68]</td>
<td>2090</td>
</tr>
<tr>
<td>Particle Softening Temperature, (T_{ p,s}) (K)</td>
<td>1533 [69]</td>
<td>351/333</td>
</tr>
<tr>
<td>Mainstream Gas Temperature, (T_{ \infty}) (K)</td>
<td>1500 [70]</td>
<td>337/325</td>
</tr>
<tr>
<td>Particle Initial Temperature, (T_{ p,i}) (K)</td>
<td>1593 [70]</td>
<td>364</td>
</tr>
<tr>
<td>Gas Viscosity, (\mu) (kg/m-s)</td>
<td>5.55 x 10⁻⁵</td>
<td>1.82 x 10⁻⁵</td>
</tr>
<tr>
<td>Particle Travel Distance, (L_{ \infty}) (m)</td>
<td>0.26</td>
<td>2.34</td>
</tr>
<tr>
<td>Particle Velocity (Mainstream Velocity), (U_{ \infty}) (m/s)</td>
<td>93 [60]</td>
<td>6.3</td>
</tr>
<tr>
<td>Film Cooling Hole Diameter, (d) (mm)</td>
<td>0.5</td>
<td>4.6</td>
</tr>
<tr>
<td>Maximum Thermal Scaling Parameter, (T_{ SP_{max}})</td>
<td>1.2</td>
<td>1.2/2.2</td>
</tr>
<tr>
<td>Stokes number, (Stk)</td>
<td>0.004 - 40</td>
<td>0.004 - 40</td>
</tr>
</tbody>
</table>

### Table 5.4. Particle Properties and Scaling Parameters used for Blade Cascade Experiments

<table>
<thead>
<tr>
<th></th>
<th>Engine (Fly Ash)</th>
<th>Laboratory</th>
</tr>
</thead>
<tbody>
<tr>
<td>Median Particle Diameter, (d_p) (µm)</td>
<td>5</td>
<td>35</td>
</tr>
<tr>
<td>Particle Density, (\rho_p) (kg/m³)</td>
<td>1980 [9]</td>
<td>800</td>
</tr>
<tr>
<td>Specific Latent Heat of Fusion, (\Delta h_{ fus}) (J/kg)</td>
<td>650000 [67]</td>
<td>225600</td>
</tr>
<tr>
<td>Specific Heat, (C_p) (J/kg-K)</td>
<td>730 [68]</td>
<td>2090</td>
</tr>
<tr>
<td>Particle Solidification Temperature, (T_{ p,s}) (K)</td>
<td>1533 [69]</td>
<td>333</td>
</tr>
<tr>
<td>Mainstream Gas Temperature, (T_{ \infty}) (K)</td>
<td>1500 [70]</td>
<td>321/330</td>
</tr>
<tr>
<td>Particle Initial Temperature, (T_{ p,i}) (K)</td>
<td>1593 [70]</td>
<td>357</td>
</tr>
<tr>
<td>Gas Viscosity, (\mu) (kg/m-s)</td>
<td>5.55 x 10⁻⁵</td>
<td>1.82 x 10⁻⁵</td>
</tr>
<tr>
<td>Particle Travel Distance, (L_{ \infty}) (m)</td>
<td>0.25</td>
<td>2.18</td>
</tr>
<tr>
<td>Particle Velocity (Mainstream Velocity), (U_{ \infty}) (m/s)</td>
<td>63 [10]</td>
<td>10.4</td>
</tr>
<tr>
<td>Film Cooling Hole Diameter, (d) (mm)</td>
<td>0.51</td>
<td>4.4</td>
</tr>
<tr>
<td>Median Thermal Scaling Parameter, (T_{ SP_{med}})</td>
<td>0.76</td>
<td>0.3/1.1</td>
</tr>
<tr>
<td>Median Particle Stokes number, (Stk)</td>
<td>6.54</td>
<td>6.54</td>
</tr>
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</table>
Figure 5.1. Photographs of (a) sand deposition and (b) wax deposition on a cylinder.

Figure 5.2. Wax particle diameter required to match engine condition Stokes number for the flat plate study.

Figure 5.3. ESEM images of (a) coal ash [9] and (b) wax particles.
Figure 5.4. Wax particle size distribution and corresponding Stokes number in lab simulation conditions for the flat plate study.

Figure 5.5. Wax particle temperature relative to travel distance for the flat plate study.
Figure 5.6. Photographs illustrating the surface (a) without deposition, (b) with deposition, (c) after background subtraction, (d) after binary conversion, and (e) after median filter.

Figure 5.7. Wax particle generator and photograph of injection nozzle used for the leading edge study.
Figure 5.8. Particle size histograms of $T_s = 351$ K wax at an atomizing air pressure of 70kPa and various liquid wax pressures.

Figure 5.9. (a) Schematic of wax injection facility and (b) photograph of wax spray nozzle used in the leading edge study.
Figure 5.10. Wax particle size range necessary to match Stokes numbers of fly ash particles in engine conditions for the vane cascade studies.

\[ \text{Stk} = \frac{\rho_p d_p^2 U_p}{18 \mu L_c} \]

Figure 5.11. Particle size histograms at a wax flowrate of 1.9 g/s ($P_{w}=138$ kPa) and various atomizing air pressures for the injection system used for the vane cascade studies.
Figure 5.12. Vane cascade endwall photographs illustrating the (a) composite endwall photograph along with corresponding (b) 8bit and (c) binary images.

Figure 5.13. Depictions of (a) the computational domain and boundary conditions; (b) the flat endwall grid; and (c) the contoured endwall grid [10].
Figure 5.14. Particle size distributions as measured by the Malvern particle analyzer and as modeled using the Rosin-Rammler method.

Figure 5.15. Accretion rates predicted by the discrete phase model in FLUENT for the blade cascade (a) flat endwall and (b) contoured endwall.
CHAPTER 6:
Flat Plate Deposition Studies

This chapter describes the studies that were conducted to determine the effects of simulated deposition on flat plate film-cooling effectiveness. The article by Lawson and Thole [85] was approved by reviewers to be included in the ASME Turbo Expo 2009 Gas Turbine Technical Congress and Exposition presented by the International Gas Turbine Institute. The publication received a best paper award by the IGTI Heat Transfer (K-14) Committee and was accepted for publication in the *ASME Journal of Turbomachinery*. The paper as published in the online version of the *ASME Journal of Turbomachinery* is shown in Appendix C.

The objective of the flat plate study was to show that the effects of deposition on film-cooling adiabatic effectiveness could be quantified using IR thermography. The results of the flat plate study showed that deposition was highly dependent on whether particles were in solid or molten form upon impaction with a surface and that the phase of the particle was dependent on particle size range and surrounding gas temperatures. Deposition was dynamically simulated near film-cooling holes at three momentum flux ratios. Results showed that deposition reduced cooling effectiveness more at low momentum flux ratios than at high momentum flux ratios.

Section 6.1 describes the validation of the IR thermography measurement technique. The effects of simulated deposition on cooling effectiveness are described in detail in Section 6.2. A summary of the major findings from the flat plate studies is presented in Section 9.1. The test matrix for the flat plate study is shown in Appendix B, Table B.1.

### 6.1 Validation of Measurement Technique

To validate the measurement technique, adiabatic effectiveness for a single row of cylindrical film-cooling holes was measured at \( \text{Tu} = 4.6\% \) and \( \text{Tu} = 12.3\% \). Figure 6.1 shows the centerline effectiveness plotted with respect to dimensionless downstream distance for a row of cooling holes having a momentum flux ratio of 0.23. The \( \text{Tu} = 4.6\% \) case was measured to compare with literature data and the \( \text{Tu} = 12.3\% \) case was measured as a baseline for deposition studies. Both cases are shown in Figure 6.1. Note that the measurements in Figure 6.1 were made on an endwall without deposition. Table 6.1 shows
the test conditions for the literature results used for comparison in Figure 6.1. For the present cases, centerline effectiveness increases with a decrease in freestream turbulence. The $\text{Tu} = 4.6\%$ case converges with most of the literature between 4d and 12d downstream of the cooling row, while the $\text{Tu} = 12.3\%$ case falls below most of the literature and is in agreement with the data by Kunze et al. [86]. The range of data that could be acquired was limited to 12d downstream of the cooling holes because of IR access constraints.

Figure 6.2 shows the laterally averaged effectiveness with respect to dimensionless downstream distance compared with results from the literature. In contrast to the centerline data, the laterally averaged effectiveness decreases with a decrease in freestream turbulence. The $\text{Tu} = 4.6\%$ results agree reasonably well with Lutum and Johnson [87], Schmidt et al. [88] and Kunze et al. [86], but are higher than the results obtained by Sinha et al. [18]. Lutum and Johnson [87] studied the effect of cooling hole $\text{L/d}$ on adiabatic effectiveness and found that effectiveness generally decreased with decreasing $\text{L/d}$. This is a probable reason for why the laterally averaged results by Sinha et al. [18] are lower than the rest of the benchmark data. Differences between the benchmark case at $\text{Tu} = 4.6\%$ and the data from the literature exist possibly because of the difference in freestream turbulence levels measured between the present case and the studies in the literature. High freestream turbulence levels could enhance mixing between the jet and the mainstream and therefore increase lateral spreading of the coolant.

### 6.2 Effects of Deposition on Flat Plate Film-Cooling

In their study evaluating the independent effects of density ratio, blowing ratio, and momentum flux ratio on film-cooling, Thole et al. [19] concluded that a coolant jet will either remain attached, separate and reattach, or separate and remain detached from the endwall depending on momentum flux ratio. In the present study, adiabatic effectiveness tests were conducted at momentum flux ratios of $I = 0.23$, $I = 0.5$, and $I = 0.95$ to observe the effects of deposition for the three different regimes described by Thole et al. [19]. Table 6.2 shows the conditions for each adiabatic effectiveness test conducted for the deposition study.

Figure 6.3 shows contour plots of adiabatic effectiveness for each momentum flux ratio tested prior to deposition. The contours illustrate the effectiveness characteristics of each flow regime. As momentum flux ratio increases from 0.23 to 0.5, laterally averaged
effectiveness decreases implying that separation may be occurring at I = 0.5. When momentum flux ratio is increased to 0.95, the contours illustrate low effectiveness laterally and along the centerline implying that separation is occurring. Figure 6.4 shows centerline and laterally averaged effectiveness with respect to dimensionless downstream distance for all three momentum flux ratios. Figure 6.4 shows that centerline and laterally averaged effectiveness decreases with an increase in momentum flux ratio when separation occurs.

Figure 6.5 shows effectiveness contour plots and corresponding deposition photographs at I = 0.23. The appearance of the deposition in each photograph depends on the state of a particle when it impacts the surface. The deposits caused by molten particles show up as black splats while deposits caused by solid particles show up as white spots. For the initial injection, only molten particles appear to deposit on the surface. Eventually, molten deposition increases the surface roughness to a point that solid particulate begins to deposit. As more injections occur, solid and molten particles deposit and further increase surface roughness. Although endwall temperatures were measured after all eight injection cycles, the effects of deposition are well represented by the data taken after 1200 g (3 cycles), after 2400 g (6 cycles), and after 3200 g (8 cycles) as seen in Figure 6.5. Close inspection of each deposition photograph reveals that deposition concentration appears to be lower directly downstream of the film-cooling holes implying that the coolant jets themselves help prevent deposition. It can also be seen that solid particles build up over time inside the trailing edge of the film-cooling holes.

Figure 6.6 shows the centerline effectiveness at I = 0.23 after each wax injection cycle. Centerline effectiveness is affected by deposition after the first three to four injection cycles and begins to reach a steady level between five and eight injection cycles. This pattern supports the observation made by Walsh et al. [38] that the deposition process eventually reaches an equilibrium state where the rate of deposition equals the rate that existing deposits are eroded by incoming particles that do not stick. Figure 6.7 shows the laterally averaged effectiveness after each wax injection cycle for I = 0.23. As deposition increases, laterally averaged effectiveness decreases at all downstream locations. Deposition appears to have a greater impact on effectiveness in regions within 3d of the cooling hole trailing edge than in regions farther downstream.
The deposition development was quantified by counting deposits using ImageJ [79] software. For a binary image, the software counts the pixel groupings and categorizes them based on area coverage. An equivalent deposit diameter was then calculated for each pixel grouping. Figure 6.8 shows a histogram of deposit size as it developed with increased injection for I = 0.23. It is important to note that Figure 6.8 does not show the histogram of particle sizes but rather deposit sizes. Each deposit could be made up of an agglomeration of many individual particles. The figure shows that as more wax is injected, both the size and number of deposits increase because of new particles adhering directly on the surface and onto existing deposits. The histograms indicate that the amount of deposit on the surface approaches an equilibrium state after eight injection cycles (3200g). Histograms of deposit size for I = 0.5 and I = 0.95 showed a similar development trend as the histogram for I = 0.23 implying that the time to reach a deposition equilibrium state is independent of momentum flux ratio.

Figure 6.9 shows the effectiveness contour plots and corresponding deposition photographs at I = 0.5. The well defined wake region indicates that deposition is mitigated by the coolant jets immediately downstream of the film-cooling holes. There also appears to be more deposition of solid particles at I = 0.5 than at I = 0.23 which could be caused by particles solidifying in the coolant layer prior to impacting the surface. This cooling effect would be enhanced by the higher coolant flow at I = 0.5. The contour plots show that effectiveness decreases with an increase in deposition. From the contours it is evident that deposition shortens the downstream influence of each coolant jet.

Figure 6.10 shows the effectiveness contour plots and corresponding deposition photographs for I = 0.95. Similar to the two previous cases, the photographs show that deposition in regions near the cooling hole trailing edge are mitigated by the coolant jet itself. Deposition in these near wake regions is mitigated for two reasons. First, the coolant jets cool the large molten particles causing them to solidify prior to impacting the surface. Second, the trajectories of the small solid particles are greatly influenced by the coolant jets preventing them from impacting the surface in near wake regions. The only particles that deposit in near wake regions are particles that are large enough to maintain their trajectories and withstand solidification by coolant jets.
Figure 6.10 shows that deposition has less of an effect on cooling at I = 0.95 than it does at the lower momentum flux ratios. Decreased sensitivity to deposition at I = 0.95 occurs because the jet separation at I = 0.95 prevents high levels of adiabatic effectiveness downstream of the coolant holes even without deposition on the surface. Because of coolant jet separation at I = 0.95, deposition has less of an effect at that momentum flux ratio than for I = 0.5 or I = 0.23. From the contours, the deposition enhances jet interaction which ultimately improves lateral spreading of the coolant in far wake regions. The conduction effect of the wax is very small given it has a relatively low thermal conductivity (k = 0.24 W/m-K). Because the conduction effect can be neglected, the improved lateral spreading is attributed to the roughness created by large deposits in far wake regions rather than a conductive effect.

Figure 6.11 shows the area-averaged effectiveness reduction with respect to deposition area coverage. For I = 0.23 and I = 0.5, area-averaged effectiveness is reduced sharply through the first three injection cycles, and approaches 20% as deposition area coverage increases. On the other hand, at I = 0.95, the reduction in effectiveness reaches a maximum of 6% before actually improving to 2% after the final injection cycle. The reduction in effectiveness was low at I = 0.95, because the initial cooling provided by these separated jets was very low prior to deposition.

6.3 Summary of Flat Plate Deposition Findings

A methodology was developed to observe the effects of simulated deposition on film-cooling through the use of a low melting temperature wax injectant. Cooling effectiveness was quantified by obtaining spatially resolved temperatures to calculate adiabatic effectiveness in the vicinity of a row of film-cooling holes. The method for quantifying cooling effectiveness was validated by comparison with adiabatic effectiveness results from the literature.

Deposition was simulated by injecting molten wax particles into the mainstream air flow. Simulation conditions were controlled so that the size range of injected wax particles scaled with Stokes number to the size range of fly ash particles in engine conditions.

Deposition was simulated for three momentum flux ratios: I = 0.23, I = 0.5, and I = 0.95 to observe the effects of deposition on cooling for three different jet conditions. For all
momentum flux ratios, deposit coverage appeared to be less concentrated in near wake regions of each film-cooling hole. Deposition in near wake regions was prevented because coolant jets solidified large particles and changed the trajectories of small particles preventing both from adhering to the surface. Large particles in molten form were the only particles found to have deposited in the near wake of the jets.

As deposition developed, the deposit sizes and total number of deposits increased implying that incoming particles adhered to the surface creating new deposits and to existing deposits increasing their size. Based on deposit size histograms and reductions in adiabatic effectiveness, the deposition process seemed to approach an equilibrium state after eight injection cycles. For \( I = 0.23 \) and \( I = 0.5 \), the effectiveness was reduced by 20\% as deposition approached an equilibrium state. For \( I = 0.95 \) effectiveness reached a maximum reduction of 6\% and a final reduction after eight injection cycles of 2\%. There was a different effect on cooling at \( I = 0.95 \) because the coolant jets were separated from the surface prior to deposition which inhibited cooling even without the existence of deposits.
Table 6.1. Comparison of Film-Cooling Conditions

<table>
<thead>
<tr>
<th>Study</th>
<th>L/d</th>
<th>P/d</th>
<th>α (°)</th>
<th>I</th>
<th>M</th>
<th>VR</th>
<th>DR</th>
<th>Tu (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sinha et al. [18]</td>
<td>1.8</td>
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<td>35</td>
<td>0.21</td>
<td>0.50</td>
<td>0.42</td>
<td>1.20</td>
<td>0.2</td>
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<tr>
<td>Schmidt et al. [88]</td>
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<td>3.0</td>
<td>35</td>
<td>0.23</td>
<td>0.60</td>
<td>0.38</td>
<td>1.60</td>
<td>0.2</td>
</tr>
<tr>
<td>Lutum and Johnson [87]</td>
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<td>2.9</td>
<td>35</td>
<td>0.24</td>
<td>0.52</td>
<td>0.45</td>
<td>1.15</td>
<td>3.5</td>
</tr>
<tr>
<td>Kunze et al. [86]</td>
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<td>4.0</td>
<td>35</td>
<td>0.19</td>
<td>0.50</td>
<td>0.36</td>
<td>1.37</td>
<td>1.5</td>
</tr>
<tr>
<td>Present Case</td>
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<td>3.0</td>
<td>30</td>
<td>0.23</td>
<td>0.49</td>
<td>0.47</td>
<td>1.06</td>
<td>4.6/12.3</td>
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</table>

Table 6.2. Test Conditions for Deposition Tests

<table>
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<th>M</th>
<th>VR</th>
<th>DR</th>
<th>Tu (%)</th>
</tr>
</thead>
<tbody>
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<td>0.47</td>
<td>1.06</td>
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</tr>
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<td>1.07</td>
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<td>1.01</td>
<td>0.95</td>
<td>1.07</td>
<td>12.3</td>
</tr>
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</table>

Figure 6.1. Centerline effectiveness with respect to downstream distance for I = 0.23 with no deposition.
Figure 6.2. Laterally averaged effectiveness with respect to downstream distance for $I = 0.23$ with no deposition.

Figure 6.3. Effectiveness contour plots at momentum flux ratios of (a) 0.23, (b) 0.5, and (c) 0.95.
Figure 6.4. Centerline and laterally averaged effectiveness plotted with respect to dimensionless downstream distance for all momentum flux ratios.

Figure 6.5. Deposition development photographs and corresponding effectiveness contour plots at I = 0.23.
Figure 6.6. Centerline effectiveness development with deposition for \( I = 0.23 \).
Figure 6.8. Histograms of deposit sizes at different stages of deposition for $I = 0.23$.

Figure 6.9. Deposition development photographs and corresponding effectiveness contour plots at $I = 0.5$. 
Figure 6.10. Deposition development photographs and corresponding effectiveness contour plots at $I = 0.95$.

Figure 6.11. Area-averaged effectiveness reduction with respect to deposition area coverage for all three momentum flux ratios.
CHAPTER 7:
Leading Edge Deposition Studies

The work conducted to determine the effects of dynamically simulated deposition on leading edge film-cooling is described in Chapter 7. The results presented in this chapter were composed for a paper [89] that is currently under review for publication in the *ASME Journal of Turbomachinery* and ASME Turbo Expo: Turbine Technical Conference and Exposition.

Because the leading edge of the airfoil is the region most susceptible to the effects of deposition, a test matrix was developed to test the effects of deposition on film-cooling at two TSP values and three blowing ratios. To determine the effects of deposition on film-cooling, experiments were first conducted to determine the quantity of injected wax necessary for the deposition to reach an equilibrium state. Recall from the Chapter 6 that the effects of deposition on a film-cooled flat plate reached an equilibrium state at which point additional deposition buildup on the surface no longer affected cooling effectiveness. Albert et al. [52] observed that deposition reached an equilibrium state on a leading edge model. For the current study, multiple tests with different wax injection durations were conducted at $M = 1.0$ to determine the proper injection duration to reach the equilibrium state. It is important to note that for each condition tested, a deposition simulation test as well as an adiabatic effectiveness measurement experiment was conducted. The test matrix for the leading edge study is shown in Appendix B, Table B.2.

7.1 Baseline Effectiveness Results

Prior to deposition simulation, a set of baseline adiabatic effectiveness experiments were conducted. Figure 7.1 shows the baseline adiabatic effectiveness contour plots at each blowing ratio tested. An interesting finding from the baseline experiments is that the row of holes closest to stagnation at s/d = 1.2 ejects no coolant at the lowest blowing ratio. There is no coolant ejecting from the s/d = 1.2 row at $M = 0.5$ because of the high pressure region near stagnation. The s/d = 1.2 hole locations are illustrated by the blue ellipses in Figure 7.1a. As the blowing ratio was increased to $M = 1.0$ coolant began to exit the s/d = 1.2 row and provided good coolant coverage around the circumference of the leading edge. As the blowing ratio was increased further to $M = 1.8$, the coolant jets separated from the surface.
and the total surface coolant coverage decreased. Table 7.1 lists the local blowing ratios at each row of holes for each cooling condition.

Figure 7.2 shows the spanwise-averaged cooling effectiveness relative to circumferential location (s/d) near the leading edge showerhead region. The location of the center of each cooling row is shown by dashed lines in Figures 7.1a and 7.2. The general trend of the data shows that peak effectiveness increased with an increase in blowing ratio; however, because the coolant jets are separated at M = 1.8, the effectiveness between holes was highest at M = 1.0. The exception to these trends is the row of holes located at s/d = -6. For the row at s/d = -6, the blowing ratio exhibiting the highest peak effectiveness as well as the highest surrounding effectiveness was M = 0.5. At s/d = -6.0, the local blowing ratios at M = 0.5, 1.0, and 1.8 were $M_L = 1.2, 1.4$, and 2.0 respectively. The highest effectiveness probably occurred at M = 0.5 ($M_L = 1.2$) because separation effects reduced effectiveness as the local blowing ratio exceeded 1.2. The same thing happened at the s/d = 4.4 row. When the local blowing ratio exceeded 1.2 effectiveness decreased indicating separation.

Also shown in Figure 7.2 is the data reported by Polanka et al. [20] for M = 0.5 and M = 1.8 at Tu = 2.2%. Similar to the data for the current study, Polanka et al. [20] showed an increase in cooling effectiveness with an increase in blowing ratio; however, they did not observe separation at M = 1.8 because their cooling holes had a more shallow injection angle of 25° as compared to the 40° injection angle in the current study. Polanka et al. [20] also had a cooling row at stagnation that prevented the local minimum in effectiveness observed in the current study. Polanka et al. [20] measured higher effectiveness between cooling rows than the current study which can also be attributed to differences in geometric specifications such as injection angle.

### 7.2 Deposition Development on a Vane Leading Edge with Film-Cooling

The results from the deposition development experiments at M = 1.0 are shown in Figure 7.3. The photographs in Figure 7.3 show an asymmetric deposition pattern due to the asymmetric cooling hole pattern near the stagnation region. Deposition collected most densely around the cooling holes close to the stagnation region. In theory, the particles that deposit close to stagnation are the largest particles with the highest Stokes numbers and are therefore minimally affected by changes in the flowfield. These large particles are not
diverted by the cooling holes as effectively near stagnation as the small particles are by the cooling holes away from stagnation.

