COOLING OF A TURBINE VANE ENDWALL
THROUGH CONTOURING AND FLOW INJECTION

A Dissertation in
Mechanical Engineering

by

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ABSTRACT

High oil prices and environmental concerns serve to drive up the efficiencies of land based, power generation gas turbines. Increasing efficiencies requires raising the temperature of the air entering the turbine section of the engine. Turbine components must be protected from the increased air temperatures by advanced cooling designs that provide coolant to the hot flow surfaces. Secondary flows reduce the effectiveness of coolant injected along the vane endwalls as well as increase endwall heat transfer. Endwall contouring and strategic coolant injection can alter secondary flows to allow for improved endwall cooling, ultimately allowing for higher engine efficiencies.

This research initially focused on understanding the flow physics and subsequent cooling characteristics of an axisymmetric contoured vane passage. Results indicated that coolant injected from discrete holes provided lower effectiveness values on the contoured endwall in comparison to the flat endwalls of the planar and contoured passages. Coolant coverage from the upstream interface slot, however, was spread over a larger area of the contoured endwall in comparison to the flat endwalls as the interface slot was oriented closer to the plane of the contoured endwall. Seeking a fundamental understanding of interface slot coolant injection, further investigation into the effects of orientation and position of slot injection on secondary flows and the net heat flux experienced by a vane endwall was conducted. Results indicated that cooling effectiveness levels can be improved and the horseshoe vortex reduced in size by moving the interface slot closer to the passage inlet. At large injection rates, reducing the slot orientation resulted in the removal of the horseshoe vortex and a subsequent reduction in passage secondary flows leading to a reduction in the average heat load experienced by the endwall.
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Nomenclature

$A_h$ area of film cooling hole, [m$^2$]

$C$ vane chord, [m]

$C_{ax}$ axial vane chord, [m]

$C_D$ discharge coefficient, $m/m_{is}$

$C_L$ lift coefficient, $F_L/(0.5\rho_xU_{x,in}^2)$

$C_p$ pressure coefficient, $(P_s - P_{s,in})/0.5\rho U_{x,in}^2$

$c_p$ specific heat of air, [J/kgK]

$d_{eff}$ effective leading edge diameter [m]

$F_L$ lift force on vane in y-direction, [N]

$h$ heat transfer coefficient, $q''_{conv}/(T_w - T_{\infty})$, [W/m$^2$K]

$I$ momentum flux ratio, $\rho_c U_{c}/\rho_{\infty} U_{\infty}$

$M$ blowing ratio, $\rho_c U_{c}/\rho_{\infty} U_{\infty}$

$Ma$ mach number, $U/U_a$

$m$ mass flow rate, [kg/s]

$\dot{m}$ mass flux ratio, $\dot{m}_c/\dot{m}_{passage}$

$N$ total number of measurements

$NHFR$ Net heat flux reduction, $(q_{o"}'-q_{c"}')/q_{o"} = 1-h_c/h_o(1-\eta/\phi)$

$n_i$ number of measurements in a given bin size

$Nu$ Nusselt number, $q''_{conv}C_{ax}/(T_w - T_{\infty})k_{air}$

$P$ vane pitch or pressure, [m] or [Pa]

$P_o$ total gauge pressure, [Pa]

$PDF$ probability density function, (measurements above threshold/N)

$q''$ heat flux, [W/m$^2$]

$Re$ Reynolds number, $UC/\nu_{air}$

$S$ vane span or distance along vane circumference, [m]

$s$ distance along streamline, [m]

$St$ Stanton Number, $h/\rho U_{in}c_p$

$T$ static temperature, [K]

$t$ time, [s]
Tu  turbulence intensity, \((U_{rms}^2 + W_{rms}^2)^{1/2}/U_{\infty}\)

U  streamwise velocity, [m/s]

V  pitchwise velocity, [m/s]

W  spanwise velocity, [m/s]

w  slot width, [m]

x, X  axial direction or distance, [m]

y, Y  pitch direction or distance, [m]

z, Z  span direction or distance, [m]

Vs'  streamwise velocity in x-y plane, \(U\cos\Psi_{inv} + V\sin\Psi_{inv}\), [m/s]

Vs  final streamwise velocity, \(V_s\cos\Phi_{inv} - W\sin\Phi_{inv}\), [m/s]

Vz  normal velocity, \(V_z\sin\Phi_{inv} + W\cos\Phi_{inv}\), [m/s]

Greek:

\(\delta\)  boundary layer thickness, [m]

\(\delta^*\)  displacement thickness, [m]

\(\eta\)  corrected adiabatic effectiveness, \((\eta_{exp} - \eta_o)/(1 - \eta_o)\)

\(\eta_{exp}\)  measured adiabatic effectiveness, \((T_{\infty} - T_{aw})/(T_{\infty} - T_c)\)

\(\theta\)  momentum thickness

\(\rho\)  air density, [kg/m\(^3\)]

\(\nu\)  kinematic viscosity, [m\(^2\)/s]

\(\zeta\)  energy loss coefficient, \((U_{2,2,ms}^2 - U_{2,2,ms}^2)/U_{2,2,ms}^2\)

\(\Psi_{inv}\)  inviscid turning angle in x-y plane, \(\tan^{-1}(V_{inv}/U_{inv})\)

\(\Phi_{inv}\)  inviscid turning angle in x-z plane, \(\tan^{-1}(W_{inv}/V_{inv}\)s\_,inv\)

\(\alpha\)  flow angle

\(\phi\)  non-dimensional metal temperature, \((T_{\infty} - T_w)/(T_{\infty} - T_{c,\text{internal}})\)

\(\tau_f\)  characteristic fluid time scale, \((\delta/U_{\infty})\), [s]

\(\Omega_z\)  vorticity about the z-axis, [s\(^{-1}\)]

\(\lambda_T\)  threshold swirling strength, \((U_l/0.39d_{eff})^2\), [s\(^{-2}\)]

Subscripts:

2  exit of vane passage beyond the throat
a  speed of sound
avg  average
aw  adiabatic wall
c  coolant
cnd  conduction
conv  convection
d  Reynolds number based on effective diameter
exit  exit of vane passage at throat
FC  film cooling
i  instantaneous velocity
in  measured at inlet of the vane cascade
inv  inviscid
is  isentropic condition
l  local value
m  time-averaged measurement
max  maximum length along vane circumference
ms  at midspan
o  refers to no cooling case
rad  radiation
rms  root mean square
s  static measurement
slot  measured in the slot or slot plenum
w  wall measurement
∞  freestream
θ  based on momentum thickness
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To my mom, Donna Thrift, I am sorry for all the worry and stress I have put you through. I would not have made it through my life let alone the last six years without your love and support. To those in my family that I have lost, my uncle Eric Albright, my pawpaw Odell Thrift, my doll Gladys Thrift, my grandfather-in-law Willard Bess, and my father Tommy Thrift, I miss you all and I hope I have become the man you wanted me to be. Daddy, I promise to take care of the family to the best of my ability. I also promise that one day I will own that fishing boat that you always dreamed of having.