The contours in Figure 7.3 show how cooling effectiveness is affected by deposition as it develops at $M = 1.0$ and TSP = 1.0. The impact of deposition collection can be seen in the effectiveness contours. Comparing Figures 7.3a and 7.3d, the effectiveness downstream of the s/d = 1.2 row is reduced dramatically compared to the baseline because of cooling hole blockage due to deposition.

The spanwise-averaged effectiveness plots in Figure 7.4 show how effectiveness changed with increased deposition across the circumference of the leading edge region at $M = 1.0$. An increase in deposition caused a decrease in cooling effectiveness at all rows except the row located at s/d = 4.4. Although peak effectiveness improved with deposition near the hole exit at s/d = 4.4, the deposition near the hole trailing edge prevented coolant from spreading and reduced spanwise-averaged effectiveness downstream of the row. A theory for the increase in effectiveness with deposition at s/d = 4.4 is presented in Section 7.2.

Following each deposition test, the thickness of deposition was measured around the leading edge at three spanwise locations. Figure 7.5 shows the average measured thickness after 100g, 200g, and 300g at $M = 1.0$. The plot in Figure 7.5 shows that thickness is on the order of a cooling hole diameter, similar to Albert et al. [52], and increases linearly with wax injection up to 300g.

Area-averaged effectiveness for each case was calculated for the entire area shown in each contour. The area-averaged adiabatic effectiveness and effectiveness reduction caused by deposition are plotted relative to wax injection mass in Figure 7.6. The largest reduction in effectiveness was observed after just 200g of wax injection. Between 200g and 300g of injection, effectiveness reduction changed by less than 2%. The small change in effectiveness between 200g and 300g of wax injection justified the use of 300g of wax to capture the effects of deposition for the remaining experiments.

7.3 Effects of Blowing Ratio on Leading Edge Film-Cooling with Deposition

Deposition experiments were conducted at blowing ratios of $M = 0.5$, 1.0, and 1.8. Results from the literature show the effects blowing ratio on cooling effectiveness; however, the literature provides very few studies to help us understand how cooling may be affected by
deposition at different operating conditions. The relationship between deposition and film-cooling is a two-way coupled dependency. Not only is deposition dependent on the cooling condition, but the cooling effectiveness is dependent on the resulting deposition. Albert et al. [52] showed results to support this two-way coupling theory. They determined that deposition collected in areas downstream of cooling rows where heat transfer augmentation is typically high. They also observed deposition collection within and downstream of holes that affected the dynamics of the coolant jets. So not only was deposition affected by the coolant, the coolant was affected by the deposition.

Figure 7.7 shows adiabatic effectiveness contour plots before and after deposition with photographs of resulting deposition after 300g of wax injection for all three blowing ratios at TSP = 1.0. The dependence of deposition on blowing ratio can be seen clearly in the deposition photographs in Figure 7.7. As blowing ratio increased (left to right), the cooling hole blockage decreased. Decreased hole blockage with increased blowing occurred because jet velocity increased with increased blowing ratio and, therefore, was more effective at diverting particle trajectories. This effect of jet velocity on hole blockage is most obvious in the s/d = 1.2 row of holes where high external pressure prevented high jet velocities at low blowing ratios. Close-up images of deposition at mid-span for each blowing ratio are shown in Figure 7.8. The images in Figure 7.8 are close-ups of the regions indicated by the orange boxes in Figure 7.7c.

Deposition that occurred in the wakes of cooling holes is more complicated than deposition in the cooling holes. The wake deposition behavior can be most clearly observed downstream of the row located at s/d = -3.5 in Figure 7.8. The deposition that occurred in the wakes of the s/d = -3.5 row decreased with blowing ratio while the coolant jets were attached to the surface. As blowing ratio increased to M = 1.8 the coolant jets separated from the surface and deposition collected downstream of cooling holes in regions of high vorticity caused by the complex nature of the flow associated with coolant jet separation. The effect of jet separation on deposition is most obvious downstream of the s/d = -3.5 row in Figure 7.8c.

The effect of deposition on cooling is illustrated in the adiabatic effectiveness contour plots at each blowing ratio in Figure 7.7. Deposition collection in the s/d = 1.2 row of holes caused a blockage effect and decreased cooling effectiveness within and downstream of the
s/d = 1.2 row at all three blowing ratios. At M = 0.5, any cooling that did exist near the s/d = 1.2 row was eliminated by the complete blockage of that row illustrated in Figure 7.8a. At M = 1.0, the cooling effectiveness of the s/d = 1.2 row was decreased dramatically by dense collection of deposition within the s/d = 1.2 row. At M = 1.8, the effect of deposition within the s/d = 1.2 row was not as great as at lower blowing ratios because the high jet velocities prevented dense deposition within the holes. At M = 0.5 and M = 1.0 the blockage of the s/d = 1.2 row caused by deposition caused coolant to be re-routed to other cooling holes improving effectiveness at those locations. This improved effectiveness is most obvious near the row located at s/d = 4.4 in Figure 7.7b at M = 1.0. Increased coolant to the s/d = 4.4 row improved effectiveness close to the hole exits; however, deposition close to the hole exit of the s/d = 4.4 row prevented coolant from spreading downstream of that row as effectively as it did in the baseline case.

The direct effects of deposition on cooling can be seen in the local effectiveness reduction plot in Figure 7.9. It is important to note that effectiveness reduction was normalized with the area-averaged effectiveness for each corresponding baseline case. At M = 0.5, effectiveness reduction was negative in most locations indicating that deposition caused a slight improvement in effectiveness. The improvement can be seen at and downstream of the rows located at s/d = -6.0 and -3.5. At the row of holes located at s/d = 4.4, the peak cooling within the row improved due to deposition at M = 0.5 and M = 1.0; however, as described in the previous paragraph, deposition near the exits of the cooling holes prevented the coolant from spreading effectively downstream of the cooling row. The prevention of coolant spread at s/d = 4.4 caused a reduction in effectiveness downstream of the cooling holes at all three blowing ratios as shown in Figure 7.9. The maximum effectiveness reduction occurred at M = 1.0 and 1.8 just downstream of the s/d = 1.2 row. The maximum improvement in effectiveness occurred at M = 1.0 near the s/d = 4.4 row of holes.

Figure 7.10 shows the area-averaged effectiveness at each blowing ratio before and after deposition as well as the resulting effectiveness reduction caused by the deposition. Before deposition, area-averaged effectiveness increased with blowing ratio between M = 0.5 and M = 1.0 and reached a peak value of 0.28. As blowing ratio increased to M = 1.8 the coolant jets separated from the surface causing a decrease in area-averaged cooling
effectiveness to a value of 0.20. Deposition caused a 2.9% improvement in effectiveness at M = 0.5 while deposition reduced effectiveness by 13% at M = 1.0 and 20% at M = 1.8. Although baseline experiments showed that the maximum effectiveness occurred at M = 1.0 before deposition, the maximum effectiveness occurred at M = 0.5 after deposition. However, even though the area-averaged effectiveness was higher at M = 0.5 than at M = 1.0 after deposition, M = 1.0 may still be the best operating condition because it provided better cooling near stagnation with and without deposition as shown in Figure 7.7. Because stagnation is the region of highest heat transfer from the hot mainstream, cooling conditions should be designed to provide maximum cooling to that area.

The deposition thickness relative to circumferential location for every blowing ratio tested is presented in Figure 7.11. As discussed previously, deposition was prevented from collecting inside cooling holes at M = 1.8 because high jet velocities diverted particle trajectories. The data in Figure 7.11 suggests that the high jet velocities prevented deposition from collecting at all locations around the circumference of the leading edge, not just local to the cooling holes. Of the blowing ratios tested, the thickest deposition occurred at M = 1.0 near s/d = 2. Deposition was thickest downstream of the s/d = 1.2 row and between cooling holes. Thick ridges of deposition were formed between cooling holes because some particles that were diverted by coolant jets likely deposited between holes which is why the deposition layer was not as thick at M = 0.5 as it was at M = 1.0. At M = 0.5, the s/d = 1.2 row of cooling holes had little effect on approaching particles which resulted in a more uniform deposition layer at M = 0.5 than at M = 1.0. Because the deposition layer was most uniform at M = 0.5, the maximum thickness was not as great at M = 0.5 as it was at M = 1.0. This effect is evident when comparing the deposition photos in Figure 7.8a and 7.8b. The deposition ridge between holes in the s/d = 1.2 row can be seen in Figure 7.8b.

The fraction of total area covered by deposition was measured using a digital image analysis method described in Section 5.2. The deposition area coverage at M = 0.5, 1.0, and 1.8 were 79%, 77%, and 74% respectively. Deposition area coverage decreased with an increase in blowing ratio because the increase in jet velocity decreased the deposition likelihood of approaching particles. Although the deposition thickness was maximum at M = 1.0, the deposition area coverage was maximum at M = 0.5 and decreased with an increase in blowing ratio.
7.4 Effects of Particle Phase on Leading Edge Film-Cooling with Deposition

Deposition in engine conditions is highly dependent on factors such as combustion temperature and particle material properties that determine the phase of the particle upon impaction. Recall from the review of literature that deposition likelihood is highly dependent on the phase of a particle upon impaction of a surface. The thermal scaling parameter (TSP) described in detail in Section 3.3 was developed to scale the particle phase upon impaction. To test the effects of particle phase on deposition and cooling effectiveness, experiments with TSP = 1.0 and TSP = 2.0 were conducted at M = 1.0.

Figure 7.12 shows the cooling effectiveness contours and surface deposition photographs for the baseline case as well as the cases with TSP = 1.0 and TSP = 2.0 at M = 1.0. Deposition area coverage was 77.0% at TSP = 1.0 and 82.8% at TSP = 2.0. Particles were more likely to stick to the surface which led to more deposition in coolant wakes at TSP = 2.0 than at TSP = 1.0. Deposition in coolant wakes is suppressed at TSP = 1.0 because the coolant solidifies entrained particles reducing their likelihood of deposition upon impaction. At TSP = 2.0, particles are sticky enough to deposit even after passing through the coolant jet. Deposition in coolant wakes at TSP = 2.0 reduced the cooling effectiveness between cooling holes for every row of holes.

Deposition was more dense and difficult to remove from the surface at TSP = 2.0 than at TSP = 1.0. Interestingly, cooling hole blockage seemed more severe at TSP = 1.0 than at TSP = 2.0. Even though the s/d = 4.4 cooling holes appear to be more blocked at TSP = 1.0 than TSP = 2.0 they still provide better cooling at the low TSP case. The less dense deposition at TSP = 1.0 was more porous than deposition resulting from TSP = 2.0 allowing for coolant to pass through significant hole blockage at s/d = 4.4.

The spanwise-averaged effectiveness plot shown in Figure 7.13 illustrates that effectiveness between cooling rows decreased dramatically with an increase in TSP. The area-averaged effectiveness values at the baseline case, at TSP = 1.0, and at TSP = 2.0 were 0.280, 0.24, and 0.18 respectively. These values resulted in effectiveness reduction values of 13.0% and 36.7% at TSP = 1.0 and TSP = 2.0 respectively. It can be concluded that an increase in TSP leads to an increase in deposition rate and an increase in effectiveness reduction. This conclusion means that the improvement obtained in gas turbine performance
by increasing combustion temperatures may come with a side-effect of increased particle deposition rates that can lead to reduced turbine component life.

7.5 Summary of Leading Edge Deposition Findings

The effects of wax injection amounts, coolant blowing ratio, and particle thermal scaling parameter on deposition and cooling effectiveness were quantified for a staggered showerhead cooling geometry. Deposition was simulated at M = 1.0 using varying amounts of wax to observe the development of deposition and the effects of deposition development on cooling effectiveness. Deposition thickness measurements showed a linear increase in deposition thickness with the mass of injected wax; however, between 200g and 300g of injected wax, effectiveness reduction changed by less than 2% indicating that effectiveness had reached an equilibrium state.

Cooling effectiveness was quantified before and after deposition at blowing ratios of 0.5, 1.0, and 1.8. Prior to deposition, cooling effectiveness had a peak area-averaged value at M = 1.0. At low blowing ratios, cooling near stagnation was insufficient while at high blowing ratios coolant jet separation led to decreased effectiveness. Deposition simulations revealed that cooling hole blockage decreased with an increase in blowing ratio because increased coolant jet velocity prevented particles from depositing within cooling holes. The thickest deposition occurred downstream of the s/d = 1.2 row at M = 1.0 between cooling holes where particles entrained in coolant jets were likely to collect and deposit.

Cooling effectiveness was reduced by deposition at M = 1.0 and M = 1.8 while deposition led to a slight improvement in cooling effectiveness at M = 0.5. Although area-averaged effectiveness after deposition was highest at M = 0.5 than any other blowing ratio, the coolant coverage near stagnation was most effective at M = 1.0 which leads to the conclusion that M = 1.0 is the best cooling condition with or without deposition on the surface.

The effects of particle phase on deposition and resulting cooling effectiveness were investigated by simulating deposition at two different mainstream temperatures to achieve thermal scaling parameters of 1.0 and 2.0 at M = 1.0. An increase in thermal scaling parameter led to an increase in deposition rate and a dramatic decrease in cooling effectiveness. Although increased combustion temperatures improve the performance of
modern gas turbines, the resulting deposition enhanced by the elevated combustion temperatures may lead to critically reduced turbine component life.
Table 7.1. Local Blowing Ratios

<table>
<thead>
<tr>
<th>Cooling Row, s/d</th>
<th>M = 0.5</th>
<th>M = 1.0</th>
<th>M = 1.8</th>
</tr>
</thead>
<tbody>
<tr>
<td>-6</td>
<td>1.2</td>
<td>1.4</td>
<td>2.0</td>
</tr>
<tr>
<td>-3.5</td>
<td>0.6</td>
<td>1.0</td>
<td>1.8</td>
</tr>
<tr>
<td>1.2</td>
<td>-0.7</td>
<td>0.4</td>
<td>1.5</td>
</tr>
<tr>
<td>4.4</td>
<td>0.9</td>
<td>1.2</td>
<td>1.9</td>
</tr>
</tbody>
</table>

Figure 7.1. Adiabatic effectiveness contours at (a) M = 0.5, (b) M = 1.0, and (c) M = 1.8 with no deposition.

Figure 7.2. Spanwise-averaged effectiveness for the baseline film-cooling tests with no deposition.
Figure 7.3. Adiabatic effectiveness contours and deposition photographs of the showerhead at $M = 1.0$ (a) before deposition, (b) after 100g, (c) after 200g, and (d) after 300g of wax injection at $TSP = 1.0$.

Figure 7.4. Spanwise-averaged effectiveness before deposition and after 100g, 200g, and 300g of wax injection at $M = 1.0$ and $TSP = 1.0$. 
Figure 7.5. Average circumferential deposition depth after 100g, 200g, and 300g of wax injection at \( M = 1.0 \) and TSP = 1.0.

Figure 7.6. Area-averaged effectiveness plotted with respect to mass of injected wax at \( M = 1.0 \) and TSP = 1.0.
Figure 7.7. Adiabatic effectiveness contours (a) before deposition and (b) after 300g of wax injection at TSP = 1.0 and (c) deposition photographs at M = 0.5, 1.0, and 1.8.
Figure 7.8. Mid-span deposition photographs at (a) $M = 0.5$, (b) $M = 1.0$, and (c) $M = 1.8$ with TSP = 1.0.

![Mid-span deposition photographs](image)

Figure 7.9. Local effectiveness reduction for all blowing ratios tested at TSP = 1.0.

![Local effectiveness reduction](image)
Figure 7.10. Area-averaged effectiveness and effectiveness reduction with respect to blowing ratio for all cases tested at TSP = 1.0.

Figure 7.11. Non-dimensional deposition thickness at all three blowing ratios after 300g at TSP = 1.0.
Figure 7.12. Adiabatic effectiveness contours and deposition photographs for $M = 1.0$ (a) with no deposition, (b) after 300g at TSP = 1.0 and (c) after 300g at TSP = 2.0.

Figure 7.13. Spanwise-averaged effectiveness plots at $M = 1.0$ illustrating effects of TSP.
CHAPTER 8:
Endwall Deposition Studies

This work in Chapter 8 was conducted to determine the effects of deposition on endwall film-cooling with and without geometric modifications. Two separate industry provided geometries were used for the endwall experiments making up the bulk of the research conducted for this dissertation. A PW6000 first stage turbine vane geometry was used for the endwall film-cooling experiments conducted with and without transverse trenches. A Pack-B low pressure turbine blade geometry was used to determine the effects of deposition on film-cooling with and without endwall contouring. The effects of deposition on a film-cooled endwall with no geometric modifications are presented in Section 8.1 and were included in a paper by Lawson and Thole [90] which was accepted for publication in the *ASME Journal of Turbomachinery*. The effects of deposition on endwall film-cooling with transverse trenches are presented in Section 8.2 and were included in a paper by Lawson and Thole [91] which is currently being reviewed for publication in the *ASME Journal of Turbomachinery*. The effects of deposition on endwall film-cooling with and without endwall contouring are presented in Section 8.3 and will eventually be used to compose a paper to be submitted to the *ASME Journal of Turbomachinery*. The test matrix for the endwall deposition study conducted in the vane cascade is shown in Appendix B, Table B.3 and the test matrix for the endwall deposition study conducted in the blade cascade is shown in Table B.4.

8.1 Deposition on a Film-Cooled Endwall with No Geometric Modifications

For the current study, a test matrix was designed to explore the effects of film-cooling momentum flux ratio, TSP\(_{\text{max}}\), and wax injection duration on endwall deposition and adiabatic effectiveness. By increasing the wax injection duration, the total mass of wax injected in a given test was increased.

Prior to simulating deposition on a film-cooled surface, a flat endwall without film-cooling holes was installed to observe the deposition that occurred because of secondary flow structures alone. Figure 8.1 shows the 8 bit composite photographs for the deposition tests conducted using a flat endwall with no film-cooling. As shown in Figure 8.1a, deposition that occurred with TSP\(_{\text{max}}\) = 0.3 was isolated mostly to the leading edge region. When
exposed to a mainstream temperature of 337 K, the TSP\textsubscript{max} = 1.2 particles are soft and more likely to stick resulting in deposition that was more widespread but less dense at the leading edge than the TSP\textsubscript{max} = 0.3 deposition.

Deposition patterns shown by both cases in Figure 8.1 illustrate well where deposition collects because of secondary flow structures near the endwall. The densest deposition collected near the stagnation point downstream of the leading edge vortex. The leading edge vortex is formed by the total pressure gradient that becomes the static pressure gradient as the boundary layer flow stagnates at the vane leading edge. The pressure gradient causes the flow to move toward the endwall at which point particles with high inertia do not follow the vortex streamlines resulting in deposition on the endwall between the vortex and the stagnation point. Also illustrated well by the deposition is the saddle point where flow separates from the endwall upstream of the leading edge vortex.

Dense deposition can also be observed inside the trailing edge of the slot upstream of the stagnation region. Recall that the upstream slot flow was set to 0.75% of the mainstream flow for every experiment conducted in the vane cascade test section. Local pressures on the endwall at the exit of the slot upstream of stagnation are high resulting in either ingestion or very low coolant velocities exiting the slot. The low coolant velocity is not enough to prevent particles from impacting the surface.

Following deposition tests on the uncooled endwall, the film-cooled endwall illustrated in Figure 4.8a was installed. Prior to simulating deposition, a series of baseline (no deposition) adiabatic effectiveness tests were conducted at momentum flux ratios of I = 0.23, I = 0.95, and I = 3.6. Contours illustrating spatially resolved adiabatic effectiveness values on the film-cooled endwall are shown in Figure 8.2. Recall that the coolant flow condition is characterized by the momentum flux ratio of the leading edge coolant hole immediately upstream of the stagnation point. The contours show that coolant from the leading edge row is pulled toward the suction side of the vane causing increased effectiveness on the suction side of stagnation. Similar to the findings of Sundaram and Thole [14], effectiveness increases with an increase in blowing ratio. Separation reduces the effectiveness immediately downstream of the leading edge row at I = 3.6; however, the leading edge vortex pulls the coolant toward the endwall near the stagnation region resulting in increased effectiveness.
Adiabatic effectiveness was laterally averaged in the streamwise direction at different locations across the pitch of the leading edge cooling row. Figure 8.3 shows the baseline laterally averaged effectiveness distribution for the leading edge cooling row at the three momentum flux ratios tested. Laterally averaged effectiveness increased with an increase in momentum flux ratio with a large jump in effectiveness between $I = 0.95$ and $I = 3.6$.

**Deposition Development on Endwall Film-Cooling**

Prior to testing the effects of mainstream temperature and momentum flux ratio, the effect of wax injection duration was explored to determine the effects of deposition evolution on cooling effectiveness. Three separate tests were conducted with three wax injection durations at $I = 0.23$. Figure 8.4 shows the effectiveness contours and surface deposition plots for the case with no deposition along with the cases after the injection of 300g, 600g, and 900g of wax. Similar to the cases with no film-cooling, deposition was densest at the stagnation region of the vane and inside the trailing edge of the upstream slot. As expected, surface deposition increased with an increase in wax deposition with a larger difference between 300g and 600g than between 600g and 900g. Comparing directly between effectiveness contours and deposition plots, it is evident that the regions with the highest effectiveness (i.e. coldest surface temperatures) had the least deposition. Ai et al. [52] found that deposition capture efficiency decreased with an increase in blowing ratio and the flat plate results in Chapter 6 showed a decrease in coolant wake deposition with an increase in momentum flux ratio. Bons et al. [8] observed trough patterns caused by the lack of deposition in cooling hole wakes on actual turbine hardware. The lack of deposition in the wakes of coolant holes on the suction side of the vane leading edge in this case is most likely because of high coolant jet velocities preventing surface impaction of particles.

Not only did the coolant holes have an effect on deposition, but deposition had an effect on the cooling effectiveness of the leading edge row. The laterally averaged effectiveness plots in Figure 8.5 show that an increase in deposition causes a decrease in effectiveness everywhere along the leading edge cooling row. Deposition has a greater effect on the cooling effectiveness on the suction side of stagnation than on the pressure side. It is interesting to note that laterally averaged effectiveness values for the 600g and 900g cases are practically the same indicating that effectiveness approaches an equilibrium state between
600g and 900g of injection. The flat plate deposition study in Chapter 6 showed that effectiveness reached an equilibrium state. As deposition approaches an equilibrium state, the rate at which particles deposit approaches the rate at which particles are eroded from the surface.

Area-averaged effectiveness values were calculated for the region illustrated by the white box in Figure 8.4a. Figure 8.6 shows the leading edge area-averaged effectiveness and effectiveness reduction relative to the baseline plotted with respect to deposition area coverage for the three deposition development tests. Figure 8.6 shows that after 300g of wax injection the leading edge cooling effectiveness is reduced by 10%. After 600g, effectiveness is reduced by 25% and after 900g, effectiveness reduction reaches 26%. The difference in effectiveness reduction between 600g and 900g of wax injection is only 1%, which is well within the experimental uncertainty indicating that the effect of deposition on cooling effectiveness is approaching an equilibrium state as it did in the flat plate study discussed in Chapter 6.

The spray duration tests presented in Figures 8.4 through 8.6 revealed that 900g of wax injection was adequate to capture the effects of deposition on cooling effectiveness. For each experiment conducted in the vane cascade, 900g of wax was injected to explore the effects of momentum flux ratio and mainstream temperature on cooling effectiveness.

**Effects of Momentum Flux Ratio**

As the momentum of the jet exceeds the momentum of the mainstream fluid, the jet becomes more likely to separate from the surface. Figures 8.7, 8.8, and 8.9 show adiabatic effectiveness contour plots and surface deposition plots for $I = 0.23$, $I = 0.95$, and $I = 3.6$ respectively. The leading edge coolant jets are likely separated at $I = 3.6$; therefore, there are vast differences between the deposition pattern observed at $I = 3.6$ compared with the two lower momentum flux ratios. At $I = 3.6$ deposition is widespread downstream of the leading edge cooling holes on the pressure side of stagnation. It is clear that deposition collects everywhere outside of the coolant jet wakes as seen in Figures 8.9a and 8.9b. At $I = 3.6$ deposition is dense between cooling holes particularly near the leading edge of the cooling holes. Mounds of deposition developed between the leading edge coolant holes at $I = 3.6$ implying there was a blockage effect that created a recirculation region.
Sundaram and Thole [5] made laser Doppler velocimetry measurements at the stagnation point for the same cooling geometry tested in the current study. The location of the flowfield measurement plane illustrated in Figure 8.10 is shown by the white line in Figure 8.9a. The flowfield in Figure 8.10 shows the recirculation regions upstream and downstream of the coolant holes at $M = 2.5$. The recirculation regions correspond with deposition collection in the surface deposition plots shown in Figure 8.9. Deposition also collected downstream of the passage cooling holes most likely because jet separation created a recirculation region immediately downstream of these cooling holes.

It is apparent from Figures 8.7 through 8.9 that deposition has a negative impact on cooling at all three momentum flux ratios tested. Figure 8.11 shows the area-averaged effectiveness of the leading edge cooling row for the baseline case along with the $TSP_{\text{max}} = 0.3$ and $TSP_{\text{max}} = 1.2$ cases. For validation purposes, Figure 8.11 shows the area-averaged effectiveness of the same leading edge cooling row geometry tested by Sundaram and Thole [14]. The results in Figure 8.11 show that leading edge effectiveness increased with an increase in momentum flux ratio with and without surface deposition. Deposition caused a noticeable reduction in area-averaged effectiveness of the leading edge cooling row at all three momentum flux ratios.

**Effects of Thermal Scaling Parameter**

For the cases illustrated in Figures 8.7 through 8.9, the results are similar to those with no cooling in that deposition at $TSP_{\text{max}} = 1.2$ appears less dense at the stagnation region but more widespread than deposition at $TSP_{\text{max}} = 0.3$. Recall that TSP was varied by changing the mainstream air temperature. Effectiveness contours show that deposition at $TSP_{\text{max}} = 1.2$ has a more negative impact on cooling than deposition at $TSP_{\text{max}} = 0.3$. Deposition at $TSP_{\text{max}} = 1.2$ collected densely around cooling holes near stagnation and appeared to partially block some cooling holes. This blocking effect is the explanation for the increased reduction in effectiveness observed at $TSP_{\text{max}} = 1.2$. Particles with $TSP_{\text{max}} = 1.2$ have higher temperatures than particles with $TSP_{\text{max}} = 0.3$. Particles with higher temperatures are softer and stickier allowing them to deposit more readily upon impaction in the vicinity of cooling holes. Deposition at $TSP_{\text{max}} = 1.2$ is less dense near stagnation than deposition at $TSP_{\text{max}} = 0.3$ because the sticky particles entrained in the leading edge vortex at
TSP\(_{\text{max}}\) = 1.2 are likely to deposit on the vane surface before impacting the endwall. The photographs in Figure 8.12 at I = 0.23 for both TSP\(_{\text{max}}\) conditions illustrate the thick deposition on the vane surface at TSP\(_{\text{max}}\) = 1.2.

Area-averaged effectiveness was calculated for the passage row in addition to the leading edge row. The region used for taking the passage row area-average was 11d by 28d and is indicated by the black box in Figure 8.4a. Figure 8.13 shows the area-averaged effectiveness reduction relative to the baseline case for the passage and leading edge rows at TSP\(_{\text{max}}\) = 0.3 and TSP\(_{\text{max}}\) = 1.2 plotted with respect to momentum flux ratio. The results show that leading edge effectiveness reduction decreases with an increase in momentum flux ratio while passage row effectiveness reduction increases with an increase in momentum flux ratio.

In addition to the different effect of momentum flux ratio between leading edge and passage cooling rows, the effect of TSP\(_{\text{max}}\) is much greater on the passage row than on the leading edge row. Passage row effectiveness reduction reaches 32% at TSP\(_{\text{max}}\) = 1.2 and only as high as 25% at TSP\(_{\text{max}}\) = 0.3. The difference in effects observed between the passage row and the leading edge row is most likely because the leading edge coolant is less susceptible to separation because of the presence of the leading edge vortex. The leading edge vortex holds the coolant on the endwall which allows for increased effectiveness and reduces downstream deposition buildup. The passage row coolant, on the other hand, is more susceptible to separation which leads to deposition accumulation close to the trailing edges of coolant holes decreasing effectiveness. The extent of deposition downstream of the coolant holes on the passage side is dependent on the TSP\(_{\text{max}}\) value. The soft sticky particles that exist at TSP\(_{\text{max}}\) = 1.2 lead to dense deposition downstream of passage row cooling holes ultimately leading to increased effectiveness reduction.

**Summary of Deposition Effects on a Film-Cooled Endwall with No Modifications**

A wax injection method similar to that used in the leading edge study was used to dynamically simulate solid and molten particle deposition in a large scale turbine vane cascade. The effects of deposition development, momentum flux ratio, and particle thermal scaling parameter on endwall film-cooling effectiveness were quantified.
At the lowest momentum flux ratio, three tests were conducted with different wax injection durations to observe deposition development and measure the effects of deposition evolution on cooling. Little change in effectiveness and surface area coverage between 600g and 900g of wax injection indicated that deposition approached an equilibrium state.

Areas of dense deposition were observed at the stagnation region of the vane, downstream of the passage cooling rows, and upstream of leading edge cooling rows at high momentum flux ratios. Based on these observations compared with laser Doppler velocimetry measurements, dense deposition most likely collects in areas where vortex behavior causes particle impaction on the endwall.

Area-averaged effectiveness values were calculated for the leading edge and passage cooling rows on the endwall. Results showed that leading edge cooling row effectiveness reduction was comparable for both low and high thermal scaling parameter values and decreased with an increase in momentum flux ratio. On the other hand, the passage cooling row experienced an increase in effectiveness reduction with an increase in momentum flux ratio. The difference in sensitivity to deposition between leading edge and passage cooling rows was because jet separation at the leading edge cooling row was suppressed at high momentum flux ratios by the leading edge vortex. The buildup of deposition downstream of passage cooling holes led to a greater decrease in effectiveness with the high thermal scaling parameter than with the low thermal scaling parameter.

This study showed that endwall cooling was highly sensitive to deposition. The effects of deposition on cooling were sensitive to film-cooling operating condition, particle thermal scaling parameter, and film-cooling geometry. Excess deposition could lead to reduced cooling effectiveness and possibly turbine component failure. It is essential to understand the driving mechanisms behind deposition so that advanced film-cooling configurations can be designed to mitigate deposition.

### 8.2 Deposition on Endwall Film-Cooling with Transverse Trenches

This section presents work that was discussed in a paper [91] that is currently under review for publication in the *ASME Journal of Turbomachinery*. The endwall film-cooling geometry discussed in Section 8.1 was modified by embedding the cooling rows in transverse trenches. This section discusses the effects of trench depth, momentum flux ratio, and
thermal scaling parameter on deposition and the resulting cooling effectiveness in the leading edge region.

**Effects of Trench Depth with No Deposition**

By embedding the film-cooling holes into a transverse trench, the tendency of a jet to separate from the surface is reduced. The trench effectively channels the coolant along the length of the trench improving cooling within the trench as well as downstream of the trench. Sundaram and Thole [14] concluded that the effectiveness of the trench is sensitive to its depth; therefore, deposition simulation and adiabatic effectiveness experiments were conducted with trench depths of 0.4d, 0.8d, and 1.2d. Figure 8.14 shows adiabatic effectiveness contours before deposition at momentum flux ratios of 0.23 and 3.6 for all trench depths tested in the current study along with the case with no trench which was discussed in Section 8.1. The contours show that the trenches improve cooling effectiveness between holes for both the leading edge and passage cooling rows. As trench depth increases, cooling effectiveness within the trench improves; however, cooling effectiveness downstream of the trench diminishes. Because of the opposite trends, the medium trench depth of 0.8d provides improved cooling effectiveness within the trench while maintaining adequate cooling downstream of the trench.

The area-averaged effectiveness for the leading edge and passage cooling rows for all trench depths with no deposition are given in Figure 8.15. The white and black outlines shown in Figure 8.14a represent the areas used for calculating the area-averaged effectiveness for the leading edge and passage cooling rows, respectively. Trench depth had little effect on the leading edge row effectiveness; however, the 0.8d trench yielded the highest area-averaged effectiveness for the passage cooling row. The distinction between trench depths is most obvious at high momentum flux ratios because the trench improves effectiveness by mitigating separation effects that occur at high momentum flux ratios. The trenches allow for increased coolant mass to be supplied without the risk of separation effects. The result is improved cooling between cooling holes and downstream of cooling rows with the trench.
Effects of Trench Depth with Deposition

Deposition simulations were conducted to determine the effects of deposition on cooling effectiveness for all three trench depths. These simulations were carried out at TSP = 1.2 for all cooling conditions at each trench depth. Figure 8.16 shows photographs of the surface and corresponding adiabatic effectiveness contours after deposition simulation at I = 3.6. For each case, deposition was most dense in the stagnation region. From the results presented in Section 8.1 it was concluded that deposition in the stagnation region was caused by the leading edge vortex. The static pressure gradient through the boundary layer at the stagnation region pulls the flow toward the wall forming a recirculation region. Particles with high inertia become entrained in the leading edge vortex and deposit on the endwall. Figure 8.16 shows that deposition patterns are highly dependent on trench depth. For the case with no trench, a dense number of deposits collected upstream and between the leading edge cooling holes and was prevented downstream of that cooling row in areas with good coolant coverage. Figures 8.16b through 8.16d show that deposition between the leading edge cooling holes decreased with an increase in trench depth. Downstream of the leading edge cooling row, deposition patterns change noticeably with trench depth. For the 0.4d trench, the downstream deposition pattern looks very similar to the no trench case with deposition surrounding the areas where coolant jets divert particle trajectories. For the 1.2d trench, deposition in the stagnation region extends upstream almost to the trailing edge lip of the trench with little to no effect of the coolant jets on deposition. The 0.8d trench, on the other hand, prevents deposition downstream of the trench. The difference in deposition patterns between the 0.8d and 1.2d trenches exists because coolant coverage downstream of the 0.8d trench is uniform minimizing deposition in the stagnation region.

The area-averaged effectiveness resulting from the deposition as well as the percentage reduction of effectiveness for the leading edge cooling row at all trench depths are shown in Figure 8.17. Note that the reduction values are relative to that same trench geometry with no deposition. All three trenches clearly outperformed the holes with no trench. Deposition of particles reduced the leading edge effectiveness by as much as 30% with no trench while the highest reduction for any of the trenched cases was 13%. Area-averaged effectiveness after deposition was highest through the entire range of momentum
flux ratios for the case with the 0.8d trench. The cases with the 0.8d trench also yielded the lowest effectiveness reduction out of the three trench depths tested.

In the case with no trench for the passage row of film-cooling holes, there was dense deposition downstream of the row. As the trench depth increased, the deposition that collected in and around the trench decreased. Effectiveness contours in Figure 8.16 show that the 0.8d and 1.2d trenches allow for coolant to fill in between cooling holes creating a layer of cool air in the trench. The cool air layer prevents deposition from forming on the trailing edge lip of the trench for the 0.8d and 1.2d trench cases. For the case with no trench and the 0.4d trench, effectiveness contours show minimum cooling between and downstream of holes. The result is deposition collection on the trailing edge lip of the 0.4d trench between cooling holes.

Figure 8.18 shows the area-averaged effectiveness after deposition and percentage reduction in effectiveness caused by deposition for the passage cooling row at all trench depths. Results indicate the passage row effectiveness is more sensitive to trench depth than the leading edge row, which is because of the strong influence of the horseshoe vortex at the leading edge. Similar to the leading edge row, the 0.8d trench for the passage row provides better cooling than the other trench depths. The percentage reduction in effectiveness for the case with no trench was as high as 32% while the percentage reduction with the trench was at most 17%. It is interesting to note that the reduction in effectiveness was near zero for the high momentum flux ratio cases at both the leading edge and passage cooling rows. The lack of any effect of deposition for the high momentum flux case is discussed further in the following section.

To quantify the combined cooling effectiveness of the leading edge and passage cooling rows, a total area-average of the two cooling rows was calculated for each case. The total area-average was calculated by taking a weighted area-average of the two representative areas, shown in Figure 8.14a, using their respective area-averaged values. Figure 8.19 shows the total area-averaged values for the cases with no trench and the 0.8d trench before and after deposition for all three momentum flux ratios tested. The results show that the 0.8d trench is much less sensitive to deposition than the case with no trench. Note that even with deposition, the 0.8d trench performs better than the baseline with no trench and no deposition.
Figure 8.20 shows photographs of particle trajectories in the stagnation plane for $I = 0.23$ and $I = 3.6$ at TSP = 1.2 for no trench and the 0.8d trench. Recall that the image plane is in line with the flow through the centerline of a film-cooling hole at stagnation as illustrated in Figure 4.8b. Lines have been added below each photo in Figure 8.20 to illustrate the film-cooling hole location in each image. The images in Figure 8.20 are composite photographs taken over a time period of approximately 0.01s (20 frames). A range of representative frames were specifically chosen to compose each image to illustrate the dominant flow structures for each cooling condition. It is important to note that particles are generally moving from left to right and tend to be pulled toward the endwall by the leading edge vortex. The high speed photographs complement the assertion that deposition builds in the endwall corner.

For all cases in Figure 8.20, particles entrained in the leading edge vortex are pulled toward the endwall downstream of the leading edge cooling row. Figure 8.20 indicates the leading edge vortex is larger at $I = 0.23$ than at $I = 3.6$. At $I = 3.6$ the corner vortex at stagnation is paired with a secondary vortex that forms upstream of the cooling row. This secondary vortex forms because the coolant jets at $I = 3.6$ create a blockage effect that causes the flow to wrap toward the endwall upstream of the jets. For the case with no trench, this upstream vortex results in dense deposition between and upstream of cooling holes. The upstream vortex forms farther downstream with the trench than without the trench pulling particles into the trench. Particles that are pulled into the trench either deposit within the trench or are solidified and ejected by the coolant. At $I = 0.23$ the leading edge vortex is larger with the trench than without the trench. Particles entrained in the larger vortex above the trench have less inertia and are more likely to follow fluid streamlines preventing deposition. In contrast, particles entrained in the leading edge vortex with no trench have high inertia and are more likely to exit the vortex and deposit on the endwall.

**Effects of Momentum Flux Ratio**

Thole et al. [19] showed that momentum flux ratio is the parameter that determines the separation tendency of a film-cooling jet. At low momentum flux ratios, coolant jets remain attached to the surface and provide good coolant coverage while at high momentum flux ratios, coolant jets separate leaving the surface exposed to hot mainstream gases. For
the case with no trench in Figure 8.14a, coolant from the leading edge row is pulled toward the suction side of the vane at \( I = 0.23 \) providing little coolant coverage along the pressure side. At \( I = 3.6 \) coolant separates but is then brought back to wash the surface as a result of the spanwise pressure gradient along the vane stagnation thereby providing good coolant coverage near stagnation.

The cooling trend is slightly different for the case with the 0.8d trench as shown in Figure 8.14c. At \( I = 0.23 \) coolant is channeled along the trench toward the suction side of the vane and exits the trench along the suction side of the airfoil. At \( I = 3.6 \) coolant also fills the trench but exits along the entire trench width. The additional spreading of the coolant in the trench before exiting to interact with the horseshoe vortex provides better cooling.

Figure 8.21 shows surface photographs and adiabatic effectiveness contours at all three momentum flux ratios for the 0.8d trench with deposition. All three experiments in Figure 8.21 were conducted with \( \text{TSP}_{\text{max}} = 1.2 \). It is clear when comparing deposition photographs to the effectiveness contours that the coolant locations correlate with deposition patterns. Deposition is prevented in regions of high cooling for two reasons. First, the jet momentum carries the particles away from the surface. Second, when entrained in coolant gases, particles fully solidify making deposition less likely in the event of surface impaction.

Another interesting trend illustrated in Figure 8.21 is that deposition between cooling holes in the leading edge trench increases with an increase in momentum flux ratio. This trend can be explained by observing the particle behavior at low and high momentum flux ratios for the 0.8d trench in Figure 8.20b. At \( I = 0.23 \), the leading edge vortex near stagnation pulls particles toward the endwall to deposit downstream of the cooling row. At \( I = 3.6 \) two vortices are present; one upstream and the other downstream of the film-cooling holes. The upstream vortex pulls particles toward the endwall creating a mechanism for high inertia particles to be carried into the trench to deposit. Even though particles are pulled into the trench at high momentum flux ratios, the amount of coolant in the trench reduces deposition as compared to the case with no trench. As illustrated in Figure 8.16a, little coolant coverage between cooling holes results in large amounts of deposition with no trench.

In general, the 0.8d trench mitigates the negative effects of deposition on cooling effectiveness particularly at high momentum flux ratios. Coolant coverage improves with an
increase in blowing ratio which leads to a decrease in deposition coverage. As the extent of deposition decreases, the affect that it has on cooling effectiveness also decreases. In addition, the trench reduces deposition between cooling holes because it creates a pocket of coolant along the length of the cooling row that acts as a protective barrier against deposition.

**Effects of Thermal Scaling Parameter**

For results presented in the previous sections, deposition was simulated with $TSP_{\text{max}} = 1.2$ for which most particles were solid and only the largest particles were molten. Additional experiments were conducted with $TSP_{\text{max}} = 2.2$ to determine the effects of particle phase on cooling effectiveness for the 0.8d trench. At $TSP_{\text{max}} = 2.2$, particles smaller than 70µm were in solid form and particles larger than 70 µm were in molten form upon reaching the test section.

Figure 8.22 shows deposition photographs and cooling effectiveness contours at $I = 0.23$ and 3.6 for $TSP_{\text{max}} = 1.2$ and 2.2. The differences between the two $TSP_{\text{max}}$ values are small. At $I = 0.23$ deposition collected farther downstream on both the suction and pressure sides of the vane at $TSP_{\text{max}} = 2.2$ than at $TSP_{\text{max}} = 1.2$. Deposition collected more densely downstream of the passage row at $TSP_{\text{max}} = 2.2$ than at $TSP_{\text{max}} = 1.2$. These differences exist because particles at $TSP_{\text{max}} = 2.2$ are softer and stickier than at $TSP_{\text{max}} = 1.2$.

The photographs in Figure 8.22b show that deposition downstream of the leading edge row was again very similar between the two $TSP_{\text{max}}$ values. Dense deposition extended farther downstream on the pressure side at $TSP_{\text{max}} = 2.2$ than it did at $TSP_{\text{max}} = 1.2$; however suction side deposition looks identical between the two $TSP_{\text{max}}$ cases. Downstream of the leading edge row near stagnation, the deposition pattern between the two $TSP_{\text{max}}$ cases appears slightly different. At $TSP_{\text{max}} = 2.2$, deposition filled in between cooling jets making individual jets more visible than at $TSP_{\text{max}} = 1.2$. Downstream of the leading edge row, coolant spreading was better at $TSP_{\text{max}} = 2.2$ than at $TSP_{\text{max}} = 1.2$ because of reduced deposition in the stagnation region at $TSP_{\text{max}} = 2.2$. The results in Section 8.1 showed that deposition near stagnation decreased with increasing $TSP_{\text{max}}$ because particles entrained in the leading edge vortex were more likely to deposit on the vane surface than the endwall when they were soft and sticky.
Effectiveness reduction compared with the baseline cases without deposition for the leading edge and passage cooling rows at both TSP\textsubscript{max} values is shown in Figure 8.23. At low momentum flux ratios, effectiveness reduction is slightly higher at TSP\textsubscript{max} = 2.2 than at TSP\textsubscript{max} = 1.2 while at I = 3.6 differences in effectiveness reduction between the two TSP\textsubscript{max} cases is almost negligible. The small differences in effectiveness reduction between the two TSP\textsubscript{max} cases, indicates that the 0.8d trench mitigates the negative effects of deposition regardless of particle phase especially at high momentum flux ratios.

**Summary of Deposition Effects on Film-Cooling with and without Trenches**

The large scale turbine vane cascade model was used to quantify the effects of deposition on endwall film-cooling with and without transverse trenches. The effects of trench depth, momentum flux ratio, and thermal scaling parameter on adiabatic effectiveness were quantified for two rows of cooling holes near the vane endwall junction. The leading edge row with holes in line with the mainstream flow was located just upstream of the vane stagnation point and a passage row with holes oriented with a compound angle of 90° relative to the flow was located upstream and between neighboring vanes.

Although all three trench geometries tested clearly outperformed the geometry with no trench, results showed that the medium trench depth of 0.8d outperformed all other geometries both before and after deposition for both the leading edge and passage cooling rows. As trench depth increased, cooling effectiveness within the trench increased while effectiveness downstream of the trench decreased.

The effectiveness reduction caused by deposition was similar for all three trench depths but clearly decreased with an increase in momentum flux ratio. For trench film-cooling geometries, deposition reduced effectiveness by less than 5% at I = 3.6.

Deposition within trenches increased with an increase in momentum flux ratio. Flow visualization results from a high speed camera system showed that an upstream vortex that formed at high momentum flux ratios pulled particles into the trench; however, deposition within the trench was not as severe as deposition between cooling holes with no trench. The trench creates a pocket of coolant along the entire row of holes that acts as a protective barrier against deposition.
Experiments conducted at two TSP\textsubscript{max} values for the best trench depth of 0.8d revealed that effectiveness reduction was insensitive to particle phase. The findings from the trench cooling experiments confirm that trench cooling geometries can be used to mitigate the negative effects of particle deposition particularly at high momentum flux ratios. The 0.8d trench depth should be considered for use in the first stage vane row in turbines with traces of impurities in the fuel and air.

8.3 Deposition on Endwall Film-Cooling with Endwall Contouring

This section describes the results from experiments that were conducted to determine the effects of deposition on film-cooling with and without endwall contouring. A Pack-B blade cascade previously tested by Lynch et al. [10] was used to conduct these experiments. First the effects of endwall contouring with no deposition are discussed followed by the effects of deposition development, effects of contouring with deposition, effects of blowing ratio, and effects of thermal scaling parameter. The results presented here will eventually be composed for an article for submission to the \textit{ASME Journal of Turbomachinery}.

\textbf{Effects of Endwall Contouring with No Deposition}

Lynch et al. [10] measured adiabatic effectiveness on a film-cooled flat endwall and a film-cooled contoured endwall with identical film-cooling locations as shown in Figures 4.10 and 4.11. For this study, adiabatic effectiveness experiments were conducted on the same geometries tested by Lynch et al. [10] to provide a baseline for experiments with simulated deposition. Figure 8.24 shows adiabatic effectiveness contours for the flat and contoured endwalls at M = 1.0 and M = 2.0 with no deposition. The contoured endwall was effective at reducing secondary flows [10]; however, as shown in Figure 8.24, cooling effectiveness through the passage was reduced by the contour because the region of high elevation in the passage prevented coolant from spreading across the passage.

Centerline and laterally averaged effectiveness line plots for the flat endwall at M = 1.0 are shown in Figure 8.25 and effectiveness line plots for the contoured endwall at M = 1.0 are shown in Figure 8.26. The data obtained in the current study and the data obtained by Lynch et al. [10] are in excellent agreement. It is important to note that the data in Figures 8.25 and 8.26 are representative of the data from hole 3 on the flat and contoured endwalls.
Flat endwall centerline data decreases from approximately 0.7 at the hole exit to approximately 0.15 at a distance of 20d downstream. Contoured endwall centerline data decreases from approximately 0.58 at the hole exit to approximately 0.1 at a distance of 20d downstream.