This dissertation is dedicated to my late father, Tommy Thrift
Chapter 1

Introduction

1.1 Background

Land based gas turbines provide high efficiencies and power-to-weight ratios for electrical power generation. Although several variations of the gas turbine exist for different applications, every engine is made up of the same five sections as shown in Figure 1.1 which presents a typical power generation gas turbine.

![Figure 1.1. A schematic of a land based, power generation gas turbine highlighting the five basic components inherent to all gas turbine engines [1].](image)

The air is ingested into the engine through a rounded inlet to prevent separation or as shown in Figure 1.1 a fan-shaped inlet is used to impart a circumferential motion to the incoming flow. The air then passes through a compressor where work is performed on
the fluid by the rotation of the compressor blades to increase the pressure over several stages. Next the compressed air enters a combustion chamber where, with the addition of fuel, combustion takes place drastically increasing the temperature of the air. This high temperature air then expands through the turbine section where the turning of the fluid through the turbine blades places a lateral force on the blades as a result of changing the direction of the fluid momentum. The lateral force results in a torque on the blades and the subsequent rotation of the power shaft. The work extracted from the fluid in the turbine section provides power to the compressor along this common shaft. Note that like the compressor, between each stage of rotating blades in the turbine are stationary airfoils called vanes which redirect the air to the optimal angle of attack on the subsequent blade stage. The work not used by the turbine to power the compressor is transmitted as shaft power. Finally, the air is exhausted. A common variation in most power generation gas

**Figure 1.2.** A schematic of a land based, power generation gas turbine with concentric, dual shafts driven by a low and high pressure turbine [2].
turbines make use of a concentric, dual shaft design where a high pressure turbine drives the compressor and a low pressure turbine is used to drive the power shaft as shown in Figure 1.2. In some variants, the low pressure turbine also drives a low pressure compressor located upstream of the high pressure compressor.

The Brayton cycle shown in Figure 1.3 is the ideal cycle describing the thermodynamic cycle of a gas turbine. Based on the relation between turbine inlet temperature and thermal efficiency derived from the Brayton cycle analysis, engine designers have consistently increased the turbine inlet temperature to gain efficiency.

![Ideal Brayton cycle with definition of thermal efficiency.](image_url)

In modern gas turbines, turbine inlet temperatures (~1500°C) are higher than the melting temperature (~1200°C) of the turbine components [3]. The most critical turbine component is the first stage, nozzle guide vanes which are directly downstream of the
combustor and thus subject to the highest temperatures within the turbine. To maintain the integrity of the first stage vanes in this extreme environment, discrete film cooling holes are placed on both the vane surface and the vane endwalls as shown in Figure 1.4. Relatively cool air (~300°C) bled from the compressor is used to pressurize internal plenums within the vane as well as plenums below the endwall platform to supply coolant to the film cooling holes. An interface gap between the first stage vanes and combustor section allows for thermal expansion between the two components. To prevent hot freestream gas from being ingested into the internal cavities, the cavities are pressurized with compressor bleed air. The bleed air leaks from the interface gap due to the pressure difference and serves as coolant on the vane endwalls. Coolant from discrete film cooling holes and the interface gap sheathes the first stage vanes and endwalls in a thin film of cool air to protect the components from the hot freestream flow.

Figure 1.4. Turbine vane cascade with film cooling holes and interface gap [4].
1.2 Motivation

The first stage vanes in a gas turbine engine are subject to a unique environment where fluid dynamics and heat transfer combine to create an interesting combination of physics. The vane passage behaves as a nozzle where the incoming flow is accelerated through a streamwise area reduction. In addition, the curvature of the vanes forces the flow to turn through the passage resulting in an additional centripetal acceleration. The vane passages are turned to redirect the air to the subsequent blade row at an appropriate angle of attack as the air was turned from the preceding blade row to extract work. Except for the thin boundary layers on the vane surfaces and the wake of the vane, the freestream flow outside of the endwall boundary layers is inviscid and two-dimensional. Near the endwalls, however, the velocity deficit within the boundary layer results in a radially directed pressure gradient toward the endwall at the vane stagnation and a strong endwall crossflow from pressure to suction side of the passage both of which serve to produce a three-dimensional flowfield.

The junctions between the vanes and their bounding endwalls serve as the initial locations where three-dimensional flow structures form. These structures are referred to as secondary flows as they do not follow the inviscid freestream flow outside the boundary layer. Secondary flow structures propagate through the passages, increasing the total pressure loss through a vane stage through turbulent mixing and dissipation. Secondary flows are rotational in nature thus they enhance mixing of the near wall flow with the hot freestream gas. High turbulence levels as a result of mixing increase heat transfer to the vane endwalls and reduces the effectiveness of any supplied coolant. The endwall boundary layer also gives rise to a stronger turning of the endwall flow toward
the suction side of the passage in comparison to the inviscid flow. Endwall crossflow is the result of the velocity deficit within the boundary layer, forcing endwall streamlines to follow a smaller radius of curvature through the passage than the freestream to maintain the balance between centripetal acceleration and the cross passage pressure gradient [5].