In addition to the hole 3 centerline and laterally averaged effectiveness, area-averaged effectiveness from the entire row of holes was calculated. The area represented by the white outline in Figure 8.24a was used to calculate area-averaged effectiveness for the flat and contoured endwalls. The data from hole 3 implies that the flat endwall outperforms the contoured endwall; however, Figure 8.27 shows that when the data is averaged through the passage the contour outperforms the flat endwall at M = 1.0. Cooling effectiveness was lower at M = 2.0 than at M = 1.0 for both geometries implying that coolant jets were separated from the surface. Lower effectiveness on the contoured endwall relative to the flat endwall at M = 2.0 implies that the flat endwall was less sensitive to separation effects than the contoured endwall. The difference in separation sensitivity was probably because of the differences in cooling hole inclination angle between the two geometries. Cooling holes on the contoured endwall had high inclination angles and were more likely to separate from the surface to become entrained in the hot mainstream as compared with cooling holes on the flat endwall.

**Effects of Deposition Development**

Results from flat plate cooling with deposition and endwall cooling with deposition in the vane cascade study showed that deposition developed and approached an equilibrium state. Prior to conducting experiments to determine the effects of contouring, momentum flux ratio, and thermal scaling parameter on cooling, experiments were first conducted to verify that deposition would reach an equilibrium state in a similar manner in the blade cascade facility as it did in the vane cascade facility.

Deposition simulations were individually conducted with wax injections of 300g, 600g, and 900g to determine the effects of deposition development on endwall film-cooling. These development experiments were conducted using the flat endwall at M = 1.0 and TSP$_{med}$ = 0.3. Figure 8.28 illustrates the effectiveness contours and corresponding deposition photographs before deposition, after 300g, after 600g, and after 900g of wax injection.
Figure 8.28 illustrates a significant negative effect of deposition on cooling and similar to the results obtained in the vane cascade facility, the differences in effectiveness and deposition appears small between 600g and 900g of wax injection.

Figure 8.29 shows area-averaged effectiveness and effectiveness reduction plotted with respect to wax injection mass. Deposition reduced effectiveness by 37% after 300g of injection and 51% after 600g of injection. The effectiveness reduction between 600g of injection and 900g of injection changed by less than 3% implying that the deposition had reached an equilibrium state as it had in previous experiments. Deposition reduced the area-averaged cooling effectiveness by approximately 50% on the flat endwall at M = 1.0 and TSP_{med} = 0.3.

**Effects of Endwall Contouring with Deposition**

Baseline experiments showed that although the contoured endwall prevented cooling from spreading across the passage, the area-averaged effectiveness at M = 1.0 was essentially equivalent between the flat and contoured endwalls. Experiments were conducted at M = 1.0 and M = 2.0 for both geometries using 900g of wax injection to characterize the effects of deposition.

Figures 8.30 and 8.31 show effectiveness contours and corresponding deposition photographs for the flat and contoured endwall at M = 1.0 and M = 2.0. Deposition photographs illustrate a significant difference in deposition patterns between the flat and contoured endwalls. On the flat endwall, deposition appears relatively uniform in the passage with a slight hint of a passage vortex deposition streak. Deposition appears to be most dense along the base of the pressure side of the blade between the blade and the film-cooling holes increasing toward the trailing edge. On the contoured endwall, deposition is most dense in regions of high elevation along the ridges of the contour. Deposition is virtually non-existent in the valleys of the contour. This deposition pattern exists because particles with high inertia cannot follow fluid streamlines over the ridges resulting in deposition in areas where changes in streamlines are most dramatic. This effect acts as an advantage in the case with the contoured endwall where cooling holes are located on a downhill slope between the contour ridge and the pressure side of the blade. Although there is some deposition around the cooling holes, it is not as dense around the contoured endwall cooling holes as it is around the flat endwall cooling holes. Recall the computationally
predicted accretion patterns illustrated in Figure 5.15. The experimental deposition patterns shown in Figure 8.30 are very similar to the predicted accretion patterns in Figure 5.15 at $M = 1.0$ for the flat and contoured endwalls. Particles are evenly distributed through the flat endwall passage and build up in regions of high elevation through the contoured endwall passage. The agreement between computationally predicted accretion rates and experimentally obtained deposition patterns shows promise that computational techniques could be used to aid in the design of contoured endwall cooling configurations to mitigate the negative effects of deposition.

Decreased deposition downstream of the cooling holes on the contoured endwall as compared to the flat endwall results in higher cooling effectiveness on the contoured endwall after deposition. Figure 8.32 shows the area-averaged effectiveness values before and after deposition on the flat and contoured endwalls at $M = 1.0$ and $M = 2.0$. The results in Figure 8.32 verify what could be observed in Figure 8.30. Deposition has less of an effect on contoured endwall cooling effectiveness than flat endwall cooling effectiveness. Deposition reduced effectiveness at $M = 1.0$ by 49% on the flat endwall and only 40% on the contoured endwall. Although effectiveness reduction is significant in both cases, endwall contouring shows promise in reducing the negative impact of deposition on cooling effectiveness.

**Effects of Blowing Ratio**

The results presented in the previous section showed that, in general, endwall contouring reduced the negative impact of deposition on cooling. This section discusses the effects of blowing ratio on deposition and the resulting cooling effectiveness.

The deposition photographs in Figures 8.30 and 8.31 show that the effects of blowing ratio on deposition patterns are limited to areas within close proximity to the cooling holes. Deposition downstream of cooling holes was more dense at $M = 1.0$ than at $M = 2.0$. The high jet velocities at $M = 2.0$ prevented deposition directly downstream of holes by cooling particles and diverting their trajectories. Even with local effects of blowing ratio on deposition, the surface roughness generally appears insensitive to blowing ratio.

On the other hand, the effect of deposition on cooling effectiveness is highly sensitive to blowing ratio. Unlike the obvious decrease in effectiveness caused by deposition at $M = 1.0$, the effect of deposition on effectiveness at $M = 2.0$ appears almost negligible. The area-
averaged effectiveness values in Figure 8.32 show that the effect of deposition on cooling effectiveness is small at $M = 2.0$. Recall that results from the flat plate cooling study showed a similar trend. At low coolant rates when jets are attached to the surface, effectiveness is highly sensitive to deposition while at high coolant rates when jets are separated, the effects of deposition on cooling are small. When jets are attached, surface roughness can increase mixing with the mainstream. When jets are separated, they are already mixing with the mainstream and the effects of surface roughness are low. Figure 8.32 shows that the contoured endwall outperforms the flat endwall at $M = 1.0$; however, the flat endwall outperforms the contoured endwall at $M = 2.0$. Deposition reduced effectiveness by less than 4% on the flat and contoured endwall at $M = 2.0$ as compared to the 49% and 40% reductions at $M = 1.0$. Before deposition, it is best to operate at $M = 1.0$; however, after deposition results indicate that $M = 2.0$ is the better operating condition for maximum cooling effectiveness.

**Effects of Thermal Scaling Parameter**

Experiments were conducted to determine the effects of thermal scaling parameter on deposition and cooling effectiveness on the flat and contoured endwall. In addition to the experiments at $TSP_{med} = 0.3$, experiments were conducted at $TSP_{med} = 1.1$ using $M = 1.0$.

Figure 8.33 shows effectiveness contours and corresponding deposition photographs for the flat and contoured endwalls with $TSP_{med} = 0.3$ and $TSP_{med} = 1.1$ at $M = 1.0$. The deposition photographs in Figure 8.33 show significant differences between the two $TSP_{med}$ cases. It is important to note that at $TSP_{med} = 1.1$, the median particle size of 35 $\mu$m had a TSP value of 1.1 meaning than particles larger than 35 $\mu$m had TSP values greater than 1.1. This means that most of the injected wax was in molten form upon reaching the blade cascade. The deposition resulting from the $TSP_{med} = 1.1$ case appeared to glaze the surface because of the high fraction of molten particulate. Only the deposits that were in solid form upon impacting the surface showed up as white areas in the photographs. There were dense areas of solid particle deposition near film-cooling holes on both endwalls because the jets cooled particles causing them to condense and effectively freeze to the surface. Although more difficult to remove from the surface, the deposition at $TSP_{med} = 1.1$ resulted in reduced surface roughness as compared to deposition at $TSP_{med} = 0.3$. 
Area-averaged effectiveness before and after deposition for both endwalls and both TSP_{med} values are shown in Figure 8.34. The effect of deposition on cooling was less severe at TSP_{med} = 1.1 than at TSP_{med} = 0.3 because of reduced surface roughness resulting from molten particle deposition. While effectiveness reductions were 49% and 40% for the flat and contoured endwalls at TSP_{med} = 0.3, they were slightly lower at 29% and 34% at TSP_{med} = 1.1. Unlike the results from previous studies, the data shown in Figure 8.34 suggests that operating at elevated temperatures may have some benefit in terms of reducing the negative impacts of deposition on cooling. The difference between the blade cascade experiments and the vane cascade experiments was that higher TSP values were tested in the blade cascade experiments. For comparison, at TSP_{med} = 1.1 in the blade cascade experiments, particles larger than 32 \( \mu \text{m} \) were in molten form. At TSP_{max} = 1.2 in the vane cascade experiments, particles larger than 90 \( \mu \text{m} \) were in molten form. This means that there was a much higher fraction of molten particles during testing in the blade cascade as compared to the vane cascade. This difference in results for different ranges of TSP values suggests that there exists a critical TSP. Below this critical TSP, effectiveness reduction increases with TSP while above this critical TSP, effectiveness reduction decreases with TSP. By avoiding operation near the critical TSP the effects of deposition on cooling can be mitigated.

**Summary of Deposition Effects on Film-Cooling with and without Endwall Contouring**

A blade cascade model was used to quantify the effects of deposition on film-cooling with and without endwall contouring. The effects of deposition development, blowing ratio and thermal scaling parameter on cooling effectiveness were quantified for flat and contoured endwalls with identical film-cooling hole locations.

Similar to the findings from the flat plate, leading edge, and previous endwall studies, deposition development experiments showed that deposition reached an equilibrium state. Upon reaching the equilibrium state, further injection of particulate had a negligible effect on deposition and the resulting cooling effectiveness.

Comparing surface photographs between the flat and contoured endwalls revealed significant differences in deposition patterns. Deposition on the flat endwall was relatively uniform throughout the passage while deposition on the contoured endwall was sensitive to endwall topography. Particles deposited densely in regions of increasing elevation and
sparsely in regions of decreasing elevation. Deposition in the vicinity of cooling holes on the contoured endwall was less severe than deposition near cooling holes on the flat endwall because the cooling holes were placed in a region of decreasing elevation on the contoured endwall. Adiabatic effectiveness results showed that the contour slightly decreased the negative impact of deposition on cooling effectiveness.

Deposition experiments at different blowing ratios showed that cooling was less sensitive to deposition at high blowing ratios than at low blowing ratios. This finding was similar to the findings from the flat plate study. Cooling is less sensitive to deposition at high blowing ratios than low blowing ratios because jet separation at high blowing ratios increases mixing with the mainstream regardless of the roughness effect caused by the deposition.

Experiments conducted at two thermal scaling parameter values revealed that cooling effectiveness was less sensitive to deposition at high TSP values than at low TSP values. At low TSP values, deposition resulted from solid particle agglomeration on the surface. On the other hand, at high TSP values, particles were molten upon impaction causing glassy deposition. Glassy deposition resulted in decreased surface roughness as compared to solid particle agglomeration. Recall that results from the vane cascade studies showed that effectiveness reduction increased with an increase in TSP which is the opposite of the trend observed in the blade cascade study. The difference is that the deposition experiments in the vane cascade studies were conducted at lower TSP values than those conducted in the blade cascade. This finding implies that there is a critical TSP value. Below the critical TSP, effectiveness reduction increases with an increase in TSP and above the critical TSP, effectiveness reduction decreases with an increase in TSP.

The deposition experiments from the blade cascade study showed that endwall contouring could be used to mitigate the negative effects of deposition; however, more work is required to determine the most effective locations for placing film-cooling holes.
Figure 8.1. Deposition photographs with no film-cooling for (a) TSP_{max} = 0.3 and (b) TSP_{max} = 1.2.

Figure 8.2. Adiabatic effectiveness contours at (a) I = 0.23, (b) I = 0.95, and (c) I = 3.6 with no deposition.
Figure 8.3. Laterally averaged effectiveness of the leading edge cooling holes with no deposition.

Figure 8.4. Adiabatic effectiveness contours and corresponding deposition photographs (a) before deposition (b) after 300g, (c) after 600g, and (d) after 900g of wax injection at \( l = 0.23 \).
Figure 8.5. Laterally averaged effectiveness of the leading edge cooling holes after 300g, 600g, and 900g of wax injection at $l = 0.23$.

Figure 8.6. Area-averaged effectiveness of the leading edge cooling row plotted with respect to deposition area coverage at $l = 0.23$. 
Figure 8.7. Effectiveness contours and corresponding deposition photographs at $I = 0.23$ for (a) no deposition, (b) $\text{TSP}_{\text{max}} = 0.3$, and (c) $\text{TSP}_{\text{max}} = 1.2$.

Figure 8.8. Effectiveness contours and corresponding deposition photographs at $I = 0.95$ for (a) no deposition, (b) $\text{TSP}_{\text{max}} = 0.3$, and (c) $\text{TSP}_{\text{max}} = 1.2$. 
Figure 8.9. Effectiveness contours and corresponding deposition photographs at $I = 3.6$ for (a) no deposition, (b) $TSP_{\text{max}} = 0.3$, and (c) $TSP_{\text{max}} = 1.2$.

Figure 8.10. Vane leading edge flowfield by Sundaram and Thole [5] for $M = 2.5$. 
Figure 8.11. Leading edge area-averaged effectiveness plotted with respect to blowing ratio for no deposition, $TSP_{\text{max}} = 0.3$, and $TSP_{\text{max}} = 1.2$.

Figure 8.12. Photographs of leading edge deposition at $I = 0.23$ with (a) $TSP_{\text{max}} = 0.3$ and (b) $TSP_{\text{max}} = 1.2$. 
Figure 8.13. Area-averaged effectiveness reduction for leading edge and passage cooling holes.

Figure 8.14. Contours of adiabatic effectiveness for different trench depths with no deposition.
Figure 8.15. Baseline area-averaged effectiveness for the leading edge and passage cooling rows with different trench depths and no deposition.

Figure 8.16. Deposition photographs and effectiveness contours at $I = 3.6$ for all trench depths after deposition.
Figure 8.17. Area-averaged effectiveness and effectiveness reduction for the leading edge row after deposition.

Figure 8.18. Area-averaged effectiveness and effectiveness reduction for the passage row after deposition.
Figure 8.19. Area-averaged effectiveness of passage and leading edge cooling rows for no trench and the 0.8d trench before and after deposition.

Figure 8.20. Flow visualization pictures at $I = 0.23$ and $I = 3.6$ for (a) no trench and (b) the 0.8d trench.
Figure 8.21. Deposition photographs and effectiveness contours for the 0.8d trench at all momentum flux ratios after deposition at TSP$_{\text{max}} = 1.2$.

Figure 8.22. Deposition photographs and adiabatic effectiveness contours after deposition for the 0.8d trench at (a) I = 0.23 and (b) I = 3.6.
Figure 8.23. Effectiveness reduction for the leading edge and passage rows for the 0.8d trench at $TSP_{\text{max}} = 1.2$ and $TSP_{\text{max}} = 2.2$.

Figure 8.24. Baseline adiabatic effectiveness contours for the flat endwall at (a) $M = 1.0$ and (b) $M = 2.0$ and the contoured endwall at (c) $M = 1.0$ and (d) $M = 2.0$. 
Figure 8.25. Centerline and laterally averaged effectiveness for the flat endwall at $M = 1.0$ compared with data obtained by Lynch et al. [10].

Adiabatic Effectiveness  
Dimensionless Downstream Distance, $s/d$

Figure 8.26. Centerline and laterally averaged effectiveness for the contoured endwall at $M = 1.0$ compared with data obtained by Lynch et al. [10].
Figure 8.27. Area-averaged effectiveness plotted with respect to blowing ratio for flat and contoured endwalls before deposition.

Figure 8.28. Adiabatic effectiveness contours and corresponding deposition photographs (a) before deposition, (b) after 300g, (c) after 600g, and (d) after 900g of wax injection.
Figure 8.29. Area-averaged effectiveness and effectiveness reduction for the flat endwall at $M = 1.0$.

Figure 8.30. Effectiveness contours and corresponding deposition photographs at $M = 1.0$ and $TSP_{med} = 0.3$ for the (a) flat endwall and (b) contoured endwall.
Figure 8.31. Effectiveness contours and corresponding deposition photographs at $M = 2.0$ and $TSP_{med} = 0.3$ for the (a) flat endwall and (b) contoured endwall.

Figure 8.32. Area-averaged effectiveness plotted with respect to blowing ratio for flat and contoured endwalls before and after deposition with $TSP_{med} = 0.3$. 
Figure 8.33. Effectiveness contours and deposition photographs for the flat and contoured endwalls at $M = 1.0$ with (a) $TSP_{med} = 0.3$ and (b) $TSP_{med} = 1.1$.

Figure 8.34. Area-averaged effectiveness plotted relative to $TSP_{med}$ for flat and contoured endwalls at $M = 1.0$. 
CHAPTER 9:
Overview of Deposition Effects on Film-Cooling

The results presented in Chapters 6, 7, and 8 showed the effects of deposition on cooling effectiveness in turbine airfoil regions most susceptible to heat transfer from the hot gas path. The flat plate studies were conducted to determine the effects of deposition on pressure side film-cooling, leading edge studies were conducted to determine the effects of deposition on film-cooling near stagnation, and endwall studies were conducted to determine the effects of deposition on endwall film-cooling with trench and contour geometric modifications. Although the discussions for each region of the airfoil were extensive, an overview of the results for all turbine sections was not presented. The discussion in Chapter 9 provides a direct comparison on the effects of deposition on film-cooling in different regions of the turbine airfoil at various operating conditions.

The effectiveness reduction caused by deposition on the flat plate, leading edge, vane endwall, and blade endwall is plotted with respect to blowing ratio in Figure 9.1. Each cooling location is designated by a different color in Figure 9.1. It is important to note that the same blowing ratios were not tested in every study due to the constraints and objectives of each individual study. Recall from Section 8.2, the 0.8d trench outperformed all other tested trench configurations and is therefore the only one represented in Figure 9.1. The results from the nominal TSP condition for each case are plotted in Figure 9.1.

The only coolant operating condition that was common among all tested configurations was M = 1.0. At M = 1.0, the flat plate cooling configuration was least affected by deposition with only a 6% reduction in effectiveness relative to the baseline case with no deposition. In low blowing ratio cases when jets are attached to the surface, cooling effectiveness is reduced significantly by deposition because it enhances surface roughness which augments mixing with the mainstream. Recall from Section 6.2 that coolant jets at M = 1.0 on the flat plate were separated resulting in mixing with the mainstream. These separated coolant jets provided minimal surface cooling even without deposition. Therefore, the increased roughness caused by deposition did little to affect the cooling effectiveness of flat plate coolant jets that mixed with the mainstream due to separation.

The vane endwall cooling configuration with the 0.8d transverse trench was also minimally affected by deposition at M = 1.0. It is important to note that the vane endwall
results plotted in Figure 9.1 are the total area-averaged effectiveness values taking into account both the passage row and the leading edge row on the vane endwall. Without the trench, deposition reduced effectiveness by 21% as compared to the 7% effectiveness reduction measured at the same blowing ratio with the 0.8d trench configuration. The trench performed better than the case with no trench because cooling effectiveness between holes and downstream of cooling rows was better with the trench than without the trench. Recall from the literature review that deposition decreased with reduced surface temperature [39, 48]. Trenches reduce deposition because particles are solidified by the enhanced cooling near the surface reducing their likelihood of depositing upon impaction.

Out of all geometries tested, the case most affected by deposition was the film-cooled flat endwall in the blade cascade. As shown in Figure 9.1, deposition decreased effectiveness by nearly 50% relative to the baseline case without deposition on the flat endwall in the blade cascade. Deposition had such a negative effect at the low blowing ratio because surface roughness increased mixing with the mainstream. The contoured endwall in the blade cascade performed slightly better than the flat endwall with 40% effectiveness reduction caused by deposition. The contour performed better than the flat endwall, because the surface curvature of the endwall prevented deposition around cooling holes which were located in a valley on the contoured endwall.

Few general trends can be discerned from Figure 9.1. Flat plate effectiveness reduction decreased with an increase in blowing ratio from 28% at $M = 0.5$ to 6% at $M = 1.0$. As previously stated, this trend exists because the effectiveness of a separated coolant jet is minimally affected by surface roughness. Deposition resulted in a small improvement in leading edge effectiveness at $M = 0.5$; however, leading edge effectiveness reduction increased with blowing ratio. Vane endwall effectiveness reduction with no trench was relatively constant through the entire range of blowing ratios tested; however, effectiveness reduction for the vane endwall configuration with the 0.8d trench decreased with an increase in blowing ratio. Effectiveness reduction decreased with an increase blowing ratio with the 0.8d trench because improved cooling between holes and downstream of cooling rows at high blowing ratios reduced deposition.

Effectiveness reduction for both blade endwall configurations decreased dramatically with an increase in blowing ratio. As explained in Section 8.3, the decrease in reduction with
an increase in blowing ratio is consistent with the trend observed in the flat plate results. Separation at the high blowing ratio increased mixing with the mainstream regardless of the surface roughness caused by deposition.

The dramatic variation of effectiveness reduction trends with location, geometry, and operating conditions shows that the effects of deposition on cooling are complex and difficult to predict. Another important consideration to be made when interpreting the results is that effectiveness reduction is not a perfect metric for determining which operating conditions are best. In other words, low effectiveness reduction does not necessarily mean the resulting effectiveness after deposition is high. For example, according to Figure 9.1, the best flat plate cooling condition appears to be \( M = 1.0 \); however, the effectiveness after deposition was higher at \( M = 0.5 \) than at \( M = 1.0 \) because jet separation at \( M = 1.0 \) led to reduced effectiveness even without deposition as discussed in Section 6.2. On a similar note, although effectiveness reduction values indicate that the best leading edge operating condition is \( M = 0.5 \), they do not show that effectiveness near stagnation was worst at \( M = 0.5 \). Cooling near stagnation at \( M = 0.5 \) was virtually non-existent because the high local pressure at stagnation prevented coolant from flowing through the row of holes closest to stagnation.

Perhaps a better way to determine the best operating condition for each cooling location is by normalizing the area-averaged effectiveness. By normalizing the area-averaged effectiveness for a given case with the area-averaged effectiveness of the best case from that location, the best and worst operating conditions for each location can be determined. Figure 9.2 shows the normalized effectiveness values at each operating condition for the flat plate, leading edge, vane endwall, and blade endwall regions. It is important to note that all results plotted in Figure 9.2 are representative of the measured cooling effectiveness after deposition. The operating condition for a given airfoil location with a normalized effectiveness value of 1.0 is the best operating condition for that location. Because the flat plate and leading edge normalized effectiveness values are 1.0 at \( M = 0.5 \), that means the best operating condition (based on area-averaged effectiveness) for both of those locations is \( M = 0.5 \).

The results in Figure 9.2 show that the 0.8d trench configuration on the vane endwall outperformed the vane endwall configuration with no trench at all blowing ratios. Out of all
operating conditions and configurations tested on the vane endwall, the best case was the 0.8d trench configuration at $M = 2.0$.

In the blade cascade, the contoured endwall outperformed the flat endwall at $M = 1.0$; however, at $M = 2.0$ the flat endwall outperformed the contoured endwall. The flat endwall at $M = 2.0$ outperformed all other configurations and operating conditions tested in the blade cascade after deposition. Deposition had little-to-no effect on cooling at $M = 2.0$ because of jet separation while deposition had a dramatic effect on deposition at $M = 1.0$ because of mixing caused by surface roughness. Figure 9.2 shows that the flat endwall effectiveness increased with an increase in blowing ratio while the contoured endwall effectiveness was insensitive to blowing ratio.