A classic example of the secondary flow structures that exist within a turbine vane passage was first produced by Langston et al. [6] as shown in Figure 1.5. These structures are a result of the approaching endwall boundary layer interacting with the leading-edge of the vane and endwall junction. The endwall boundary layer separates upstream of the vane leading-edge due to the adverse pressure gradient imposed by the blockage. The fluid outside of the endwall boundary layer has not experienced viscous losses and therefore stagnates to a higher pressure than the fluid within the boundary layer. The resulting pressure gradient forces the roll up of a vortex ahead of the

Figure 1.5. Secondary flow structures for a vane passage, (Langston et al. [6]).
stagnation line. This structure is referred to as the horseshoe vortex. Legs of the vortex are convected around the stagnation line and into the passage. The pressure side leg of the horseshoe vortex merges with endwall crossflow to form what is referred to as the passage vortex. This vortex gains in strength and size as it moves through the passage, crossing the passage pitch and reaching the suction side surface at the minimum pressure point located at the throat of the passage. The suction side leg of the horseshoe vortex is convected along the corner of the vane-endwall junction.

The behavior of the corner vortex once it has intersected with the passage vortex is a subject of debate and depends on the particular vane geometry. Langston et al. [6] illustrates the corner vortex as staying suppressed between the vane-endwall junction and the passage vortex thus forming a counter-rotating vortex pair. Sharma and Butler [7] show that the suction side leg wraps itself around the passage vortex instead of adhering to the suction side as shown in Figure 1.6.

**Figure 1.6.** Secondary flow structures for a vane passage where the suction side leg of the horseshoe vortex wraps around the passage vortex (Sharma and Butler [7]).
Goldstein and Spores [8] proposed that the suction leg of the horseshoe vortex stays above the passage vortex and travels with it as shown in Figure 1.7. Kawai et al. [9] modified the model suggested by Goldstein and Spores [8] by adding a small vortex located in the corner of the vane-endwall junction that was induced by the rotation of the passage vortex as shown in Figure 1.8.

**Figure 1.7.** Secondary flow structures for a vane passage where the suction side leg of the horseshoe vortex travels over the passage vortex (Goldstein and Spores, [8]).

**Figure 1.8.** Secondary flow structures for a vane passage where the passage vortex induces a corner vortex (Kawai et al. [9]).
Secondary flows serve to enhance heat transfer on the endwalls in addition to sweeping in any prospective coolant introduced upstream of the vanes into the freestream, rendering the coolant useless in cooling the endwall. Evidence of the damage caused by secondary flows is shown in Figure 1.9 where the horseshoe vortex has created a trench near the leading-edge as the endwall material was overheated and made brittle allowing for the eventual cracking and erosion of endwall material.

![Figure 1.9. Erosion of endwall material near the vane leading-edge as a result of the horseshoe vortex [5].](image)

Figure 1.10 also shows corrosion damage evident by the brownish discoloration of the surfaces due to overheating of a first stage vane. Note that damage to the endwall is concentrated near the suction side of the vane leading-edge where the horseshoe vortex and passage vortex exists. The distribution of the corrosion highlights the curved path taken by the passage vortex across the passage.
It is clear that film cooling is essential to maintaining the performance and durability of the first stage vane cascade. Secondary flows, however, complicate the cooling of the vane endwalls. To improve the ability of film cooling features to protect the endwall surfaces from the hot freestream gas it is necessary to reduce or eliminate the secondary flows. A fundamental understanding of how endwall designs and cooling features influence secondary flows can help to improve upon existing gas turbine efficiencies. Air bled from the compressor is a parasitic drain on the engine thus a net gain in efficiency by raising turbine inlet temperatures is achieved through effective utilization of coolant.

Concerning endwall designs, an area reduction from the combustor section to the turbine section often takes place over the first stage vanes in the form of the sloping of one endwall as shown in Figure 1.11. As the endwall sloping is two-dimensional and symmetric about the engine axis of rotation the endwall design is referred to as an
axisymmetric endwall contour. The axisymmetric contour changes the flow through the passage in comparison to a planar passage where both endwalls are parallel. In addition to stronger flow acceleration through the axisymmetric passage in comparison to a planar passage, the approaching flow must turn at the inlet to the passage in response to the obstructing sloped endwall.

Figure 1.11. Schematic of transition from combustor to turbine section over the first stage vanes [11].

The injection of coolant from the interface gap upstream of the vane passage serves to alter secondary flows as the coolant interacts with the endwall boundary layer. The interface gap must be present to allow for thermal expansion, thus the injected coolant is
generally not considered in the design of the endwall cooling scheme. The effects of coolant injection from the upstream interface gap are thus understudied in comparison to discrete film cooling holes. A detailed investigation into the interface gap parameters that influence the performance of the injected coolant can reveal new physics of altering secondary flows.

The work presented in this dissertation examines the influence of an axisymmetric passage and coolant injection from an upstream interface gap on the endwall secondary flows and the subsequent impact on the cooling of the vane endwalls. The overall goal of the research is to understand the fundamental physics behind secondary flow generation and mitigation for realistic endwall designs and film cooling features.

1.3 Specific Research Questions

While past researchers have investigated endwall cooling with an axisymmetric contoured vane passage, no study has been conducted where the results from the axisymmetric contoured passage are compared to those of a corresponding planar passage. To gain a fundamental understanding of the effects of axisymmetric contouring on endwall cooling both a planar passage and a contoured passage were studied in this dissertation to contrast the flow physics between the two different cases. The differences in the flowfields between the planar and contoured passages included a larger axial pressure gradient due to the additional streamwise acceleration of the freestream flow as a result of the passage area contraction through the sloping of one endwall in addition to the curvature of the streamlines away from the contoured endwall. This unique
comparison between a planar and contoured passage allows the following question to be answered:

1) What are the effects of an axisymmetric contoured passage on the performance of coolant injection from discrete film cooling holes and from the combustor-turbine interface slot in comparison to a planar passage?

   Investigating the above research question revealed that coolant coverage from the upstream interface slot was the greatest on the sloped endwall as the orientation of the interface slot relative to the sloped endwall was reduced in comparison to the flat endwalls. This observation prompted further investigation into the injection of coolant from the interface slot to understand what slot parameters are most important to altering secondary flows and improving coolant performance. Specifically this investigation sought to answer the following question:

2) What are the influences of injection flow rate, position, and orientation of the combustor-turbine interface slot on endwall secondary flows and how do the changes to the secondary flows effect the subsequent performance of the injected coolant?