Taking into consideration the trends observed in Figures 9.1 and 9.2 and the results presented in Chapters 6, 7, and 8, some recommendations can be made. There is no question that embedding cooling holes in transverse trenches mitigates the negative effects of deposition. The 0.8d trench improved effectiveness with and without deposition at all cooling conditions and locations on the vane endwall. Endwall contouring showed promise with a slight decrease in effectiveness reduction at $M = 1.0$, but more work is required to determine a method to use the contour to maximize its benefit to cooling. The effectiveness of every endwall cooling configuration tested with deposition in the vane and blade cascades with and without trenches and contouring improved with an increase in blowing ratio. For flat plate and leading edge regions, high blowing ratios should be avoided because jet separation increases deposition in cooling hole wakes and decreases cooling effectiveness between cooling rows. Operation at $M = 0.5$ should be avoided on the leading edge because of inadequate cooling of the stagnation region at such a low blowing ratio.

It is interesting that the flat plate and leading edge effectiveness decreased with an increase in blowing ratio while vane endwall and blade endwall effectiveness improved with an increase in blowing ratio. The variation in trends between locations suggests that the complex secondary flow structures play a key role in determining effectiveness trends with deposition. In some cases secondary flow structures improved cooling effectiveness. The pressure gradient within the boundary layer at stagnation pulled separated coolant back toward the endwall downstream of the leading edge row on the vane endwall. Although this leading edge vortex is the main contributor to secondary flow losses, it actually improved
cooling effectiveness near the leading edge on the vane endwall. Results also showed that cooling effectiveness after deposition was better on the flat endwall than the contoured endwall at $M = 2.0$. The ridge of the contoured endwall reduced cross-passage flow and coolant was prevented from spreading through the passage. Cooling effectiveness was better on the flat endwall because cross-passage secondary flows improved coolant spreading across the endwall.

On the other hand, these secondary flow structures are a main contributor to heat transfer from the hot gas path. Lynch et al. [15] determined that endwall contouring minimized secondary flow structures which reduced heat transfer in critical regions of a blade endwall and Bons et al. [47] measured a 27% increase in heat transfer caused by deposition alone on a flat plate coupon. Combined with high blowing ratio coolant operation, deposition induced surface roughness and secondary flows would most likely increase heat transfer coefficients dramatically. There has never been a study to quantify the combined effects of secondary flows, deposition, and blowing ratio on heat transfer. Recommendations for future work regarding the effects of deposition on heat transfer are discussed in Section 10.2.
Figure 9.1. Effectiveness reduction caused by deposition for flat plate, leading edge, and endwall cooling at different blowing ratios.

Figure 9.2. Area-averaged effectiveness normalized by the effectiveness from the best operating condition for that cooling location.
CHAPTER 10:
Conclusions and Recommendations for Future Work

A method was developed to dynamically simulate particle deposition on large scale film-cooled models of turbine components in a laboratory environment. The study involved incorporating well practiced methods for quantifying cooling effectiveness with a new method specifically developed for this study to simulate particle deposition. Particle trajectories were scaled using Stokes number and particle solidification was scaled using a newly developed Thermal Scaling Parameter. Deposition was simulated on a film-cooled flat plate, vane leading edge, and cascade endwall.

The flat plate studies were conducted to test various methods for simulating deposition and quantifying effects of deposition pressure side film-cooling. An improved method for simulating deposition was used to simulate deposition on a film-cooled leading edge model and various film-cooled endwall models. Being the primary focus of this study, endwall experiments were conducted to determine the effects of deposition on film-cooling with and without transverse trenches and endwall contouring. Results from every film-cooling study showed that deposition was dependent on cooling condition and the resulting cooling effectiveness was dependent on deposition. Because the two-way coupled relationship makes deposition and its effects on cooling difficult to predict, an experimental methodology such as the one developed for this study is essential for the development of film-cooling configurations to be used in dirty environments.

10.1 Summary of Deposition Effects on Turbine Film-Cooling

Flat plate experiments were conducted to determine the effects of deposition on airfoil pressure side film-cooling. Wax was injected in varying amounts incrementally to quantify the effects of deposition development on cooling effectiveness. Results showed that deposition reached an equilibrium state at which point cooling effectiveness was no longer reduced by additive deposition on the surface. Development experiments were conducted for flat plate, leading edge, and endwall deposition experiments to confirm that deposition and its effect on cooling reached an equilibrium state in all regions of the airfoil.

When coolant jets were attached to the surface, deposition was prevented in near wake regions for two reasons. First, jets cooled particles making them less likely to deposit
in the event of surface impaction. Second, the momentum of the coolant prevented small particles from colliding with the surface. For the flat plate studies and the blade endwall studies, effectiveness reduction caused by deposition decreased with an increase in coolant velocity. Effectiveness reduction was lower at high coolant jet velocities, because jet separation resulted in mixing with mainstream gases even without deposition-induced surface roughness. In cases where coolant jets are separated, the roughness effect of the deposition is negligible compared to the mainstream mixing effect caused by jet separation.

Leading edge deposition experiments resulted in dramatic coolant hole blockage near stagnation caused by large particle collisions with the surface. Leading edge cooling hole blockage decreased with an increase in blowing ratio; however, jet separation at high blowing ratios resulted in increased deposition in cooling hole wakes. For the leading edge studies, effectiveness reduction increased with an increase in blowing ratio. Although effectiveness reduction was lowest at $M = 0.5$, cooling effectiveness near stagnation, where it is most critical, was highest at $M = 1.0$. Experiments conducted at two different TSP values showed that cooling effectiveness decreased dramatically with an increase in TSP. The added benefit in gas turbine efficiency achieved by elevating turbine inlet temperatures may come with a dramatic decrease in cooling effectiveness caused by molten particle deposition at elevated temperatures.

Endwall deposition experiments were conducted in a vane cascade to determine the effects of deposition on cooling with and without transverse trenches. Trenches with depths of 0.4d, 0.8d, and 1.2d were tested for comparison with an equivalent cooling configuration without transverse trenches. Baseline experiments without deposition showed that cooling effectiveness between holes increased with an increase in trench depth; however, effectiveness downstream of cooling rows decreased with an increase in trench depth. The trench with depth 0.8d performed best by providing superior cooling between cooling holes with adequate cooling downstream of cooling holes. When tested with simulated deposition, all three trench depths outperformed the case with no trench. Improved cooling between holes within the transverse trenches protected the surface from deposition by solidifying particles and ejecting them from the trench. The 0.8d trench was the best configuration tested for mitigating the negative effects of deposition on vane endwall cooling.
Endwall deposition experiments were conducted in a blade cascade to determine the effects of deposition on cooling with and without a non-axisymmetric endwall contour. Deposition patterns on the flat endwall were relatively uniform through the blade passage with moderate collection around cooling holes. On the contoured endwall, however, deposition collected only in regions with increasing elevation. Deposition on the ridges of the contoured endwall was severe while deposition in the valleys was virtually non-existent. Computationally predicted particle accretion patterns agreed well with experimental deposition results on the flat and contoured endwalls. Because cooling holes were placed in a region of decreasing elevation on the contoured endwall, the effects of deposition on cooling effectiveness were mitigated by the contour at M = 1.0.

Experiments were conducted at two TSP values in the blade cascade test section. At the low TSP value, deposition increased surface roughness severely; however, at high TSP values, deposition appeared glassy from molten particle deposition. Effectiveness reduction decreased with an increase in TSP because roughness was less severe at the high TSP than the low TSP.

The overview of deposition effects presented in Chapter 9 showed that the effects of deposition on cooling are highly dependent on various factors, and no single trend can be discerned to simplify the complex nature of the problem. Effectiveness after deposition decreased with an increase in blowing ratio in pressure side and leading edge regions, but increased with an increase in blowing ratio on cascade endwalls. Section 10.2 describes the author’s recommendations for future work that could be conducted to better understand the mechanisms of deposition and to develop sophisticated cooling geometries that can incorporate transverse trenches and endwall contouring to mitigate the negative effects of deposition.

10.2 Recommendations for Future Work

The simulation and measurement methods developed for the current study allowed for experiments to be conducted at numerous operating conditions and geometric configurations with relative simplicity. The results showed that geometric modifications such as transverse trenches and endwall contouring could be used to mitigate the negative effects of deposition on cooling. This section describes how the methods developed for this
study could be used in the future to develop a stronger understanding of deposition and its effects on film-cooling.

The thermal scaling parameter developed for this study served the purpose of characterizing the particle phase for each experiment conducted. Results from the experiments conducted for the current study are in agreement with the literature that molten particles are more likely to deposit than solid particles; however, little is known about the intermediate regime where a particle may be only partially molten. Much could be learned by conducting a set of experiments to closely observe the effects of particle phase on deposition paying close attention to the intermediate regime where particles may have a solid crust and molten core.

Transverse trenches and endwall contouring showed promise as geometric modifications to mitigate the negative effects of deposition. In the current study, only three trench configurations were tested and one endwall contour cooling configuration was tested. It would be worth the effort to test a more in depth matrix of geometries utilizing both trenches and contours in conjunction. Various trench geometries could be placed at different locations on a contoured endwall to determine the most effective utilization of the trench and the contour. These experiments could be conducted using the methods developed from the current study by quantifying surface effectiveness with an adiabatic wall condition.

Endwall contouring reduces secondary flow losses through turbine cascades without deposition; however, the effect of deposition induced roughness on secondary losses is unknown. In addition to quantifying the effects of deposition on cooling with and without endwall contouring, it would also be beneficial to understand the effects of deposition on secondary losses with and without endwall contouring. Laser diagnostic measurement methods such as laser Doppler velocimetry and particle image velocimetry could be used to determine how the secondary flow structures, and thus effectiveness of endwall contouring, are affected by deposition induced roughness.

Although the method used in the current study proved successful at quantifying the effects of deposition on surface cooling effectiveness, it is still unknown how deposition affects the actual metal temperature of the turbine component. The results from the current study showed that deposition had a negative impact on surface cooling effectiveness because of the roughness effect; however, the deposition on the surface could have a positive
insulating effect reducing heat transfer from the hot gases to the cooled turbine component. The method used in the current study could be further developed to quantify the insulating effect of the deposition using a conductive model on which scaled surface temperature, or overall effectiveness, of an operational turbine part can be quantified. The conductive heat transfer effects can be scaled from actual engine hardware by using a material with appropriate thermal conductivity to match the Biot number [92]. This type of model could be used to determine the effects of surface temperature on deposition and also to determine the insulating effect of the resulting deposition on overall effectiveness.

As discussed in Chapter 9, operating at high blowing ratios with deposition induced surface roughness in the presence of violent secondary flows can dramatically increase heat transfer from the hot gas path to turbine components. There has been little research conducted to determine the effects of dynamically simulated deposition on heat transfer. A method should be developed to measure heat transfer coefficients in turbine cascades before and after dynamic deposition simulation to determine the combined effects of deposition induced roughness, secondary flows, and coolant operating conditions on heat transfer. Results from heat transfer experiments would complement adiabatic effectiveness measurements to determine the overall effect of deposition on net heat flux reduction in film-cooled turbine sections.

A method was developed in the current study to perform flow visualization experiments using a high speed camera/laser system to qualitatively observe particle interaction with cooling jets. This method could be used to make measurements at numerous locations and operating conditions for various cooling geometries. Qualitative flow visualization could lead to a better understand of the particle interaction with coolant jets and secondary flow structures. Sophisticated particle tracking algorithms could be developed to quantify particle tracks measured using PIV for direct comparison with computational predictions. By comparing computational predictions with experimentally resolved particle tracks, sophisticated computational methods could also be developed to simulate newly understood deposition mechanisms.

Particle deposition is a highly complex physical problem and an understanding of the deposition effects on cooling is in high demand for further development of future cooling technologies. Although the novel approach described in this document for simulating
deposition is a big step toward understanding particle deposition and its effects on cooling, there is still a vast amount of work to be conducted. With every new set of experiments comes a new discovery and thus new ideas for further development of experimental and computational methods. The results and recommendations presented in this dissertation are a good starting point for future work and further development of cooling technologies insensitive to particle deposition.
REFERENCES


[60] Johnson, D., 1996, Original Pratt & Whitney contact regarding operating conditions and geometric specifications of PW6000 nozzle guide vane.


[94] Southwest Wax, LLC, 2010, Contact with Gary Latto, Technical Sales Manager, glatto@remet.com.
APPENDIX A: Uncertainty Analyses

The following sections describe the methods used to calculate uncertainties for the flat plate, leading edge, and endwall experiments completed for this study. As stated in Chapter 5, the uncertainty propagation method used to perform the uncertainty analyses was developed by Moffat [82]. The following sections describe all of the pertinent measured uncertainties along with the calculated uncertainties of blowing ratio, momentum flux ratio, and adiabatic effectiveness for the flat plate, leading edge, and endwall studies.

A.1 Flat Plate Uncertainty Analysis

Figure A.1 shows a block diagram of the calculation progression for calculating momentum flux ratio for the flat plate study starting with the measured parameters at the top of Figure A.1a. Mainstream and coolant densities along with coolant volumetric flowrate were calculated first. The coolant volumetric flowrate was measured using a laminar flow element (LFE). A correlation shown in Figure A.1a based on measured pressure differential across the LFE was used to calculate the volumetric flowrate. The B and C calibration constants were 5.201 and -0.04025 respectively and were provided by the LFE manufacturer. With the results from the first set of calculations, the mainstream and coolant jet velocities could be calculated. The blowing ratio could then be calculated knowing the mainstream and coolant jet velocities and densities. Figure A.1b shows the equations used for calculating the uncertainties of each parameter for each corresponding step in the calculation progression. Table A.1 shows the bias and precision uncertainties (where applicable) along with the total uncertainties for each measured parameter used to calculate the blowing ratio and momentum flux ratios for the flat plate study. The calculated uncertainties for the blowing ratio and momentum flux ratio are also listed in Table A.1.

Mainstream and coolant pressures were assumed to be near atmospheric as measured by a Setra barometer. The measurement uncertainty associated with the barometer was 0.02% of the full scale which amounted to 10Pa as shown in Table A.1.

The bias uncertainty for type E thermocouples is 0.5°C as listed in Table A.1. Because experiments were conducted at steady state, all temperature measurements were averaged over time which reduced the precision uncertainties. The precision uncertainty is
calculated by multiplying the standard deviation of the measurements by a t-value of 2.0 as appropriate for averages of more than 30 data points.

The cooling hole diameter, \( d_c \), was measured using dial calipers which had measurement resolution up to 0.01mm. The measurement uncertainty of such an instrument is half of the smallest measureable value (0.005mm).

The mainstream pressure differential, \( \Delta p_{\infty} \), was measured using a pitot-static probe. The uncertainty associated with this measurement is dependent on the pressure transducer used to measure the pressure differential. For these measurements a 125Pa pressure transducer was used with a bias uncertainty equal to 1% of the full scale. The precision uncertainty was calculated using a t-value of 2.0 as was used for the temperature measurements.

Figure A.1. Block diagram of (a) uncertainty calculation progression and (b) uncertainty equations for the momentum flux ratio in the flat plate and leading edge studies.
Table A.1. Flat Plate Momentum Flux Ratio Uncertainties

<table>
<thead>
<tr>
<th></th>
<th>Bias</th>
<th>Precision</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p_c$ (Pa)</td>
<td>-</td>
<td>-</td>
<td>10</td>
</tr>
<tr>
<td>$T_c$ (°C)</td>
<td>0.5</td>
<td>0.154</td>
<td>0.523</td>
</tr>
<tr>
<td>$\Delta p_{LFE}$ (Pa)</td>
<td>0.872</td>
<td>0.0817</td>
<td>0.876</td>
</tr>
<tr>
<td>$d_c$ (mm)</td>
<td>-</td>
<td>-</td>
<td>0.005</td>
</tr>
<tr>
<td>$p_\infty$ (Pa)</td>
<td>-</td>
<td>-</td>
<td>10</td>
</tr>
<tr>
<td>$T_\infty$ (°C)</td>
<td>0.5</td>
<td>0.154</td>
<td>0.523</td>
</tr>
<tr>
<td>$\Delta p_\infty$ (Pa)</td>
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<td>1.68</td>
<td>2.09</td>
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<td>$M$</td>
<td>-</td>
<td>-</td>
<td>0.015</td>
</tr>
<tr>
<td>$I$</td>
<td>-</td>
<td>-</td>
<td>0.01</td>
</tr>
</tbody>
</table>

To calculate adiabatic effectiveness as shown in Equation A.1, three temperature measurements were made. The mainstream and coolant temperatures were measured using one thermocouple each while the wall temperatures were measured using an IR camera. The IR camera images were calibrated as described in Chapter 4 with the use of discretely placed thermocouples on the flat plate surface. The uncertainty associated with the adiabatic effectiveness was calculated using Equation A.2.

$$\eta = \frac{T_\infty - T_{aw}}{T_\infty - T_c}$$  \hspace{1cm} (A.1)

$$u_\eta = \left[ \left( \frac{\partial \eta}{\partial T_\infty} u_{T_\infty} \right)^2 + \left( \frac{\partial \eta}{\partial T_{aw}} u_{T_{aw}} \right)^2 + \left( \frac{\partial \eta}{\partial T_c} u_{T_c} \right)^2 \right]^{1/2}$$  \hspace{1cm} (A.2)

The adiabatic effectiveness uncertainty was calculated in a location with high effectiveness and a location with low effectiveness both corresponding to a calibration thermocouple location. As stated in Chapter 5, five images were taken for a given data set. Table A.2 shows the measured temperatures for each of the five IR images for low and high effectiveness locations. The bias error associated with the IR measurement was calculated by taking the difference between the average IR temperature and the average corresponding thermocouple measurement. The precision error associated with the IR temperatures was calculated by multiplying the t-value for five measurements by the standard deviation of the five measurements. The total bias uncertainty was calculated by taking the magnitude of the IR bias and the thermocouple bias and the total precision uncertainty was calculated by
taking the magnitude of the IR precision and thermocouple precision errors. The total uncertainty of the wall temperature was calculated by taking the magnitude of the total bias and precision errors.

### Table A.2. Flat Plate Adiabatic Effectiveness Measurement Uncertainties

<table>
<thead>
<tr>
<th></th>
<th>High $\eta$</th>
<th>Low $\eta$</th>
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<tr>
<td>Temperature (°C, IR Image 1)</td>
<td>35.70</td>
<td>44.50</td>
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<tr>
<td>Temperature (°C, IR Image 2)</td>
<td>35.70</td>
<td>44.60</td>
</tr>
<tr>
<td>Temperature (°C, IR Image 3)</td>
<td>35.70</td>
<td>44.60</td>
</tr>
<tr>
<td>Temperature (°C, IR Image 4)</td>
<td>35.80</td>
<td>44.50</td>
</tr>
<tr>
<td>Temperature (°C, IR Image 5)</td>
<td>35.60</td>
<td>44.70</td>
</tr>
<tr>
<td>Average Temp (IR)</td>
<td>35.70</td>
<td>44.58</td>
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<tr>
<td>IR Standard Deviation</td>
<td>0.063</td>
<td>0.075</td>
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<td>IR t value (5 measurements)</td>
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<td>2.77</td>
</tr>
<tr>
<td>Temperature (°C, Thermocouple)</td>
<td>35.70</td>
<td>44.52</td>
</tr>
<tr>
<td>IR Bias Uncertainty (TC - IR)</td>
<td>0.00</td>
<td>0.060</td>
</tr>
<tr>
<td>IR Precision Uncertainty</td>
<td>0.175</td>
<td>0.207</td>
</tr>
<tr>
<td>Thermocouple Standard Deviation</td>
<td>0.068</td>
<td>0.033</td>
</tr>
<tr>
<td>Thermocouple t value (&gt;30 measurements)</td>
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<td>2.0</td>
</tr>
<tr>
<td>Thermocouple Bias Uncertainty</td>
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<td>0.5</td>
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<tr>
<td>Thermocouple Precision Uncertainty</td>
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<td>0.07</td>
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<td>0.50</td>
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<tr>
<td>$T_{c}$ Uncertainty</td>
<td>0.523</td>
<td>0.523</td>
</tr>
<tr>
<td>$T_{\infty}$ Uncertainty</td>
<td>0.523</td>
<td>0.523</td>
</tr>
</tbody>
</table>

### A.2 Leading Edge Uncertainty Analysis

The uncertainty calculation for the leading edge study was similar to the calculations performed for the flat plate study. The same logical progression of calculations illustrated in Figure A.1 was used for the leading edge study to determine the blowing ratio. Table A.3 shows the bias and precision uncertainties (where applicable) and corresponding total uncertainties for the measured parameters and the calculated blowing ratio. It is important to note that the coolant temperature was measured using two thermocouples which reduced the total coolant temperature uncertainty by a factor of $\sqrt{2}/2$. The mainstream temperature was measured by taking the average of five thermocouple measurements which reduced the total mainstream temperature uncertainty by a factor of $\sqrt{5}/5$. 

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The uncertainties associated with the adiabatic effectiveness measurements for the leading edge study are shown in Table A.4. Uncertainties for adiabatic effectiveness were calculated using the same procedure described in Section A.1; however, the reduced uncertainty associated with mainstream and coolant measurements achieved by increasing the number of thermocouples decreased the adiabatic effectiveness uncertainty as compared to those calculated for the flat plate study.

**Table A.3. Leading Edge Blowing Ratio Measurement Uncertainties**

<table>
<thead>
<tr>
<th></th>
<th>Bias</th>
<th>Precision</th>
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</tr>
</thead>
<tbody>
<tr>
<td>( p_c ) (Pa)</td>
<td>-</td>
<td>-</td>
<td>10</td>
</tr>
<tr>
<td>( T_c ) (°C)</td>
<td>0.5</td>
<td>0.06</td>
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<td>( \Delta p_{LFE} ) (Pa)</td>
<td>6.23</td>
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<td>( d_e ) (mm)</td>
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<td>-</td>
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<td>( p_\infty ) (Pa)</td>
<td>-</td>
<td>-</td>
<td>10</td>
</tr>
<tr>
<td>( T_\infty ) (°C)</td>
<td>0.5</td>
<td>0.06</td>
<td>0.225</td>
</tr>
<tr>
<td>( \Delta p_\infty ) (Pa)</td>
<td>0.623</td>
<td>0.233</td>
<td>0.665</td>
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<tr>
<td>( M )</td>
<td>-</td>
<td>-</td>
<td>0.02 (4.2% at M=0.5)</td>
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**Table A.4. Leading Edge Adiabatic Effectiveness Measurement Uncertainties**

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<tr>
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<td>Temperature (°C, IR Image 3)</td>
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<td>Temperature (°C, IR Image 4)</td>
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<tr>
<td>Temperature (°C, IR Image 5)</td>
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<tr>
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<td>IR Precision Uncertainty</td>
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<tr>
<td>( T_c ) Uncertainty</td>
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<td>( T_e ) Uncertainty</td>
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<td>0.36</td>
</tr>
<tr>
<td>( \eta ) Uncertainty</td>
<td>0.026</td>
<td>0.024</td>
</tr>
</tbody>
</table>
A.3  Endwall Uncertainty Analysis

Figure A.2 shows a block diagram of the calculation progression for calculating momentum flux ratio for the endwall studies starting with the measured parameters at the top of Figure A.2a. Rather than using an LFE to directly measure the coolant flowrate, the blowing ratio and momentum flux ratio were characterized using the ideal blowing ratio using an inviscid analysis. Therefore, the desired blowing ratio and momentum flux ratio were achieved by setting the pressure difference between the coolant plenum and the mainstream. Table A.5 shows the bias, precision, and total uncertainties for each of the measured parameters as well as the calculated blowing ratio and momentum flux ratio for endwall experiments.

Uncertainties associated with quantifying adiabatic effectiveness for the endwall studies are listed in Table A.6. The effectiveness uncertainties were calculated using the same method described in Section A.1 for the flat plate studies. Similar to the leading edge method, five thermocouples were used to measure the mainstream temperature and two thermocouples were used to measure the coolant temperature thus reducing the calculated adiabatic effectiveness uncertainty.
Table A.5. Endwall Momentum Flux Ratio Uncertainties

<table>
<thead>
<tr>
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<th>Bias</th>
<th>Precision</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p^c$ (Pa)</td>
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<td>-</td>
<td>10</td>
</tr>
<tr>
<td>$T^c$ (°C)</td>
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<td>0.229</td>
</tr>
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<td>-</td>
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</tr>
<tr>
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<td>0.06</td>
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<td>0.0218 (2.1% at $M=1.0$)</td>
</tr>
<tr>
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<td>0.0285 (3.0% at $I=0.95$)</td>
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Table A.6. Endwall Adiabatic Effectiveness Measurement Uncertainties

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APPENDIX B: Test Matrices and Operating Conditions

The test matrices and operating conditions for each set of experiments conducted for the current study are presented in Appendix B. Table B.1 shows the test matrix and operating conditions for the flat plate experiments. It is important to note that TSP values are not reported in Table B.1 because the TSP had not been developed prior to conducting the flat plate experiments. The test matrix for the leading edge deposition experiments is shown in Table B.2. Table B.3 shows the test matrix for the endwall deposition experiments conducted in the vane cascade to determine the effects of deposition on cooling with and without transverse trenches. Table B.4 shows the test matrix for the endwall deposition experiments conducted in the blade cascade to determine the effects of deposition on cooling with and without endwall contouring. It is important to note that the values reported in Tables B.1 through B.4 are nominal values and may not be the exact values measured for each experiment. The types of wax as well as a reference to each distributor are listed in the last column of each table.
### Table B.1. Flat Plate Experimental Test Matrix

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<th>Tc (K)</th>
<th>Tp,s (K)</th>
<th>dp,med (µm)</th>
<th>dp,max (µm)</th>
<th>Wax Mass (g)</th>
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### Table B.2. Leading Edge Experimental Test Matrix

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<th>Tp,i (K)</th>
<th>Tp,s (K)</th>
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<th>Pw (kPa, PSI)</th>
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APPENDIX C: Peer Reviewed Journal Publications

The discussion presented in this dissertation was composed from the materials from five articles submitted to the ASME Journal of Turbomachinery. At the time when the dissertation was completed, two of those articles had been accepted for publication, two were under review, and one was in preparation for submission. The article entitled “Effects of Simulated Particle Deposition on Film Cooling [85]” was the first article from this work accepted for publication in the ASME Journal of Turbomachinery and received a best paper award from the IGTI Heat Transfer Committee. The article by Lawson and Thole [85] is shown in double column format in Section C.1. The results from Lawson and Thole [85] were presented at the ASME Turbo Expo 2009: Turbine Technical Conference and Exposition in Orlando, Florida.

The second article accepted for publication in the ASME Journal of Turbomachinery entitled “Simulations of Multi-Phase Particle Deposition on Endwall Film-Cooling [90]” is shown in double column format in Section C.2. The results from Lawson and Thole [90] were presented at the ASME Turbo Expo 2010: Turbine Technical Conference and Exposition in Glasgow, Scotland.
C.1 Effects of Simulated Particle Deposition on Film Cooling [85]

Effects of Simulated Particle Deposition on Film Cooling

Diminishing natural gas resources has increased incentive to develop cleaner, more efficient combined-cycle power plants capable of burning alternative fuels such as coal-derived synthetic gas (syngas). Although syngas is typically filtered, particulate matter still exists in the hot gas path that has proven to be detrimental to the life of turbine components. Solid and molten particles deposit on film-cooled surfaces that can alter cooling dynamics and block cooling holes. To gain an understanding of the effects that particle deposition have on film cooling, a methodology was developed to simulate deposition in a low-speed wind tunnel using a low melt wax, which can simulate solid and molten phases. A facility was constructed to simulate particle deposition on a flat plate with a row of film cooling holes. Infrared thermography was used to measure wall temperatures for quantifying spatially resolved adiabatic effectiveness values in the vicinity of the film cooling holes as deposition occurred. Results showed that deposition reduced cooling effectiveness by approximately 30% at momentum flux ratios of 0.33 and 0.5 and only 6% at a momentum flux ratio of 0.95. [DOI: 10.1115/1.400573]
thickness of the deposition increased, the temperature of the deposition surface exposed to the hot gases became hot enough that it remained sticky allowing both molten and solid particles to deposit. They found that, eventually, the rate that incoming particles deposited on the surface equaled the rate that particles detached because of erosion from incoming particles. Richards et al. [8] studied the deposition of bituminous coal ash in a reactor specifically designed to measure deposition at different gas and surface temperatures. They determined that a decrease in gas temperature increased ash particle size and resulted in increased sticking fraction. Richards et al. [8] also determined that increasing surface cooling decreased sticking fraction when particles were small enough to be efficiently cooled by the thermal boundary layer prior to impaction. In their study, evaluating alternative fuels for gas turbines, Wenglarz and Wright [9] concluded that the gas temperature relative to the melting temperature of an entrained particulate was the relationship best determined the sticking probability of particles that deposited by inertial impaction. This means that molten particles are more likely to stick to a surface than solid particles.

The idea of shifting to alternative fuels has led to numerous studies related to the effects of deposition on cooling effectiveness. Highlighted here are a few studies conducted to determine the effects of idealized deposits on endwall film cooling. Goldstein et al. [10] emulated roughness using cylindrical elements, which were sized and located based on turbine blade inspection. They measured a 10–20% decrease in wall effectiveness caused by roughness at low blowing ratios and a 40–59% improvement in effectiveness caused by roughness at high blowing ratios.

Bono et al. [11] measured surface roughness and generated detailed three-dimensional maps of is-service turbine components. They found that deposition created surfaces made up of random combinations of peaks, valleys, and plateaus. In regions downstream of stream cooling holes, they observed a periodic distribution of fines suggesting that coolant jet prevented deposition in the wake of holes.

Cardwell et al. [12] and Sundaram et al. [13] simulated uniform roughness on a film-cooled endwall using sandpaper. Cardwell et al. [12] found that roughness caused a decrease in cooling effectiveness at high blowing ratios. They attributed the decreased effectiveness to coolant jet separation caused by a roughness induced thicker boundary layer. In a later study, Sundaram et al. [11] conducted a study to determine the effects of deposition on endwall film cooling near the leading edge of an inlet guide vane. In addition to the sandpaper, Sundaram et al. [15] manufactured deposits of ideal shapes and sizes based on measurements made by Bono et al. [11]. Sundaram et al. [15] found that when placed downstream of leading edge cooling holes, small deposits actually enhanced cooling effectiveness by 25% because of the jet interaction with the idealized deposit.

There have been few studies that have attempted to accomplish the task of actually simulating the process of particle deposition. Jensen et al. [14] operated the Turbine Accelerated Deposition Facility (TADF), which was designed to simulate deposition in an accelerated manner. By increasing the concentration of particulate matter in the hot gas path, they could simulate 0.001 h of turbine operation in a 4 h test. The deposition simulation was carried out by injecting particles into a natural gas combustion process. The particle-laden exhaust gases impacted onto a removable coupon that could be analyzed following the test. Particle morphology was validated using scanning electron microscope and x-ray spectroscopy analyses, which showed similarities to deposits found on actual turbine hardware.

Rossi et al. [15] studied the evolution of deposition over time in the TADF by injecting particulate matter in different stages called "burns." They measured the surface topology after every stage and recreated roughness models of the deposits. The effect of roughness on deposition was analyzed. Instability was observed in Stanton number after the first burn. Stanton number then decreased through the third burn before increasing again after the fourth burn. Crooby et al. [16] studied the independent effects of particle size, gas temperature, and surface temperature on deposition in the TADF. They found that deposition rate increased with an increase in particle size, an increase in gas temperature, and an increase in surface temperature.

Al et al. [17] conducted a study using the TADF to investigate the effects of deposition on a film-cooled coupon. The coupon was oriented such that particle-laden exhaust flow impacted the surface at a 45° angle simulating deposition by inertial impaction. In addition to measuring surface coverage, they measured surface temperatures using an ICB camera and were able to conclude that increased deposit height resulted in increased surface temperatures. By observing deposition behavior over time, they concluded that increased surface temperatures accelerated deposition resulting in a nonlinear deposit growth rate with time. They also observed the effects of film cooling blowing ratio on deposition growth and surface temperature. They concluded that an increase in blowing ratio resulted in a decrease in surface temperatures, which reduced deposition in coolant wakes.

The studies described above reveal much about the deposition process; however, there has never been a study versatile enough to both simulate the deposition process and closely observe the deposition process in the vicinity of film cooling holes. The goal of the present study is to simulate the deposition process in a way that allows for a close observation of the interaction between particle-laden flow and a film-cooled endwall.

3 Experimental Methods

Deposition simulation and film cooling effectiveness tests were conducted in a low-speed open loop wind tunnel shown in Fig. 1. Even air was heated by space heaters at the motion side of the blower at the inlet to the plenum. Upon exiting the blower, the heated air impinged onto a splash plate, which prevented the air from propagating through the tunnel without mixing. The flow developed in a duct with a height of 6.9 cm (10.56), a width of 41.7 cm (65.68), and a hydraulic diameter of 11.8 cm (17.8).
(11.60). A row of nine film cooling holes with an L/d ratio of 4, P/d = 3, and r/d = 30 deg was drilled in a removable endwall plate. Placed 150 m (72") upstream of the holes at the test section entrance was a round inlet, a section of honeycomb, a double layer of screens, and a 2 mm trip wire. The endwall plate was constructed from a 2.5 cm thick low thermal conductivity polyvinyl foam block (k=0.25 W/m K) that minimized conduction losses to create an adiabatic wall condition. The surface of the film cooling plate exposed to the flow was covered with a thin layer of black contact paper. The contact paper provided a smooth surface that was easily replicable between test cases. The mainstream flow continued 152 m (2740 ft) downstream from the film cooling plate before exiting through two separate filters. Laser Doppler velocimetry and particle image velocimetry were both used to measure turbulence intensity, Tm, with and without the honeycomb and screens installed. The freestream turbulence intensity was 5.5% with the honeycomb and screens installed and 12.3% without the honeycomb and screens. All baseline and deposition tests were conducted at high freestream turbulence (Tm = 12.3%), while only the baseline that provided a benchmark was completed at low freestream turbulence.

3.1 Adiabatic Effectiveness Measurements. Adiabatic effectiveness was quantified by heating the mainstream air to approximately 315 K and injecting coolant air at approximately 295 K. The mainstream velocity was set at 8 m/s, while the coolant flow was manually controlled to set the momentum flux ratio for a given test. Compressed air was used as the coolant and was routed through a plenum located beneath the film cooling plate as shown in Fig. 1. The coolant volumetric flowrate was measured using a laminar flow element (LFE).

A PIV EFR IR camera was used to acquire a temperature map every 800 ms. The temperature distribution in the film-cooled boundary layer was measured using a thin-film thermocouple wire. A thermocouple wire with a diameter of 0.1 mm was used to measure the temperature at the film-cooling holes. The thermocouple wire was embedded into the film cooling plate using a bonding agent. The temperature measurements were performed using a thermocouple wire with a diameter of 0.1 mm. The maximum and precision uncertainties of the wall temperatures measured by the IR camera were 0.582°C and 0.222°C, respectively. Uncertainties in adiabatic effectiveness were calculated as ±0.037 at an y value of 0.3 and ±0.049 at an y value of 0.9. Uncertainties of major parameters and associated temperature bias and precision values are presented in Table 1.

To validate the measurement technique, adiabatic effectiveness for a single row of cylindrical film cooling holes was measured at Tm = 4.9% and Tm = 12.3%. Figure 2 shows the centerline effectiveness plotted with respect to dimensionless downstream distance for a row of cooling holes having a momentum flux ratio of f = 0.25. The Tm = 4.9% case was measured to compare with literature data and the Tm = 12.3% case was measured as a baseline for deposition studies. Both cases are shown in Fig. 2. Note that the measurements in Fig. 2 were made on an endwall without deposition. Table 3 shows the test conditions for the literature used for comparison in Fig. 2. For the present case, centerline effectiveness increases with a decrease in freestream turbulence.

The Tm = 4.9% case converges with most of the literature between f = 0.25 and f = 0.5 downstream of the cooling row, while the Tm = 12.3% case falls below most of the literature and is in agreement with the data by Porzio et al. [19]. The range of data that could be acquired was limited to 15% downstream of the cooling row because of visual access for IR measurement. Figure 3 shows the laterally averaged effectiveness with respect to dimensionless downstream distance compared with the results from the literature. In contrast to the centerline data, the laterally averaged effectiveness decreases with a decrease in freestream

### Table 1 Uncertainty values

<table>
<thead>
<tr>
<th>Variable</th>
<th>Bias</th>
<th>Precision</th>
<th>Uncertainty</th>
<th>%</th>
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<td>-0.015</td>
<td>3.33</td>
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<td>±0.015</td>
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</tr>
<tr>
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<td>0.015</td>
<td>±0.015</td>
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</tr>
<tr>
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<td>0.015</td>
<td>±0.015</td>
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</table>

### Table 2 Comparison of film cooling conditions

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<tr>
<th>Study</th>
<th>L/d</th>
<th>P/d</th>
<th>n (deg)</th>
<th>f</th>
<th>M</th>
<th>VR</th>
<th>DR</th>
<th>Tn (%)</th>
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</thead>
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<tr>
<td>Minna et al. [22]</td>
<td>1.0</td>
<td>3.0</td>
<td>15</td>
<td>0.21</td>
<td>0.25</td>
<td>0.42</td>
<td>1.20</td>
<td>0.2</td>
</tr>
<tr>
<td>Schmidt et al. [21]</td>
<td>4.0</td>
<td>3.0</td>
<td>15</td>
<td>0.23</td>
<td>0.24</td>
<td>0.60</td>
<td>1.90</td>
<td>0.2</td>
</tr>
<tr>
<td>Lann and Johnson [20]</td>
<td>3.5</td>
<td>2.0</td>
<td>15</td>
<td>0.24</td>
<td>0.25</td>
<td>0.60</td>
<td>1.90</td>
<td>0.2</td>
</tr>
<tr>
<td>Kumar et al. [19]</td>
<td>4.5</td>
<td>4.0</td>
<td>15</td>
<td>0.25</td>
<td>0.25</td>
<td>0.60</td>
<td>1.90</td>
<td>0.2</td>
</tr>
<tr>
<td>Present cases</td>
<td>5.0</td>
<td>3.0</td>
<td>10</td>
<td>0.26</td>
<td>0.28</td>
<td>0.60</td>
<td>1.90</td>
<td>0.2</td>
</tr>
</tbody>
</table>


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turbulence. The Ta = 4.6% results agree reasonably well with Lut-tem and Johnson [20], Schmidt et al. [21] and Kane et al. [13], but are higher than the results obtained by Sinha et al. [22]. Luttem and Johnson [20] studied the effect of cooling hole L/d on adiabatic effectiveness and found that effectiveness generally decreased with decreasing L/d. This is a probable reason for why the laterally averaged results by Sinha et al. [22] are lower than the rest of the benchmark data. Differences between the benchmark case at Ta = 4.5% and the data from the literature exist possibly because of the difference in freestream turbulence levels measured between the present case and the studies in the literature. High freestream turbulence levels could enhance mixing between the jet and the mainstream and therefore increase lateral spreading of the coolant.

3.3 Deposition Simulation and Analysis. The objective of the current study was to develop a methodology that could be used in a laboratory setup to simulate deposition using a low speed wind tunnel. The method was developed as an adiabatic effectiveness test could be conducted on a surface of interest following the deposition simulation. It was necessary to inject particles into a hot gas path that could simulate the deposition of both solid and molten materials. It is important to note that no deposition tests were conducted at high freestream turbulence (Ta = 12.7%), which was necessary to simulate the turbulence levels existing in a gas turbine combustor. Prior to the first cooling studies, various types of particles were injected into a flow with a cylindrical obstacle to observe the deposition behavior. Initially, sand was injected to simulate the deposition of solid particulate. Then, multiple gases having different cooling temperatures were injected to observe their deposition behavior. For the initial tests, red dye was added to the wax to make deposition easier to observe. Photos of sand deposition and wax deposition are shown in Fig. 4. Both materials deposited in a manner that illustrated the horseshoe vortex and saddle point locations on the endwall. The photos in Fig. 4 show examples of large particle deposition by inertial impact on the cylinder and small particle deposition by turbulent diffusion and eddy impact on the endwall. Ultimately, wax was chosen to simulate both solid and molten particles under the assumption that only the largest particles would be in motion upon reaching the deposition test surface. Calculations were performed to determine the distance a particle could travel before solidification [23]. For a particle with a given residence time and heat of fusion, solidification time was directly dependent on the initial wax temperature and the mainstream temperature, and particle size. Based on the achievable test conditions in the low speed wind tunnel, wax having a melting temperature of 328 K was chosen as the material for injection.

To simulate particle trajectories, it was necessary to match the wax Stokes number range to the Stokes number range that exists in actual engine conditions [22]. Table 3 shows the engine conditions and the simulation conditions used for the current study to calculate Stokes numbers. Figure 5 shows the wax particle size necessary to match Stokes number for a given engine particle size. The wax particle diameter required to match the engine condition Stokes number is approximately 13 times the diameter of a coal ash particle in actual engine conditions.

The wax particle size distribution in the low speed wind tunnel was measured using a Malvern Spraytec particle analyzer capable of characterizing aerosol droplets in the size range 0.1–3000 μm. Figure 6 shows the size distribution and corresponding Stokes numbers for the wax particles in the low speed wind tunnel facility. Recall from Fig. 5 that a 1.0 μm wax particle in simulation conditions is scaled by Stokes number to match the trajectory of a 10 μm coal ash particle in engine conditions. In Fig. 6, there is a vertical line at 1.0 μm which intersects the 90% mass passing curve at approximately 99% (as shown on the secondary axis) indicating that approximately 99% (by mass) of the wax particles are smaller than 130 μm in diameter. Figure 6 indicates temperatures of wax particles that are and are
not exposed to the film cooling jet relative to convective distance for a 13 µm particle and a 130 µm particle. The particle temperature relationship to travel distance was calculated using a simple lumped mass approximation. Initially, all particles are in molten form and are losing heat to the mainstream while they cool. The heat loss rate is shown by

\[ Q = A_p(T_p - T_c) \]

By assuming the particle moves at the same velocity as the mainstream, the heat transfer coefficient can be calculated from the analytic solution for conduction from a sphere [22]. The particle temperature as a function of time is shown by

\[ T_p = (T_p(0) - T_c) e^{-\frac{Q}{C_p p}} + T_c \]

When the temperature of a given particle reaches the material solidification temperature, the phase change process begins. During the phase change process, the particle continues to lose heat to the surroundings; however, the particle remains at the solidification temperature of the material. The time that the phase change process takes to complete is calculated using

\[ t_p = \frac{\Delta H_{fg}}{Q} \]

When the particle is exposed to the film coolant, the governing equations remain unchanged; however, the temperature of the surroundings changes to the coolant temperature.

As seen in Fig. 7, the 13 µm particle cools faster than the 130 µm particle and is at the same temperature as the mainstream upon reaching the film cooling row. When the 13 µm solid particle enters the coolant, the particle temperature further reduces due to the heat loss from the particle but does not reduce the particle temperature until the phase change process has completed. When exposed to the coolant, the 130 µm particle will be in solid form 370 s downstream of the cooling row. If the 130 µm particle was not exposed to the coolant flow, it would not solidify until it reached a distance of 900 s downstream of the cooling row. Because the 130 µm particle is in molten form upon reaching the cooling row, it has a greater chance of deposing just downstream of the coolant holes than the 13 µm particle.

Figure 8 shows photographs of coal ash [2] and wax deposits taken using an environmental scanning electron microscope (ESEM). The photos show that wax deposits are spurious like the coal ash particles but are much larger.

Deposition tests were carried out by injecting molten wax particles using a spray gun especially designed for high viscosity fluids. The spray gun was attached to the particle injection port located on the side of the mainstream plenum downstream of the splitter plate. A resistance heater was used to heat the wax reservoir that was instrumented with a thermocouple to control the initial wax temperature prior to injection. For each test case, 460 g of wax was injected right after the initial temperature of the molten wax was set so that particles greater than 50 µm in diameter would remain in molten form for a distance of 100 cm allowing them to reach the film-cooled endwall. After each injection, a series of IR images of the film-cooled endwall was taken and calibrated using the method described in Sec. 2. Initial tests indicated that during deposition there were no significant changes to the emissivity of the surface, which verified the typical calibration procedure for a surface with deposits.

In addition to acquiring IR images, digital photographs were taken between each injection cycle. Digital photos were taken using a Nikon D50x 10.2 megapixel camera that was mounted in the same way as the IR camera. Surface lighting was provided by two fluorescent lamps mounted at opposing angles. To reduce surface glare, a polarizing filter was used with the Nikon 52h. The digital photographs were post-processed using IMAGE software to determine the overall mass fraction covered by wax deposits after each injection.

Figure 9 illustrates the post-processing procedure to capture the deposition starting with an image of the surface without deposition and an image of the surface with deposition. For the post-processing, each digital image is converted to an 8 bit image to which each pixel is assigned a gray value between 0 and 255 corresponding to its brightness. After an image is converted to 8 bit, it can be treated as a two-dimensional matrix. The background is then subtracted by taking the difference between an image of the surface with deposition and an image of the surface without deposition. The resulting image showing only pixels representing deposition is then converted to a binary image. Conversion to binary is accomplished by setting pixels with low gray
values to 0 and pixels with high gray values to 1 resulting in a black and white picture in which black pixels represent deposits. At this point, it is necessary to remove any background noise that was not eliminated by the initial background subtraction. To do this, a median filter is used, which effectively removes any pixel grouping that is smaller than a user-defined pixel value. Then, the deposition area fraction is represented by the ratio of black to white pixels.

4 Discussion of Results

In their study evaluating the independent effects of density ratio, blowing ratio, and momentum flux ratio on film cooling, Thole et al. [25] concluded that a coolant jet will either remain attached, separate and reattach, or separate and remain detached from the endwall depending on momentum flux ratio. In the present study, adiabatic effectiveness tests were conducted at momentum flux ratios of $I=0.23$, $I=0.5$, and $I=0.95$ to observe the effects of deposition for the three different regimes described by Thole et al. [25]. Table 4 shows the conditions for each adiabatic effectiveness test conducted for the deposition study.

Figure 10 shows the contour plots of adiabatic effectiveness for each momentum flux ratio tested prior to deposition. The contours illustrate the effectiveness characteristics of each flow regime. As the momentum flux ratio increases from 0.23 to 0.5, laterally averaged effectiveness decreases implying that separation may be occurring at $I=0.5$. When momentum flux ratio is increased to 0.95, the contours illustrate low effectiveness laterally and along the centerline implying that separation is occurring. Figure 11 shows centerline and laterally averaged effectiveness with respect to dimensionless downstream distance for all three momentum flux ratios. Figure 11 shows that centerline and laterally averaged effectiveness decreases with an increase in momentum flux ratio when separation occurs.