   The effects of endwall contouring and flow injection on endwall secondary flows were determined through predictions and time-resolved measurements of the stagnation plane flowfield at the leading-edge of the vane where the secondary flows originate in the formation of the horseshoe vortex. The subsequent cooling of the vane endwall as a result
of endwall contouring and flow injection was ascertained through predictions and measurements of adiabatic effectiveness as well as measurements of heat transfer on the endwall surfaces.

1.4 Outline of Dissertation

The following dissertation has been prepared in manuscript format, whereby several peer reviewed journal papers serve as individual chapters to report the overall results of the dissertation. Chapter 3\* presents cooling performance results on the endwalls of a planar and axisymmetric contoured passage with coolant injection from discrete film cooling holes and an upstream interface slot. Making use of the same passages as in Chapter 3, Chapter 4† presents heat transfer results on the passage endwalls with coolant injection from an upstream interface slot only.

Chapters 5-7 present results concerned with the effects of the interface slot position and orientation. Chapter 5‡ discusses the effects of three interface gap positions relative to the vane leading-edge on the cooling performance of the injected coolant. In addition, Chapter 5 presents results for two interface slot orientations. Based on the study in Chapter 5, it was concluded that the orientation of the interface slot is the largest contributing factor to the control of secondary flows and the performance of the injected coolant. The following studies were thus focused on investigating two additional interface slot orientations in respect to what was presented in Chapter 5. Chapter 6§ presents both cooling performance and heat transfer results related to four interface slot orientations.

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\* Presented at 2010 IGTI Turbo Expo and published in the *Journal of Turbomachinery*

† Presented at 2010 IGTI Turbo Expo and published in the *Journal of Turbomachinery*

‡ Presented at 2011 IGTI Turbo Expo and accepted for publication in the *Journal of Turbomachinery*

§ Submitted for review to the 2012 IGTI Turbo Expo
orientations. In addition, Chapter 6 presents time-averaged flowfield results in the stagnation plane of the vane leading-edge which capture the formation and in some cases the mitigation of the horseshoe vortex. Chapter 7* also presents cooling performance and heat transfer results related to the four different interface slot orientations, but the main focus of the chapter is on the time-resolved dynamics of the stagnation plane flowfields.

An overview of the major results and conclusions of the dissertation are presented at the end of the document in Chapter 8 in addition to recommendations for future work. Also included are appendices comparing predicted and experimentally measured adiabatic effectiveness levels, the derivation of predicted secondary flows, and the experimental measurement uncertainty analysis.

* Submitted for review to the *International Journal of Heat and Mass Transfer*
Chapter 2

Review of Relevant Literature

The following sections discuss past work related to axisymmetric contouring on a turbine vane endwall as well as coolant injection through the endwall from a two-dimensional interface slot upstream of the vane cascade. The uniqueness of the current study is discussed following the review of the relevant literature.

2.1 Effects of Axisymmetric Contouring and Coolant Injection on Secondary Flows

Many variations of endwall contouring exist. Endwall contouring can be applied to either one or both endwalls. Contouring can be upstream of the vane passage, upstream and within the vane passage, or only within the vane passage. Endwall contouring can be classified as either two- or three-dimensional. Two-dimensional, also referred to as axisymmetric endwall contouring, is a modification to the endwall surface along the axial direction only. This type of contouring typically involves a modification to the entire endwall in the form of a linear or non-linear surface. A three-dimensional contour, otherwise known as a non-axisymmetric contour, makes use of localized hills and valleys to reduce the radial pressure gradient at the endwall, leaving the freestream flow unchanged. While three-dimensional contouring has shown to be beneficial for aerodynamic losses (Knezevici et al. [12], Praisner et al. [13], Gustafson et al. ([14]-[15]), Okita and Nakamata [16], Lynch [17]), it has generally only been applied to turbine blade endwalls. The following presents findings related to axisymmetric endwall contouring where all the contouring is located on one endwall of a vane passage as in this dissertation.
A detailed investigation of the flowfield in a three vane, linear cascade with axisymmetric contouring of one endwall in the form of a non-linear slope as shown in Figure 2.1 was performed by Dossena et al. [18]. The passage had a height contraction of 30%, beginning at the passage inlet and ending at the exit. A comparison between the co-

![Figure 2.1](image-url)

**Figure 2.1.** A schematic of the axisymmetric contoured passage and total pressure loss contours in the exit plane of a planar and axisymmetric contoured passage (Dossena et al. [18]).

-noutred and flat endwalls showed that the secondary vortex structures were strongly affected by the endwall contour. On the flat endwall of the contoured passage, the secondary structures in the exit plane of the passage were similar to that of a planar passage. On the contoured endwall, the area contraction inhibited the formation of the
passage vortex and its growth toward midspan. Consequently, the contoured passage produced lower total loss levels than the planar passage.

Introducing coolant injection, Barigozzi et al. [19] investigated the effects of endwall film cooling on the aerodynamic performance of a seven vane, linear cascade with axisymmetric endwall contouring. The contoured passage was similar to that used by Dossena et al. [18], with a height contraction of 30% which started at the passage inlet and ended at the exit. Contoured endwall results, with and without film cooling from discrete holes, were compared to previously obtained results for a planar passage as shown in Figure 2.2. Flowfield measurements in the exit plane of the contoured passage showed a non-symmetric energy loss distribution, with a reduced wake compared to the planar case. Near the flat side of the contoured passage, the energy loss resembled the planar distributions, clearly showing the loss core of the passage and corner vortex. Near the contoured side, however, the passage vortex appeared to be suppressed at the endwall and joined to the corner vortex. Barigozzi et al. [19] reported that the overall loss associated with the contoured passage was 20% lower than the overall loss of the planar passage. For both the planar and contoured passages coolant injection modified secondary flows, elongating the structures and pushing them closer to the endwall. Note that for the planar passage film cooling was present on both endwalls while for the contoured passage film cooling was only present on the contoured endwall. With film cooling, losses in the contoured passage were always smaller than those in the planar passage.
Figure 2.2. A schematic of the axisymmetric contoured passage with film cooling and local energy loss contours in the exit plane of a planar and axisymmetric passage (Barigozzi et al. [19]).