Figure 12 shows effectiveness contour plots and corresponding deposition photographs at $I=0.23$. The appearance of the deposition in each photograph depends on the state of a particle when it impacts the surface. The deposits caused by molten particles show up as black spots, while deposits caused by solid particles show up as white spots. For the initial injection, only molten particles appear to deposit on the surface. Eventually, molten deposition begins to deposit. As more injections occur, solid and molten particles deposit and further increase surface roughness. Although endwall temperatures were measured after all eight injection

<table>
<thead>
<tr>
<th>$I$</th>
<th>$M$</th>
<th>$VR$</th>
<th>$DR$</th>
<th>$T_d$ (%)</th>
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<tbody>
<tr>
<td>0.23</td>
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<td>0.47</td>
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<td>0.69</td>
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<td>0.95</td>
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<td>1.07</td>
<td>12.3</td>
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</table>

Fig. 10. Effectiveness contour plots at momentum flux ratios of (a) 0.23, (b) 0.5, and (c) 0.95

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cycles, the effects of deposition are well represented by the data taken after 1200 g (3 cycles), after 2400 g (6 cycles), and after 3600 g (8 cycles), as seen in Fig. 12. Close inspection of each deposition photograph reveals that deposition concentration appears to be lower directly downstream of the film cooling holes, implying that the coolant jets themselves help prevent deposition. It can also be seen that solid particles build up over time inside the trailing edge of the film cooling holes.

Figure 13 shows the centerline effectiveness at t=0.25 after each wax injection cycle. Centerline effectiveness is affected by deposition after the first three to four injections shown and begins to reach a steady level between five and eight injection cycles. This pattern supports the observation made by Walsh et al. [7] that the deposition process eventually reaches an equilibrium state where the rate of deposition equals the rate that existing deposits are eroded by incoming particles that do not stick. Figure 14 shows the laterally averaged effectiveness after each wax injection cycle for t=0.25. As deposition increases, laterally averaged effectiveness decreases at all downstream locations but appears to have a greater impact on effectiveness in regions within 3 of the cooling hole trailing edge than in regions farther downstream.

The deposition development was quantified by counting deposits using MACE software. For a binary image, the software counts the pixel groupings and categorizes them based on area coverage. An equivalent deposit diameter was then calculated for each pixel grouping. Figure 15 shows a histogram of deposit size as it developed with increased injection for t=0.25. It is important to note that Fig. 15 does not show the histogram of particle sizes but rather deposit sizes. Each deposit could be made up of an aggregation of many individual particles. The figure shows that as more wax is injected, both the size and number of deposits increase because of new particles adhering directly on the surface and onto existing deposits. The histograms indicate that the amount of deposit on the surface approaches an equilibrium state after eight injection cycles (3600 g). Histogram of deposit size for t=0.25 and t=0.55 showed a similar development trend as the histogram for t=0.25 implying that the time to reach a deposition equilibrium state is independent of momentum flux rates.

Figure 16 shows the effectiveness contour plots and corresponding deposition photographs at t=0.5. The well-defined wake region indicates that deposition is mitigated by the coolant jet immediately downstream of the film cooling holes. There also appears to be more deposition of solid particles at t=0.5 than at t=0.25, which could be caused by particles solidifying in the coolant layer prior to impacting the surface. This cooling effect would be enhanced by the higher coolant flow at t=0.5. The contour plots show that effectiveness decreases with an increase in deposition. From the contours, it is evident that deposition shortens the downstream influence of each coolant jet.

Figure 17 shows the effectiveness contour plots and corresponding deposition photographs for t=0.99. Similar to the two previous cases, the photographs show that deposition in regions near the cooling hole trailing edge are mitigated by the coolant jet itself. Deposition in these near wake regions is mitigated for two reasons. First, the coolant jet cools the large molten particles causing them to solidify prior to impacting the surface. Second,
trajectories of the small solid particles are greatly influenced by the coolant jets preventing them from impacting the surface in near-wake regions. The only particles that deposit in near-wake regions are particles that are large enough to maintain their trajectories and withstand solidification by coolant jets.

Figure 17 shows that deposition has less of an effect on cooling at I=0.95 than it does at the lower momentum flux ratios. This lesser effect is because the jet separation at I=0.95 prevents high levels of adiabatic effectiveness downstream of the coolant holes even without deposition on the surface. Because of coolant jet separation at I=0.95, deposition has less of an effect at this momentum flux ratio than for I=0.5 or I=0.23. From the contours, the deposition enhances jet interaction, which ultimately improves lateral spreading of the coolant in far-wake regions. The conduction effect of the wax is very small given it has a relatively low thermal conductivity (k=0.24 W/mK). Because the conduction effect can be neglected, the improved lateral spreading is attributed to the roughness created by large deposits in far-wake regions rather than a conductive effect.

Figure 18 shows the area-averaged effectiveness reduction with respect to deposition area coverage. For I=0.23 and I=0.5, area-averaged effectiveness is reduced sharply through the first three injection cycles. For I=0.23 and I=0.5, the reduction in effectiveness approaches 25% as deposition area coverage increases. On the other hand, at I=0.95, the reduction in effectiveness reaches a maximum of 6% before actually improving to 2% after the final injection cycle. The reduction in effectiveness was low at I=0.95 because the initial cooling provided by these separated jets was very low prior to deposition.

5 Conclusions

A methodology was developed to observe the effects of simulated deposition on film cooling through the use of a low melting temperature wax that solidifies large particles. Quantification of deposition effectiveness was validated by obtaining spatially resolved temperatures to calculate adiabatic effectiveness in the vicinity of a row of film cooling holes. The method for quantifying cooling effectiveness was validated by comparison with adiabatic effectiveness results from the literature.

Deposition was simulated by injecting melted wax particles into the mainstream. Simulation conditions were controlled so that the size range of injected particles varied with Stokes number to the size range of coal ash particles in engine conditions. The present study focused on simulating deposition by initial impact and turbulent diffusion enhanced by eddy injection.

Deposition was simulated for three different momentum flux ratios: I=0.23, I=0.5, and I=0.95 to observe the effects of deposition on cooling for these three different jet conditions. For all momentum flux ratios, deposit coverage appeared to be less concentrated in near-wake regions of each film cooling hole. Deposition in near-wake regions was prevented because coolant jets solidified large particles and changed the trajectories of small particles preventing both from adhering to the surface. Large particles in molten form were the only particles found to have deposited in the near-wake of the jets.

As deposition developed, the deposit size and total number of deposits increased, implying that incoming particles adhered to the surface creating new deposits and to existing deposits increasing their size. Based on deposit size histograms and reductions in adiabatic effectiveness, the deposition process seemed to approach an equilibrium state after eight injection cycles. For I=0.23 and I=0.5, the effectiveness was reduced by 20% as deposition approached an equilibrium state. For I=0.95, effectiveness reached a maximum reduction of 6% and a final reduction after eight injection cycles of 2%. This was a different effect on cooling at I=0.95 because the coolant jets were separated from the surface prior to deposition, which inhibited cooling even without the existence of deposits. Because deposition had a similar effect on the reduction of cooling at I=0.23 and I=0.5, the results indicated no significant advantage of deposition on cooling for these flow regimes.

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Nomenclature

$A_p$ = particle surface area
$b$ = bond length
$C_p$ = particle specific heat
$\Delta t_{m}$ = latent heat of fusion
$\Delta p$ = pressure drop
$d$ = film cooling hole diameter, $d = 0.635$ cm
$F_p$ = particle diameter
$F_{0.5}$ = plenum hydraulic diameter
$D_R$ = density ratio, $D_R = p_d / p_{\infty}$
$h$ = heat transfer coefficient
$I$ = momentum flux ratio, $I = p_{\infty} U_{\infty}^2 / p_d U_d^2$
$L$ = film cooling hole length
$L_c$ = characteristic length for Stokes number
$M$ = Blowing ratio, $M = U_{\infty} / U_d$
$p$ = film cooling hole pitch
$q$ = particle heat loss to surroundings
$S_b$ = Stanton number, $S_b = h / p_d U_d$
$t_c$ = time constant
$T_c$ = coolant jet temperature
$T_a$ = wall temperature
$T_m$ = mainstream temperature
$Tu$ = turbulence intensity, $Tu = u_{\theta} / U_{\infty}$
$U_d$ = coolant jet velocity
$U_d^*$ = particle velocity
$U_{\theta}$ = mainstream velocity
$V_R$ = velocity ratio, $V_R = U_{\theta} / U_{\infty}$
$x$ = streamline distance from cooling hole trailing edge
$y$ = spanwise distance from center cooling hole

Greek

$\alpha$ = film cooling hole incidence angle
$\gamma$ = adiabatic effectiveness, $\gamma = (T_d / T_a - 1)$
$\eta$ = centerline effectiveness
$\delta$ = laterally averaged effectiveness
$h$ = area-averaged effectiveness
$s_{\theta}$ = static area-averaged effectiveness (no deposition)
$\rho$ = coolant jet density
$\varphi$ = particle density
$\rho_{\infty}$ = mainstream density
$\mu$ = gas dynamic viscosity

References


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C.2 Simulations of Multi-Phase Particle Deposition on Endwall Film Cooling [90]

Proceedings of ASME Turbo Expo 2010: Power for Land, Sea and Air
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GT2010-22376

SIMULATIONS OF MULTI-PHASE PARTICLE DEPOSITION ON ENDWALL FILM-COOLING

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ABSTRACT

Demand for clean energy has increased motivation to design gas turbines capable of burning alternative fuels such as coal derived synthetic gas (syngas). One challenge associated with burning coal derived syngas is that trace amounts of particulate matter in the fuel and air can deposit on turbine hardware reducing the effectiveness of film cooling.

For the current study, a method was developed to dynamically simulate multi-phase particle deposition through injection of a low melting temperature wax. The method was developed so the effects of deposition on endwall film cooling could be quantified using a large scale vane cascade in a low speed wind tunnel. A microcrystalline wax was injected into the mainstream flow using atomizing spray nozzles to simulate both solid and molten particulate matter in a turbine gas path. Infrared thermography was used to quantity cooling effectiveness with and without deposition at various locations on a film cooled endwall. Measured results indicated reductions in adiabatic effectiveness by as much as 30% whereby the reduction was highly dependent upon the location of the film cooling holes relative to the vane.

NOMENCLATURE

\( a \) speed of sound
\( A \) surface area
\( C \) chord length
\( C_p \) particle specific heat
\( \Delta h_{\text{ps}} \) specific latent heat of fusion
\( D \) film cooling hole diameter, D=0.46cm
\( \delta_p \) particle diameter
\( h \) heat transfer coefficient
\( l \) momentum flux ratio, \( l = \rho_p U_p^2 / \rho_\infty U_\infty^2 \)
\( L \) film cooling hole length
\( L_p \) characteristic length for Stokes number
\( M \) film cooling blowing ratio
\( M_{\text{blow}} \) blowing ratio of a loss free hole
\( Ma \) Mach number, \( Ma = \frac{U_m}{a} \)
\( p \) static pressure
\( P \) vane cascade pitch
\( P_c \) total pressure
\( Q \) heat loss to surroundings
\( Re \) Reynolds number, \( Re = \frac{\rho U C_p}{\mu} \)
\( S \) nozzle guide vane span
\( S_t \) Stokes number, \( S_t = \frac{\rho_p U_p^2}{18 \mu_x} \)
\( T \) temperature
\( T_u \) turbulence intensity percent, \( T_u = \frac{u_m}{U_\infty} \)
\( U \) velocity
\( V \) volume
\( X, Y, Z \) local coordinates

Greek

\( \eta \) adiabatic effectiveness, \( \eta = (T_\infty - T_{\text{in}})/(T_\infty - T_{\text{ex}}) \)
\( \bar{\eta} \) laterally averaged effectiveness
\( \bar{\eta}_{\text{avg}} \) area-averaged effectiveness
\( \bar{\eta}_{\text{bl}} \) baseline area-averaged effectiveness (no deposition)
\( \rho \) density
\( \nu \) gas dynamic viscosity

Subscripts

aw adiabatic wall
\( c \) coolant
\( \text{ex} \) exit
\( i \) initial
\( in \) inlet
\( p \) particle
\( s \) solidification
\( \infty \) mainstream

INTRODUCTION

Utilizing available coal resources to generate power efficiently while minimizing the production of greenhouse gases has become a major challenge for the current generation of engineers. Among the methods that exist for utilizing
available coal resources in a clean manner is coal gasification. Integrated gasification coal cycle (IGCC) power plants utilize coal gasification technology to produce hydrogen and carbon monoxide-based synthesis gas (syngas) that can be combusted in a combined cycle gas turbine power system.  

A major challenge associated with gas turbine design is developing cooling technologies for turbine components that operate at temperatures above material melting limits. Turbine components must be cooled to withstand mechanical stresses that are exacerbated by the extreme temperatures and pressures that exist in the turbine section. In addition, particles as large as 10 μm that originate from the fuel and air supply can create a potential hazard by depositing on turbine components and impairing sophisticated cooling technologies such as film cooling. It is essential to understand how particle deposition occurs so that film cooling designs can be developed to mitigate the negative effects of deposition. Because full engine tests are costly, methods for simulating deposition in a laboratory setting are needed.  

The region most susceptible to the negative effects of particle deposition is the first stage vane where gas temperatures are highest and the largest particles are most likely to deposit because of their momentum. For the current study, particle deposition is simulated dynamically using a wind tunnel with a large-scale turbine vane cascade. Lawson and Thole [1] showed that deposition is highly dependent on whether a particle is in a solid or molten form upon impacting the surface. Deposition is simulated at different thermal conditions to observe how deposition develops. The effects of deposition on endwall film-cooling are quantified using infrared thermography to measure surface temperatures and thereby calculate adiabatic effectiveness in a spatially-resolved manner.

REVIEW OF RELEVANT LITERATURE

The idea of shifting to alternative fuels has led to numerous studies related to the effects of deposition on turbine cooling. These studies can be categorized based on the methods used to mimic surface deposition. One method used to mimic surface deposition was to condition a surface with deposits based on surface measurements from actual turbine hardware. Cardwell et al. [2], Sundaram and Thole [3], and Somawarshana and Bogard [4] conducted studies related to the effects of roughness on airfoil and endwall cooling relevant to gas turbines for which roughness simulations were based on turbine hardware measurements made by Bons et al. [5].  

Cardwell et al. [2] simulated roughness on a film-cooled endwall using sandpaper on a large-scale turbine airfoil platform to model the roughness measured by Bons et al. [5]. By measuring adiabatic effectiveness at various blowing ratios, Cardwell et al. [2] determined that roughness had little-to-no effect on cooling at low blowing ratios but caused a decrease in cooling effectiveness at high blowing ratios because of the thick boundary layer promoted by the roughened surface.  

An extensive study was conducted by Sundaram and Thole [3] to determine the effects of deposition on endwall film cooling near the leading edge of a nozzle guide vane. Sundaram and Thole [3] manufactured deposits of ideal twodimensional shapes and sizes based on measurements made by Bons et al. [5]. They found that small deposits placed downstream of leading edge cooling holes, actually enhanced cooling effectiveness by 25% for a deposit height of 0.5D and a blowing ratio of 1.5.  

Somawarshana and Bogard [4] conducted a study to determine the effects of varying surface roughness and near-hole obstructions on adiabatic effectiveness. Similar to the methods used by Sundaram and Thole [3], Somawarshana and Bogard [4] placed idealized obstructions upstream and downstream of film cooling holes to simulate deposits caused by large agglomerations of particles that could deposit randomly. The reduction of cooling performance due to deposition was highly dependent on blowing ratio and the location of the deposits relative to the cooling holes. Deposition caused as much as a 30% reduction in adiabatic effectiveness at low blowing ratios and a 46% improvement in adiabatic effectiveness at high blowing ratios. In addition, Somawarshana and Bogard [4] showed that obstructions placed upstream of film cooling holes reduced adiabatic effectiveness while obstructions placed downstream of cooling holes could actually improve effectiveness.  

Another method used to mimic surface deposition is dynamic simulation. Jensen et al. [6], Bons et al. [7], Crosby et al. [8], Ai et al. [9], and Lewis et al. [10] conducted studies in the Turbine Accelerated Deposition Facility (TADF) designed to dynamically simulate deposition in a laboratory environment. These studies simulated particle deposition resulting from the combustion of various alternative fuels in land-based gas turbines.  

By increasing the concentration of particulate matter in the hot gas path, Jensen et al. [6] could simulate 10,000 hours of turbine operation in a four-hour test. Bons et al. [7] observed deposition evolution by operating the TADF through multiple burn cycles for each compound. By recruiting roughness models in acrylic, they measured heat transfer at different stages in the deposition evolution. Through four burn cycles, they found that heat transfer coefficients increased by 27% relative to the smooth baseline. Crosby et al. [8] conducted tests in the TADF and found that deposition rate increased with an increase in particle size, an increase in gas temperature, and an increase in surface temperature.  

Using the TADF, Ai et al. [9] concluded that increased deposition height resulted in increased surface temperatures. They found that an increase in blowing ratio decreased surface temperatures and reduced the amount of deposition in coolant waxes. Lewis et al. [10] recreated film cooling models with deposition based on experiments conducted in the TADF. They found that cooling effectiveness was highest when deposition only existed upstream of the cooling holes.  

A computational study was performed by Sundaram and Tufi [11] to determine the effect of blowing ratio on deposition for a vane leading edge film cooling geometry. They observed the deposition and erosive behavior of 5 μm and 7 μm ash particles and found that coolant jets were successful at minimizing leading edge deposition by cooling particles and pushing them away from the surface and preventing them from depositing. They found that an increase in blowing ratio from 0.5 to 2.0 increased deposition of 5 μm particles by 4%, but decreased deposition of 7 μm particles by 5%.
Much has been learned from the deposition studies described above; however, dynamic deposition simulations are missing in the presence of complex secondary flow structures that exist near endwalls of turbine cascades. Lawson and Thole [1] developed a method to simulate deposition dynamically in a laboratory environment at near standard temperature and pressure conditions (20°C and 101,325 kPa) using wax as the particulate. Wax was used to simulate deposition in the vicinity of a row of endwall cooling holes in a low-speed wind tunnel. Lawson and Thole [1] found that downstream cooling effectiveness decreased and approached an equilibrium state as deposition collected on the surface. They found that deposition reduced downstream effectiveness by as much as 25% at minimum flux ratios of 0.23 and 0.5 and only 6% at a minimum flux ratio of 0.5. Albert et al. [12] used a similar wax injection technique to simulate deposition dynamically on a trailing edge model with film cooling. Similar to Lawson and Thole [1], Albert et al. [12] found that deposition increased in time to reach a quasi-steady thickness. Albert et al. [12] also noted that film cooling blowing ratio and the difference between mainstream and wall solidification temperatures had a strong effect on deposition.

The objective of the current study was to further develop the wax deposition method used by Lawson and Thole [1]. The method was improved such that it could be used in a large scale turbine cascade to observe how deposition occurs on a film cooled endwall under the influence of complex flow structures such as the leading edge vortex. The development of the wax simulation method was discussed including environmental scanning electron microscope photos of fly ash deposition compared with wax deposition in a previous paper published by the same authors of Lawson and Thole [1].

EXPERIMENTAL METHODS

Experiments for the current study were conducted in the large scale turbine cascade model in The Pennsylvania State University Experimental and Computational Combustion Laboratory (PSECCCL). The turbine cascade test section was located in a closed loop wind tunnel shown in Figure 1. Flow through the wind tunnel was supplied by a 50 hp axial fan. Downstream of the fan, the primary flow was cooled by a heat exchanger before dividing into two secondary coolant passages and a mainstream passage. The mainstream flow is the center passage directed through a heater bank that increased the mainstream temperature to 328 K. The secondary flow passages were cooled to 298 K by heat exchangers to achieve a temperature difference between the primary and secondary air of 30 K. After passing through the heater bank, the mainstream flow was directed through a series of screens and flow straightening honeycomb. A turbulence grid located 3.6 chord lengths upstream of the vane cascade was used to achieve 4% mainstream turbulence intensity at the entrance to the vane cascade test section [13].

The turbine cascade test section consisted of two full passages with one center vane, a full neighboring vane, and a half neighboring vane. The vane geometry is a commercial vane described in detail by Kedrowski and Thole [14]. The operating conditions and geometric specifications for the vane cascade are shown in Table 1. The inlet Reynolds number shown in Table 1 was matched with the engine operating conditions to scale the mainstream flow. A film cooled endwall was constructed with low thermal conductivity polyvinyl foam (k=0.025 W/m-K) for which the specific cooling hole pattern designed by Kedrowski and Thole [15] is shown in Figure 2. For the current study, attention was focused specifically on the passage and leading edge cooling rows as identified in Figure 2. The passage row had a compound angle of 30° relative to the flow while the leading edge row had holes that were inline with the incoming flow direction. Low thermal conductivity foam was used to create an adiabatic wall condition so that adiabatic effectiveness measurements could be made. Wire mesh, window screen, and a filter were placed in the wind tunnel to allow for downstream flow visualization of the test section and upstream of the fan to capture injected wax particles that passed through the test section.

![Figure 1. Illustration of wind tunnel facility.](image)

<table>
<thead>
<tr>
<th>Table 1. Geometric and Flow Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Scaling factor</strong></td>
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<tr>
<td><strong>Scaled chord length, C</strong></td>
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<tr>
<td><strong>Pitch/Chord, P/C</strong></td>
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<tr>
<td><strong>Span/Chord, S/C</strong></td>
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<tr>
<td><strong>Hole, L/D</strong></td>
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<tr>
<td><strong>Re</strong></td>
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<tr>
<td><strong>Inlet and exit angles</strong></td>
</tr>
<tr>
<td><strong>Inlet, exit Mach number, Mach</strong></td>
</tr>
<tr>
<td><strong>Inlet mainstream velocity, U</strong></td>
</tr>
</tbody>
</table>

![Figure 2. Endwall film cooling configuration.](image)

For the current study, air from the top cooling passage was used to supply coolant to two separate plenums. One plenum was used to supply coolant to a two-dimensional flush slot angled 45° with the flow to simulate coolant flow through the combustor turbine interface. Another plenum was located
beneath the endwall of the turbine cascade test section and was used to supply coolant to the endwall film cooling holes which were angled at 30° relative to the endwall surface. A variable speed blower located on top of the wind tunnel controlled the airflow to the coolant plenums and could be used to set the coolant flow conditions. Film cooling flows were characterized by $N_{wall}$ of the leading edge cooling hole upstream of the stagnation point. For every test conducted for the current study, the coolant to mainstream density ratio was 1.10.

**Adiabatic Effectiveness Measurements**

Steady state infrared (IR) thermography was used to measure surface temperatures at various film cooling operating conditions with and without deposit. Contrary to conventional methods in which surface temperature measurement resolution is limited by the number of thermocouples on the surface, a FLIR P20 IR camera was used to measure surface temperatures with high spatial resolution. IR images were acquired at five port locations perpendicular to the endwall around the vane leading edge at a distance of 55 cm from the endwall surface. Each IR image had a resolution of 320 by 240 pixels and a viewing area of 24 cm by 18 cm resulting in 715 μm (0.16D) resolution.

During experiments, thermocouple measurements were monitored periodically to determine when steady state was achieved. Upon reaching steady state for a given experiment (approximately 4 hours), IR images were taken at each of the five port locations. At each port location, five images were acquired and calibrated using thermocouples placed in discrete locations on the endwall surface. Images were calibrated by adjusting the background temperature and emissivity until the IR temperatures matched the thermocouple measurements for each corresponding thermocouple location.

Although the endwall was constructed with low thermal conductivity foam, it was still necessary to apply a one-dimensional conduction correction described by Elbridge et al. [16]. To perform the conduction correction, cool air was supplied to the coolant plenums while the upstream slot and the film cooling holes were blocked to prevent external surface cooling. By simulating the temperature difference between the external surface with no cooling and the internal surface exposed to the coolant air, a conduction correction could be imposed on the final effectiveness results. The spatially resolved correction indicated values as high as $\eta = 0.12$ upstream of the test section and $\eta = 0.06$ on the foam endwall areas. To apply a local correction, the spatially resolved values were subtracted from the measured values and normalized.