Burd and Simon ([20]-[21]) performed an experimental investigation in a three vane, linear cascade with one flat and one axisymmetric contoured endwall which reduced the cascade span by 25% from inlet to exit through a 45° slope. Unlike previous studies, however, the endwall contouring began downstream of the vane leading-edge as shown in Figure 2.3. With contouring of one endwall, secondary flow structures in the passage
exit plane were shown to be non-symmetric and differed considerably in their size and strengths. Similar to that found by Barigozzi et al. [19], the passage vortex near the flat endwall was large in size but fairly weak in strength while the contoured endwall counterpart was more concentrated in size and stronger. Boundary layer thinning due to streamwise acceleration imposed by the contoured endwall was believed to be responsible for these differences. With the addition of low coolant injection rates from an upstream slot angled at 45° to the endwall the secondary flow structures were left unchanged. At increased injection rates, however, the flow structures near the contoured endwall were progressively suppressed closer to the endwall similar to that shown by Barigozzi et al. [19].

![Schematic of axisymmetric contoured passage with coolant injection from upstream slot (Burd and Simon [20]).](image)
2.2 Effects of Axisymmetric Contouring on Interface Slot Cooling

In a computational study, Lin et al. [22] investigated the three-dimensional flow in a nozzle guide vane passage with leakage flow from an upstream slot. The vane passage had one flat and one axisymmetric contoured endwall. Both endwalls incorporated leakage flow from an upstream slot. For the contoured endwall, two configurations of the same 45° contour were investigated as shown in Figure 2.4. In one configuration, the contraction was upstream of the vane passage. In the other, the contraction started upstream and continued through the vane passage. With coolant injection, secondary flows were found to be reduced at all endwalls for both contoured configurations. Without coolant injection, however, secondary flows were only reduced along the conto-

Figure 2.4. Schematic of axisymmetric contoured passage and contours of adiabatic effectiveness on both the flat and contoured endwalls (Lin et al. [22]).
ured endwall in which the contraction started upstream of the vane and continued through the passage. The injection coolant was significantly more effective in providing coolant coverage to both endwalls, flat and contoured, in the configuration where the contraction started upstream of the vane and continued through the passage. Coolant coverage was also improved on the contoured endwall in respect to the flat endwall for a given contoured passage configuration.

Expanding on the computational work performed by Lin et al. [22] for a nozzle guide vane passage with axisymmetric contouring of one endwall, Lin and Shih [23] considered the heat transfer on both the flat and contoured endwalls. Comparing the contoured and flat endwalls between the two configurations shown previously in Figure 2.4, heat transfer levels were shown to be lower for both the flat and contoured endwalls within the configuration where the contouring continued through the vane passage as shown in Figure 2.5. In addition, heat transfer levels were found to be lower on the contoured end-

![Figure 2.5. Contours of heat transfer coefficients on both the flat and contoured endwalls of two different axisymmetric contoured passages (Lin and Shih [23]).](image)
-wall than on the corresponding flat endwall for the configuration where the contouring continued through the vane passage.

Piggush and Simon ([24]-[25]) performed an extensive study to quantify the effects of coolant injection and misalignment of an upstream interface slot on the heat transfer of an axisymmetric contoured endwall. The endwall design also incorporated a slashface gap within the passage of the nozzle guide vane. Heat transfer and adiabatic effectiveness measurements were only performed, however, on the contoured endwall. With no baseline for comparison it is difficult to establish the effects that endwall contouring had on the heat transfer.

2.3 Effects of Interface Slot Injection Rates on Endwall Cooling

To investigate the independent effects of injection slot mass flow and momentum flux, Cardwell et al. [26] varied the width of a 45° interface slot located upstream of a planar vane passage. The slot width was varied from a nominal width to a half and double width configuration with injection mass flows ranging from approximately 0.4% to 1% of the total passage mass flow. Note that the vane passage used by Cardwell et al. [26] also included coolant injection from a mid-passage gap and discrete film cooling holes. Cardwell et al. [26] found that effectiveness levels were dependent on the injection slot mass flow as shown in Figure 2.6. Increasing the injection slot mass flow while maintaining a constant momentum flux was shown to improve effectiveness levels. In addition, Cardwell et al. [26] found that the coolant coverage pattern was a function of injection momentum flux as indicated in Figure 2.7. Increasing the injection slot
momentum flux while maintaining a constant mass flow improved the endwall coolant coverage.

![Figure 2.6. Contours of adiabatic effectiveness on the endwalls of a vane passage for a fixed injection slot momentum flux and increasing mass flow (Cardwell et al. [26]).](image)

<table>
<thead>
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<th>Injection slot mass flux ratio, MFR</th>
<th>0.38%</th>
<th>0.75%</th>
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<td>Injection slot momentum flux, I</td>
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<td>0.08</td>
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</tbody>
</table>

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

![Figure 2.7. Contours of adiabatic effectiveness on the endwall of a vane passage for a fixed injection slot mass flow and increasing momentum flux (Cardwell et al. [26]).](image)

<table>
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<tr>
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<td>0.1</td>
<td>0.39</td>
</tr>
</tbody>
</table>

\[\eta = \frac{T_\infty - T_{aw}}{T_\infty - T_c}\]
Using the same vane cascade as Cardwell et al. [26], Lynch and Thole [27] also investigated reducing the slot width while maintaining the injection slot mass flow, essentially increasing the momentum flux. Similar to that found by Cardwell et al. [26], Lynch and Thole [27] showed that increased momentum flux resulted in larger coolant coverage areas while increased mass flow results in higher effectiveness values as shown in Figure 2.8. Lynch and Thole [27] also measured heat transfer on the vane endwall and

<table>
<thead>
<tr>
<th>Injection slot mass flux ratio, MFR</th>
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<th>1.0%</th>
</tr>
</thead>
<tbody>
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<td>Injection slot momentum flux, I</td>
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<td>0.13</td>
<td>0.13</td>
<td>0.50</td>
</tr>
</tbody>
</table>

\[ \eta = \frac{T_{\infty} - T_{w}}{T_{\infty} - T_{c}} \]

**Figure 2.8.** Contours of adiabatic effectiveness on the endwall of a vane passage for different combinations of injection slot mass flow and momentum flux (Lynch and Thole [27]).

found that increasing the injection slot mass flow while keeping the momentum flux constant resulted in an increase in endwall heat transfer as shown in Figure 2.9. Increasing the injection slot momentum flux while keeping the mass flow constant, however, resulted in a slight reduction in heat transfer levels. There is no study available that investigates the aerodynamic losses associated with injecting coolant solely from the interface slot. As discussed in Section 2.1, however, Barigozzi et al. [19] showed that
coolant injection from discrete holes reduced secondary flows and subsequently lowered overall vane passage aerodynamic losses with respect to a passage with no cooling. The potential of slot coolant to alter passage aerodynamic losses is thus dependent on the resulting secondary flows.

<table>
<thead>
<tr>
<th>Injection slot mass flux ratio, MFR</th>
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<td>Injection slot momentum flux, I</td>
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**Figure 2.9.** Contours of heat transfer on the endwall of a vane passage for different combinations of injection slot mass flow and momentum flux (Lynch and Thole [27]).

**2.4 Effects of Interface Slot Position and Orientation on Endwall Cooling**

There is a wide breadth of available data on endwall cooling performance for differing interface slot geometries. Only a few studies, however, use consistent flow conditions and vane cascade geometries from which effects of varying the slot geometry and location can be properly analyzed. Axial location has been shown to be an important parameter in the cooling performance of an upstream interface slot. Kost and Nicklas [28] and Nicklas [29] made aerodynamic and thermodynamic measurements within a linear vane cascade with a transonic flowfield utilizing leakage flow from a 45° slot
located 0.2 axial chords (C_\text{ax}) upstream of the vane leading-edge. Film cooling holes were also distributed within the vane passage. Kost and Nicklas [28] showed that the horseshoe vortex was strengthened by the injection of coolant from the interface slot at a mass flow of 1.3%. This was attributed to the location of the slot, being positioned in the region of the separation saddle point. In a similar experiment, Kost and Mullaert [30] investigated the effect of moving the slot further upstream to 0.3C_\text{ax} upstream of the vane cascade. Results showed that the intensification of the horseshoe vortex could be avoided by moving the slot further upstream. In addition, it was found that the injected coolant stayed closer to the endwall and provided better cooling even with less than half the amount of mass flow as in the case where the slot was at 0.2C_\text{ax}.

Using the same vane cascade, inlet conditions, and injection flow rates, Lynch and Thole [27] and Knost and Thole [31] performed adiabatic effectiveness experiments using injection coolant from identical 45° slots. Experiments by Lynch and Thole [27] and Knost and Thole [30] differed in the upstream location of the interface gap corresponding to 0.96C_\text{ax} and 0.38C_\text{ax} upstream of the vane cascade respectively. The coverage area of the leakage coolant was similar between the two locations as shown in Figure 2.10. The effectiveness levels within the passage, however, were lower for the slot placed further upstream as the coolant had more distance to mix with the hot mainstream flow.

Utilizing an axisymmetric contoured passage, an experimental study by Rehder and Dannhauer [32] investigated the effects of tangential and perpendicular coolant injection from an interface slot upstream of a linear, three blade cascade. Measurements of the velocity flowfields indicated that injecting coolant tangential to the endwall increased the
near wall velocity in the approaching boundary layer leading to a significant reduction in strength of the horseshoe vortex. At higher injection mass flows, the horseshoe vortex was shown to be non-existent. In contrast, injecting coolant perpendicular to the endwall strengthened the horseshoe vortex and increased the size of the passage vortex as the injected coolant promoted the separation of the incoming boundary layer. Secondary losses were shown to increase with perpendicular injection and decrease with tangential injection relative to the zero leakage case. In accordance with the velocity measurements, tangential leakage injection reduced the influence of the secondary flows on the endwall heat transfer. Though similar in peak value, high heat transfer values were shown to penetrate further into the blade passage when coolant was injected perpendicular to the surface as compared to tangential injection as shown in Figure 2.11.
An experimental study by Yang et al. [33] investigated the cooling performance of slot flow injecting tangential to a surface over a range of inclination angles. Yang et al. [33] showed that when the film cooled surface was convergent at an angle of 10° or less, cooling effectiveness was reduced compared to when the film cooled surface was horizontal. Yang et al. [33] suggested that the mainstream flow may impinge the film cooled surface and destroy the film jet structure, reducing cooling effectiveness. When the convergent angle was greater than 10° however, an increase in film cooling performance was observed. Re-laminarization of the turbulent flow in the mixing zone from increased acceleration was attributed to the increase in cooling effectiveness.

![Figure 2.11](image)

**Figure 2.11.** Contours of heat transfer on the endwall of a blade passage without injection and with injection from two different interface slot orientations (Rehder and Dannhauer [32]).
2.5 Uniqueness of Current Study

To date, most studies in the available literature have focused on the aerodynamic effects of axisymmetric endwall contouring associated with passage losses. Few studies have investigated the subsequent effects of axisymmetric endwall contouring on the endwall cooling performance and heat transfer as a result of coolant injection. In addition, none of the available studies have compared a contoured passage to a planar passage to contrast the physics and the subsequent effects on secondary flows and endwall cooling. The contoured passage introduces a unique situation where both the flat and sloped endwalls are subject to increased acceleration relative to the planar case. In addition, the contoured endwall is under the influence of streamline curvature. The study reported in this dissertation is unique in that in addition to providing detailed measurements of adiabatic effectiveness and heat transfer on both endwalls of an axisymmetric contoured passage, baseline measurements on the flat endwall of a planar passage are also presented.

In the contoured passage study, coolant injection from the upstream interface slot in the contoured endwall was shown to provide a greater coverage of the endwall in comparison to the flat endwalls. The improvement in coolant coverage was a result of the reduced slot orientation relative to the contoured endwall as the interface slot was maintained perpendicular to the inlet flow. Slightly reducing the slot orientation resulted in a large change in the endwall cooling. To further explore this observation the effects of injection rate, position, and orientation of the interface slot were also considered. Only a few studies, however, have investigated changing slot parameters such as position and orientation to observe the resulting effect on endwall cooling. No studies exist that
investigate the time-resolved dynamics of the vane leading-edge flowfield with coolant injection from an upstream interface slot. The dynamics of the vane leading-edge flowfield when no injection slot is present are driven by several parameters such as the vane leading-edge diameter, velocity boundary layer, and Reynolds number. When coolant injection from an upstream slot is present additional parameters such as the slot injection rate, position, and orientation become important to determining the dynamics of the leading-edge flowfield. The study reported in this dissertation is unique in that it seeks to understand the effects of injection rate, position, and orientation of an upstream interface slot on the cooling characteristics of a vane endwall. This study is the first to report the effects of interface slot orientation on the net heat flux reduction experienced by a first stage vane endwall with coolant injection from an upstream interface slot. This study is also the first to present time-resolved measurements of the secondary flows within the leading-edge plane of a vane with coolant injection from an upstream interface slot.