**Uncertainty Analysis**

An uncertainty analysis was performed for the blowing ratio, momentum flux ratio and adiabatic effectiveness calculations using the uncertainty propagation method described by Moffat [17]. The blowing ratio uncertainty was $\pm 0.0218$ (2.1% at $M=1.0$) and the momentum flux ratio uncertainty was $\pm 0.0255$ (3.8% at $M=0.95$). Uncertainty in the adiabatic effectiveness measurements was directly attributable to the temperature measurement methods used. The bias uncertainty for thermocouples that were used to measure coolant jet and mainstream temperatures was 0.5°C while the precision uncertainty for thermocouple measurements was 0.12°C. The bias and precision uncertainties associated with the adiabatic wall temperatures measured by the IR camera were 0.51°C and 0.34°C respectively. Adiabatic effectiveness uncertainty was $\pm 0.028$ at an $\eta$ value of 0.14 and $\pm 0.031$ at an $\eta$ value of 0.59.

**Dynamic Deposition Simulation and Analysis**

Deposition was simulated dynamically using a two nozzle wax injection system with both nozzles located at 33° span as illustrated in Figure 3. A stream of liquid wax was injected through the center of each nozzle head while two atomizing air jets aimed towards the wax stream served to breakup the liquid wax into a mist of wax particles. A molten wax supply was stored in a heated reservoir and compressed air was used to pressurize the wax reservoir and supply atomizing air. Air and liquid regulators were used to control the atomizing air pressure and liquid wax flowrate independently. For a given wax flowrate, adjustment of the atomizing air pressure varied the particle size distribution. The atomizing air and heated wax lines were routed through the bars of the turbulence grid which housed the two spray nozzles. The turbulence grid was located 3.6 chord lengths upstream of the vane row allowing adequate distance for the injected particles to be evenly distributed across the width of the vane row. Deposition patterns on the endwall were periodic across the span of the test section.

Figure 3. (a) Schematic of wax injection facility and (b) photograph of wax spray nozzle.

To simulate the aerodynamic properties of fly ash particles in an engine, it was necessary to match the Stokes number range between the laboratory and engine environments. A method similar to that used by Lawson and Thole [1] was used to determine that wax particles in simulation conditions must be 10 times larger than fly ash particles in engine conditions to
achieve the same particle trajectory as shown in Figure 4. Therefore, wax particles between 1 μm and 100 μm were required to simulate particle trajectories of 0.1 μm to 10 μm fly ash particles that exist in engine conditions [18].

A Malvern Spraytec particle analyzer capable of characterizing aerosol droplets in the size range of 0.1 to 2000 μm was utilized to measure the size distribution of particles generated using the wax injection system. For a liquid wax pressure of 138 kPa (20 psi) that resulted in a wax flow rate of 1.9 g/s from each nozzle, particle sizes were measured for various atomizing air pressures. Figure 5 shows the particle size distribution for a liquid wax flow rate of 1.9 g/s at atomizing air pressures of 69 kPa (10 psi), 207 kPa (30 psi), 276 kPa (40 psi), and 414 kPa (60 psi). Increasing atomizing air pressure increased jet velocity which broke up the liquid stream and decreased particle size distribution. To achieve particle sizes less than the 100 μm limit necessary to match Stokes numbers, the atomizing air pressure was set to 276 kPa with a liquid flowrate of 1.9 g/s from each nozzle for all deposition simulation experiments.

![Stokes numbers vs. Engine (Fly Ash) Particle Diameter](image)

*Figure 4. Wax particle size range necessary to match Stokes numbers of fly ash particles in engine conditions.*

In addition to simulating particle trajectories, it was also desirable to simulate the thermal properties of particles that exist in engine conditions. Lawson and Thole [1] concluded that the particle phase (solid or liquid) is of particular importance to deposition; therefore, it was necessary to scale the phase of fly ash as it exists in engine conditions. The time it takes a particle to solidify after the combustion process was chosen as the appropriate parameter to scale the phase change process. Biot numbers for fly ash particles and wax particles were calculated to be 0.2 and 0.04 respectively. Because the Biot number for both cases was much less than one, a lumped mass approximation could be used to calculate temperature as a function of time for fly ash and wax particles. The solidification of a particle immersed in a fluid with constant temperature takes place in two separate processes. First, the temperature drops exponentially with time until it reaches the material solidification temperature. The time required for a particle to reach the material solidification temperature, \(t_s\), can then be expressed by Equation 1.

\[
\frac{t_s}{t} = \frac{\alpha_c Y}{h \alpha_s} \left[ \ln \left( \frac{T_m - T_i}{T_m - T_s} \right) \right]
\]  

(1)

Second, the temperature remains at the solidification temperature until the particle loses the equivalent of the latent heat of fusion to the surrounding gases. The time it takes for the particle to lose the heat necessary to change from liquid to solid, \(t_s\), is shown in Equation 2.

\[
\frac{1}{t} = \frac{\alpha_m Y}{h \alpha_s} \left[ T_m - T_i \right]
\]  

(2)

To scale the solidification time from engine conditions to laboratory conditions, it is normalized by the time it takes the particle to travel from the injection location to the nozzle guide vane. The expression for this thermal scaling parameter (TSP) is shown in Equation 3.

\[
TSP = \frac{L + t_s}{t_s / U_{in}}
\]  

(3)

where \(L\) is the distance a particle travels while immersed in the surrounding gases traveling at velocity \(U_{in}\). A particle with a TSP < 1 solidifies prior to reaching the turbine while a particle with a TSP > 1 is in molten form as it encounters the turbine. Because the TSP is highly dependent on particle size, the TSP of the maximum particle size, TSPmax, is used in the current study to characterize the particle phase. By ensuring that the TSPmax is matched between different experiments, the phase of the particle upon reaching the test section can be consistent.

For each experimental condition, two tests were conducted. The first test consisted of the deposition simulation during which wax was injected at steady state conditions to simulate deposition on the test section surfaces. Following the deposition simulation, a thin coat of flat black paint was applied to the walled section surfaces to ensure uniform surface emissivity with a value close to one. The IR data was then calibrated as described in the previous section to account for any changes in emissivity caused by the deposition. After paint application, an adiabatic effectiveness test was conducted to determine the effects of the existing deposition on cooling.

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To simulate deposition, microcrystalline wax with a melting temperature of 351 K was used to simulate deposition. It was necessary to simulate deposition using a high melting point wax to prevent the deposits from melting because of the 328 K mainstream temperature needed to achieve maximum temperature difference between the mainstream and coolant during the adiabatic effectiveness tests. For all experiments in the current study, wax was injected during steady state at either a mainstream temperature of 295 K to achieve a TSP$_{in}$ = 0.3 or a mainstream temperature of 357 K to achieve a TSP$_{in}$ = 1.2. It is important to note that wax was injected at a constant temperature of 364K ±1.3K throughout the duration of each deposition test. As previously stated, wax was injected from two nozzles with a mass flowrate of 1.9 g/s per nozzle amounting to a total wax mass flow rate of 3.8 g/s for a duration of 240 s. A wax mass flow rate of 3.8 g/s for a duration of 240 s results in a particle loading of 56 pppm (parts per million by weight hr) which is experienced by 8000 h of gas turbine operation with a hot gas path particle concentration of 0.007 ppmv. Table 2 shows the particle material properties, operating conditions, and particle scaling parameters for fly ash particles in a gas turbine compared to wax particles in the laboratory wind tunnel [8-23].

![Table 2. Particle Properties and Scaling Parameters](image)

![Figure 6. Fly ash and wax particle temperatures plotted with respect to TSP.](image)

Figure 6 shows particle temperatures plotted with respect to TSP for a 10 μm fly ash particle in engine conditions and a 100 μm wax particle in laboratory conditions. Although fly ash particles and wax particles are subject to different surroundings, the TSP value can be matched to scale the particle solidification time between engine and laboratory conditions. Particle solidification times, $t_{1}$ and $t_{2}$, from Equations 1 and 2 are illustrated in Figure 6.

Surface deposition was quantified using a two-dimensional area coverage technique similar to that used by Lawson and Thole [1]. To quantify the surface coverage, the endwall surface was photographed using a Nikon D40x 10.2 megapixel Digital SLR camera with a polarizing filter. For each experiment, diffuse surface lighting was created by placing a white sheet over the test section and providing upward lighting from inside the wind tunnel. By directing the lights toward the white sheet, light could be diffusely reflected evenly to prevent glare and uniformly light the surface.

After each experiment, photographs were taken through four ports located on the ceiling of the test section. The photos were then stitched together to create a composite image of the surface. The composite image was then cropped and converted to an eight-bit image in which every pixel had a gray value between 0 and 255 representing the light intensity of that pixel. The white wax that deposited on the black surface created excellent contrast and easy deposition identification. The 8-bit surface image was then converted to a binary image in which all black pixels represented deposition. Deposition area coverage could then be calculated by counting the ratio of black to white pixels for a given area of interest. ImageJ [24] software was utilized to perform the digital image processing described above.

Figure 7 shows a composite endwall photo taken at the vane leading edge along with its corresponding 8-bit composite image binary representation. Figure 7d shows a color contour in which each pixel from the 8-bit image was assigned a color based on the pixel gray value. These color surface deposition plots allowed for improved contrast for qualitative deposition analysis. It is important to note that the method described above is a two-dimensional analysis method and color contour levels do not necessarily represent deposition thickness.
DISCUSSION OF RESULTS

For the current study, a test matrix was designed to explore the effects of film cooling momentum flux ratio, $\text{TSP}_{\text{max}}$, and wax spray duration on endwall deposition and adiabatic effectiveness. By increasing the wax spray duration, the total mass of wax injected in a given test was increased. Table 3 shows the test matrix for the current study. It is important to note that the upstream slot flow was 0.75% of the mainstream for every experiment conducted in this study.

<table>
<thead>
<tr>
<th>Test #</th>
<th>$M$</th>
<th>$\dot{m}$</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-</td>
<td>600</td>
<td>Deposition Photograph - Slot Only</td>
</tr>
<tr>
<td>2</td>
<td>-</td>
<td>600</td>
<td>Deposition Photograph - Slot Only</td>
</tr>
<tr>
<td>3</td>
<td>0.25</td>
<td>600</td>
<td>Fleeting Adiabatic Effectiveness</td>
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<tr>
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<tr>
<td>5</td>
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<td>600</td>
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<tr>
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<td>0.25</td>
<td>900</td>
<td>Deposition Photograph - Slot Only</td>
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<tr>
<td>7</td>
<td>0.25</td>
<td>900</td>
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<td>8</td>
<td>0.50</td>
<td>900</td>
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<tr>
<td>9</td>
<td>0.75</td>
<td>900</td>
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<tr>
<td>10</td>
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<td>900</td>
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<td>13</td>
<td>1.75</td>
<td>900</td>
<td>Deposition Photograph - Slot Only</td>
</tr>
</tbody>
</table>

Prior to simulating deposition on a film cooled surface, a flat endwall without film cooling holes was installed to observe the deposition that occurred because of secondary flow structures alone. Figure 8 shows the eight bit composite photographs and corresponding surface deposition plots for the deposition tests conducted using a flat endwall with no film cooling. As shown in Figure 8a, deposition that occurred with $\text{TSP}_{\text{max}} = 0.3$ was isolated mostly to the leading edge region. When exposed to a mainstream temperature of $337 \text{ K}$, the $\text{TSP}_{\text{max}} = 1.2$ particles are soft and more likely to stick resulting in deposition that was more widespread but less dense at the leading edge than the $\text{TSP}_{\text{max}} = 0.3$ deposition. Deposition patterns shown by both cases in Figure 8 illustrate well where deposition occurs because of secondary flow structures near the endwall. The densest deposition collected near the stagnation point downstream of the leading edge vortex. The leading edge vortex is formed by the total pressure gradient that becomes the static pressure gradient as the boundary layer flow stagnates at the same leading edge. The pressure gradient causes the flow to move towards the endwall at which point particles with high inertia do not follow the vortex streamlines resulting in deposition on the endwall between the vortex and the stagnation point. Also illustrated well by the deposition is the saddle point where flow separates from the endwall upstream of the leading edge vortex.

Dense deposition can also be observed inside the trailing edge of the slot upstream of the stagnation region. Local pressures on the endwall at the exit of the slot upstream of stagnation are high resulting in either ingestion or very low coolant velocities exiting the slot. The low coolant velocity is not enough to prevent particles from impacting the surface.

Following deposition tests on the uncooled endwall, the film cooled endwall illustrated in Figure 2 was installed. Prior to simulating deposition, a series of baseline (no deposition) adiabatic effectiveness tests were conducted at momentum flux ratios of $I = 0.25, I = 0.5$, and $I = 3.6$. Contours illustrating spatially resolved adiabatic effectiveness values on the film cooled endwall are shown in Figure 9. Recall that the coolant flow condition is characterized by the momentum flux ratio of the leading edge coolant hole immediately upstream of the stagnation point. The contours show that coolant from the leading edge row is pulled toward the suction side of the vane causing increased effectiveness on the suction side of stagnation. Similar to the findings of Sundaram and Thole [25], effectiveness increases with an increase in blowing ratio. Separation reduces the effectiveness immediately downstream of the leading edge row at $I = 3.6$; however, the leading edge vortex pulls the coolant toward the endwall near the stagnation region resulting in increased effectiveness.
Adiabatic effectiveness was laterally averaged in the streamwise direction at different locations across the pitch of the leading edge cooling row. Figure 10 shows the baseline laterally averaged effectiveness distribution for the leading edge cooling row at the three momentum flux ratios tested. Laterally averaged effectiveness increased with an increase in momentum flux ratio with a large jump in effectiveness between $I = 0.95$ and $I = 3.6$.

Effects of Deposition Evolution

Prior to testing the effects of mainstream temperature and momentum flux ratio, the effect of differing wax spray duration were explored to determine the effects of deposition evolution on cooling effectiveness. Three separate tests were conducted with three wax spray durations at $I = 0.23$. Figure 11 shows the effectiveness contours and surface deposition plots for the case with no deposition along with the cases after the injection of 300g, 600g, and 900g of wax. Similar to the cases with no film cooling, deposition was densest at the stagnation region of the vane and inside the trailing edge of the upstream slot. As expected, surface deposition increased with an increase in wax deposition with a larger difference between 300g and 600g than between 600g and 900g. Comparing directly between effectiveness contours and deposition plots, it is evident that the regions with the highest effectiveness (i.e., coldest surface temperatures) had the least deposition. Ai et al. [9] found that deposition capture efficiency decreased with an increase in blowing ratio and Lawson and Thole [1] observed a decrease in coolant wake deposition with an increase in momentum flux ratio. Ren et al. [5] observed rough patterns caused by the lack of deposition in cooling hole wakes on actual turbine hardware. The lack of deposition in the wakes of coolant holes on the suction side of the vane leading edge in this case is most likely because of high coolant jet velocities preventing surface impaction of particles.

Not only did the coolant holes have an effect on deposition but deposition had an effect on the cooling effectiveness of the leading edge row. The laterally averaged effectiveness plots in Figure 12 show that an increase in deposition causes a decrease in effectiveness everywhere along the leading edge cooling row. Deposition has a greater effect on the cooling effectiveness on the suction side of stagnation than on the pressure side. It is interesting to note that laterally averaged effectiveness values for the 600g and 900g cases are practically the same indicating that effectiveness approaches an equilibrium state between 600g and 900g of injection. Lawson and Thole [1] also observed that effectiveness reached an equilibrium state with an increase in deposition for a simple flat plate film cooling study. As deposition approaches an equilibrium state, the rate at which particles deposit approaches the rate at which particles are eroded from the surface.

Area-averaged effectiveness values were calculated for the region illustrated by the white box in Figure 11a. Figure 13 shows the leading edge area-averaged effectiveness and effectiveness reduction relative to the baseline plotted with respect to deposition area coverage for the three wax spray duration tests. Figure 13 shows that after 300g of wax injection the leading edge cooling effectiveness is reduced by 16%. After 600g, effectiveness is reduced by 25% and after 900g, effectiveness reduction reaches 36%. The difference in

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effectiveness reduction between 600g and 900g of wax injection is only 1%, which is well within the experimental uncertainty indicating that the effect of deposition on cooling effectiveness is approaching an equilibrium state as it did in Lawson and Holt [1].

Figure 12. Laterally averaged effectiveness of the leading edge cooling holes after 300g, 600g, and 900g of wax injection at t=0.23.

Figure 13. Area-averaged effectiveness of the leading edge cooling row plotted with respect to deposition area coverage at t=0.23. The spray duration tests presented in Figures 11 through 13 revealed that 900g of wax injection was adequate to capture the effects of deposition on cooling effectiveness. For the remainder of the tests conducted for the current study, a spray duration of four minutes (900g) was used to explore the effects of momentum flux ratio and mainstream temperature on cooling effectiveness.

Effects of Momentum Flux Ratio
As the momentum of the jet exceeds the momentum of the mainstream fluid, the jet becomes more likely to separate from the surface. Figures 14, 15, and 16 show adjoint effectiveness contour plots and surface deposition plots for $I = 0.23, I = 0.95$, and $I = 3.6$ respectively. The leading edge coolant jets are likely separated at $I = 3.6$, therefore, there are vast differences between the deposition pattern observed at $I = 3.6$ compared with the two lower momentum flux ratios. At $I = 3.6$ deposition is widespread downstream of the leading edge cooling holes on the pressure side of stagnation. It is clear that deposition collects everywhere outside of the coolant jet wakes as seen in Figures 16a and 16b. At $I = 3.6$ deposition is very dense between cooling holes especially near the leading edge of the cooling holes. Mounds of deposition developed between the leading edge coolant holes at $I = 3.6$ implying there was a blockage effect that created a recirculation region.

Figure 14. Effectiveness contours and surface deposition plots at t=0.23 for (a) no deposition, (b) TSP$_{max}$=0.3, and (c) TSP$_{max}$=1.2.

Figure 15. Effectiveness contours and surface deposition plots at t=0.96 for (a) no deposition, (b) TSP$_{max}$=0.2, and (c) TSP$_{max}$=1.2.

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Sundaram and Thole [25] made laser Doppler velocimetry measurements at the stagnation point for the same cooling geometry tested in the current study. The location of the flowfield measurement plane illustrated in Figure 17 is shown by the white line in Figure 16a. The flowfield in Figure 17 shows the recirculation regions upstream and downstream of the coolant holes at M = 2.5. The recirculation regions correspond with deposition collection in the surface deposition plots shown in Figure 16. Deposition also collected downstream of the passage cooling holes most likely because jet separation created a recirculation region immediately downstream of those cooling holes.

Figure 16. Effectiveness contours and surface deposition plots at h = 3.6 for (a) no deposition, (b) TSP<sub>max</sub> = 0.3, and (c) TSP<sub>max</sub> = 1.2.

Figure 17. Vane leading edge flowfield by Sundaram et al. [25] for M = 2.5.

It is apparent from Figures 14-16 that deposition has a negative impact on cooling at all three momentum flux ratios tested. Figure 18 shows the area-averaged effectiveness of the leading edge cooling row for the baseline case along with the TSP<sub>max</sub> = 0.3 and TSP<sub>max</sub> = 1.2 cases. For validation purposes, Figure 18 also shows the area-averaged effectiveness of the same leading edge cooling row geometry tested by Sundaram and Thole [25]. The results in Figure 18 show that leading edge effectiveness increased with an increase in momentum flux ratio with and without surface deposition. Deposition caused a noticeable reduction in area-averaged effectiveness of the leading edge cooling row at all three momentum flux ratios.

Effects of Thermal Scaling Parameter

For the cases illustrated in Figures 14 through 16, the results are similar to those with no cooling in that deposition at TSP<sub>max</sub> = 1.2 appears less dense at the stagnation region but more widespread than deposition at TSP<sub>max</sub> = 0.3. Recall that TSP was varied by changing the mainstream air temperature. Effectiveness contours show that deposition at TSP<sub>max</sub> = 1.2 has a more negative impact on cooling than deposition at TSP<sub>max</sub> = 0.3. Deposition at TSP<sub>max</sub> = 1.2 collected densely around cooling holes near stagnation and appeared to partially block some cooling holes. This blocking effect is the explanation for the increased reduction in effectiveness observed at TSP<sub>max</sub> = 1.2. Particles with TSP<sub>max</sub> = 1.2 have higher temperatures than particles with TSP<sub>max</sub> = 0.3. Particles with higher temperatures are softer and stickier allowing them to deposit more readily upon impact in the vicinity of cooling holes. Deposition at TSP<sub>max</sub> = 1.2 is less dense near stagnation than deposition at TSP<sub>max</sub> = 0.3 because the sticky particles entrained in the leading edge vortex at TSP<sub>max</sub> = 1.2 are likely to deposit on the vane surface before impacting the endwall. The photographs in Figure 19 at I = 0.23 for both TSP<sub>max</sub> conditions illustrate the thick deposition on the vane surface at TSP<sub>max</sub> = 1.2.

Area-averaged effectiveness was calculated for the passage row in addition to the leading edge row. The region used for taking the passage row area-average was 11D by 2D and is indicated by the black box in Figure 11a. Figure 20 shows the area-averaged effectiveness reduction relative to the baseline case for the passage and leading edge rows at TSP<sub>max</sub> = 0.3 and TSP<sub>max</sub> = 1.2 plotted with respect to momentum flux ratio. The results show that leading edge effectiveness reduction decreases with an increase in momentum flux ratio while passage row effectiveness reduction increases with an increase in momentum flux ratio.
In addition to the different effects of momentum flux ratio between leading edge and passage cooling rows, the effect of $TSP_{\text{m}}$ is much greater on the passage row than on the leading edge row. Passage row effectiveness reduction reaches 32% at $TSP_{\text{m}} = 1.2$ and only as high as 23% at $TSP_{\text{m}} = 0.3$. The difference in effects observed between the passage row and the leading edge row is most likely because the leading edge coolant is less susceptible to separation because of the presence of the leading edge vortex. The leading edge vortex holds the coolant on the endwall which allows for increased effectiveness and reduces downstream deposition buildup. The passage row coolant, on the other hand, is more susceptible to separation which leads to deposition accumulation close to the trailing edges of coolant holes decreasing effectiveness. The extent of deposition downstream of the coolant holes on the passage side is dependent on the $TSP_{\text{m}}$ value. The soft, sticky particles that exist at $TSP_{\text{m}} = 1.2$ lead to dense deposition downstream of passage row cooling holes ultimately leading to increased effectiveness reduction.

**Figure 20. Area-averaged effectiveness reduction for leading edge and passage cooling holes.**

**CONCLUSIONS**

A method was developed to dynamically simulate solid and molten particle deposition using wax in a large scale turbine cascade. The effects of wax spray duration, momentum flux ratio, and mainstream temperature on endwall film cooling effectiveness were quantified. A thermal scaling parameter was developed to characterize the phase of particles immersed in the mainstream gas path. The thermal scaling parameter was used to scale the phase of the particles from engine to laboratory conditions.

At the lowest momentum flux ratio, three tests were conducted with different wax spray durations to observe deposition development and measure the effects of deposition evolution on cooling. Little change in effectiveness and surface area coverage between 600g and 900g of wax injection indicated that deposition approached an equilibrium state.

Areas of dense deposition were observed at the stagnation region of the vane, downstream of the passage cooling rows, and upstream of leading edge cooling rows at high momentum flux ratios. Based on these observations compared with laser Doppler velocimetry measurements from a previous study [19], dense deposition most likely collects in areas where high vorticity causes particle impaction on the endwall.

Area-averaged effectiveness values were calculated for the leading edge and passage cooling rows. Results showed that leading edge cooling row effectiveness reduction was comparable for both low and high thermal scaling parameter values and decreased with an increase in momentum flux ratio. On the other hand, the passage cooling row experienced an increase in effectiveness reduction with an increase in momentum flux ratio. The difference in effectiveness dependence on deposition between leading edge and passage cooling rows was because of the jet separation at the leading edge cooling row was suppressed at high momentum flux ratios by the leading edge vortex. The buildup of deposition downstream of passage cooling holes led to a greater decrease in effectiveness with the high thermal scaling parameter than with the low thermal scaling parameter.
This study has shown that endwall cooling is highly sensitive to deposition. Excess deposition could lead to reduced cooling effectiveness and possibly turbine component failure. It is essential to understand the driving mechanisms behind deposition so that advanced film cooling configurations can be designed to mitigate deposition.

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PUBLICATIONS


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