In summation, the main contribution of the work presented in this dissertation is a detailed explanation of the effects of axisymmetric endwall contouring on the cooling of a vane endwall in the presence of coolant injection from an upstream interface slot and discrete holes. In addition, the effects interface slot injection rate, position, and orientation are presented as related to endwall cooling and secondary flows.
Chapter 3

Effects of an Axisymmetric Contoured Endwall on a Nozzle Guide Vane: Adiabatic Effectiveness Measurements

3.1 Introduction

From a simple thermodynamic analysis of a gas turbine it is apparent that raising the gas temperature at the exit of the combustor is directly beneficial to the thermal efficiency. While this is a straightforward approach for increasing engine efficiency it does not come without consequence. Increased temperatures lead to durability issues within the turbine section. The first turbine component to realize these durability concerns from increased gas temperatures are the first stage nozzle guide vanes. To maintain component lifetimes, film cooling is employed both on the vane surface and endwall. Leakage flows from component interface gaps can also be used to cool.

Cooling of the endwall region is of particular interest and difficulty due to the presence of secondary flows. Secondary flows originate in the boundary layer, forming a horseshoe vortex at the stagnation of the vane endwall junction. Combining with cross-stream endwall flows, the pressure side vortex leg grows as it becomes the passage vortex. Consequently, any coolant introduced upstream of the passage vortex will be swept into the freestream and be ineffective in cooling the endwall. To mitigate secondary flows, contouring of the endwall surface is sometimes employed.
Modification of the mainstream and near wall flow fields through endwall contouring, however, can alter the performance of film cooling.

This paper discusses the effect of axisymmetric endwall contouring on the cooling of a low-pressure, turbine vane endwall. The effect of varying the mass flow rate through the upstream leakage slot and film cooling holes is also considered. Note that this paper is part of a larger study that also investigates the effect of axisymmetric endwall contouring on the heat transfer characteristics of the guide vane endwall. A second paper, Thrift et al. [35], presents heat transfer results on the endwalls of the planar and contoured passages over a range of leakage flow rates.

3.2 Experimental Methods and Benchmarking

All measurements were obtained in a test section containing three, scaled-up nozzle guide vanes. Flow was supplied to the test section by a low speed, closed loop wind tunnel, depicted in Figure 3.1. Driving the flow through the wind tunnel was a variable speed, 50 hp fan. Downstream of the fan, the flow was turned by a 90° elbow before passing through the primary, finned-tube heat exchanger used to cool the main flow. After being turned by another 90° elbow, the air was split into three flow paths. The flow

Figure 3.1. Depiction of the low speed, closed loop wind tunnel.
passing through the center passage simulated heated core flow. The flow which passed through the upper and lower passages was used as coolant for both leakage and film cooling flows. Air in each of the two outer flow paths traveled through secondary finned-tube heat exchangers, where additional cooling could take place before passing into respective plenums. Coolant air was drawn from the upper plenum only and into the appropriate local coolant plenums on the attached test section using a 2 hp blower.

After the flow split, the core flow passed through a heater bank, a series of screens used for flow straightening, and then into a contracted straight flow section with a rounded inlet. At the exit of this section was the experimental test section where all measurements were performed. The test section incorporated a 90° bend to assist in the turning of the flow through the vane cascade. Air exiting the test section was turned by a final 90° elbow before encountering the fan and completing the closed-loop.

The vane test section was a two-dimensional linear vane cascade. The test section contained two full nozzle guide vanes and a third partial vane connected to a flexible wall to maintain the desired pressure distribution along the center vane. The vanes were scaled up by a factor of 2.4 from actual engine size to achieve high measurement resolution. Three different configurations of the test section were used for testing: a planar passage, a contoured passage with the contoured endwall as the ceiling, and a contoured passage with the contoured endwall as the flooring. Figure 3.2 presents a schematic illustrating the three endwall configurations. In the case of the contoured passage, the relocation of the contoured endwall as either the ceiling or floor was necessary to study both the flat and contoured endwalls as measurements could only be performed on the floor due to op-
Figure 3.2. Schematic of the (a) planar passage, (b) contoured passage with contour on ceiling, and (c) contoured passage with contour on floor.

For the contoured passage the area contraction began $1.25C_{ax}$ upstream of the cascade inlet with a linear slope of $16^\circ$. The contraction continued through the passage giving an area reduction of 83% from inlet to outlet.

Table 3.1 provides the vane geometry and flow conditions for both the planar and contoured passages. Note that the exit Reynolds number was matched between the planar and contoured passages. The exit Reynolds number was matched to simulate actual engine conditions where the velocity entering subsequent blade stage, downstream of the
vanes, must remain the same even with the addition of the contour over the vane stage. Matching exit Reynolds numbers required a lower inlet velocity for the contoured passage due to the additional spanwise area contraction.

The boundary layer entering the cascade was measured at a location $4.25C_{ax}$ upstream of the vane stagnation. Table 3.2 lists the turbulent inlet boundary layer parameters, which were maintained throughout this study. The measured turbulence intensity was lower than that typically found in an engine at 1.0%. The effect of freestream turbulence was not considered in this study as past studies by the authors have evaluated this effect [36].

Table 3.1. Vane Geometry and Flow Conditions

<table>
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<tr>
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<th>Planar Passage</th>
<th>Contoured Passage</th>
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<tr>
<td>Scaling factor</td>
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</tr>
<tr>
<td>Scaled vane chord (C)</td>
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<td>Axial chord/chord ($C_{ax}/C$)</td>
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<tr>
<td>Pitch/chord (P/C)</td>
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<th></th>
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<tbody>
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<td>Inlet span/chord ($S_{in}/C$)</td>
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<td>Velocity ratio ($U_{in}/U_{exit}$)</td>
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<td>5.94</td>
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<td>Inlet Reynolds number ($Re_{in}$)</td>
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<td>$1.7 \times 10^5$</td>
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<tr>
<td>Exit Reynolds number ($Re_{exit}$)</td>
<td>$1.0 \times 10^6$</td>
<td>$1.0 \times 10^6$</td>
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</table>
Table 3.2. Inlet Boundary Layer Characteristics

<table>
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<td>Boundary layer thickness/span ((\delta/S))</td>
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<tr>
<td>Displacement thickness/span ((\delta^*/S))</td>
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<td>Momentum thickness Reynolds number ((Re_\theta))</td>
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</table>

<table>
<thead>
<tr>
<th>Contoured Passage</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Boundary layer thickness/span ((\delta/S))</td>
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</tr>
<tr>
<td>Displacement thickness/span ((\delta^*/S))</td>
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<td>Momentum thickness/span ((\theta/S))</td>
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<tr>
<td>Shape factor ((\delta^*/\theta))</td>
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</tr>
<tr>
<td>Momentum thickness Reynolds number ((Re_\theta))</td>
<td>2800</td>
</tr>
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</table>

To simulate the leakage interface between a combustor and turbine, a two-dimensional 90° slot was placed 0.17\(C_{ax}\) upstream of the vane cascade in the bottom endwall. Figure 3.2 documents the location and dimensions of the upstream leakage slot for the three endwall configurations. Also shown in Figure 3.2 is the location of slot thermocouples, used to measure the local coolant temperature of the leakage flow. The thermocouples were placed one slot width below the external surface and within the upstream slot with the thermocouple bead being positioned at the center of the slot width. The thermocouples were arranged along the slot pitch in increments of 0.125\(P\), with a thermocouple located at each vane stagnation corresponding to \(y/P = 0, -1\) and 1.

In addition to coolant flow from an upstream slot, the vane endwall also incorporated film cooling. The film cooling pattern made use of laid back, fan shaped holes inclined at 26° to the surface with a laid back and half-diffusion angle of 13°. As illustrated in Figure 3.3 the film cooling holes were distributed along the suction side leading edge and press-
3.2.1 Coolant Flow Settings

For every test condition, the inlet velocity distribution and the pressure distribution along the vane midspan were measured. The inlet velocity distribution was measured approximately 0.8C upstream of the cascade inlet. Inlet velocities measured across the width of the cascade varied less than 5%. Figure 3.4 compares the measured and predicted pressure distributions around the mid-span of the center vane for both the planar and contoured passages. Note that the mid-span plane was defined in the non-converged, inlet passage as indicated in Figure 3.2.

Figure 3.4 indicates that the inviscid flow field around the center vane was matched to the predicted curves. The predictions were obtained from a computational study performed for incompressible, viscous, low-speed conditions using FLUENT [37]. The computational domain was such that the vane was divided at the stagnation point and the trailing edge with a single flow passage being modeled. Periodic boundary conditions
were placed along the pitchwise boundaries of the computational domains. The predicted pressure distributions were representative of a vane within a continuous linear cascade. A detailed description of the computational method used during this study is provided by Thrift et al. [35].

For the leakage flow, the issuing coolant entering each passage was a prescribed percentage of a single passages mass flow rate. For the film coolant however, only the ri-
-ghtmost passage received a prescribed coolant percentage. For this study, coolant flow rates of 0.3%, 0.5%, and 0.7% were investigated from both the upstream leakage slot and film cooling holes for all three endwall configurations illustrated in Figure 3.2. In matching coolant mass flux ratio between the three endwall configurations the relative amount of coolant is the same for each passage. A total of 9 experiments were conducted for each endwall.

The mass flux issuing from the upstream leakage slot was measured using a laminar flow element, located within the supply pipe to the slot plenum. Unlike the leakage mass flow, the mass flow rate from the film cooling holes was determined using the local blowing ratios. Equation 3.1 provides the relationship for calculating the mass flow from a single film cooling hole. Note that the mass flow depends on the location of the film cooling hole as dictated by the local freestream velocity and blowing ratio. In addition, a discharge coefficient is required to obtain the true mass flow through the film cooling hole. Gritsch et al. [38] indicated that the discharge coefficient from a laidback, fan shaped hole was essentially constant at a value of 0.8 for low Mach number flows over a large range of pressure ratios. Given the conditions of the current study a discharge coefficient of 0.8 was assumed for all film cooling holes.

\[
\dot{m}_{FC} = C_D A_h \rho_\infty U_{\infty,1} M_1
\]  

(3.1)
To calculate the local blowing ratio, $M_l$ was recast in terms of a pressure difference between the inlet stagnation pressure and the film cooling plenum as shown below in Equation 3.2 [39]. Calculation of the local blowing ratio required the local freestream velocity and local pressure coefficient. The local inviscid velocity was obtained from the CFD results at the mid-span plane at a given film cooling hole location in the x-y plane. The local pressure coefficient was also obtained from the CFD predictions along the endwall at a given film cooling location.

Blowing ratios were large near the vane stagnation due to low local freestream velocities as blowing ratio is inversely proportional to freestream velocity as shown in the nomenclature. This is illustrated in Figure 3.5 which plots contours of local blowing ratio for the flat endwall of the planar passage over a range of film cooling (FC) flow rates. In addition, Figure 3.5 plots the local blowing ratios for the flat and contoured endwalls of the contoured passage at an intermediate film cooling flow rate. Near the pressure side leading edge and progressing toward the passage throat, blowing ratios gradually decreased as the freestream velocity increased. Figure 3.5 shows that as the film cooling flow rate was increased the local blowing ratio for each film cooling hole increased as well. The endwalls of the contoured passage were subject to slightly smaller blowing ratios, however, the distribution of the blowing ratios with increasing film cooling was similar to that within the planar passage. Mass flow rates were small in the regions of

\[
M_l = \sqrt{\frac{\rho_c}{\rho_\infty} \left( \frac{P_{\text{plenum}} - P_{o,\text{in}}}{0.5\rho_\infty U_{\infty,\text{in}}^2 (1 - C_{p_1})} + 1 \right)}
\] (3.2)
stagnation where blowing ratios were relatively high compared to the film cooling holes within the vane passage.

**Figure 3.5.** Contours of local blowing ratio for: the flat endwall of the planar passage (a) 0.3% FC, (b) 0.5% FC, and (c) 0.7% FC; the flat endwall of the contoured passage (d) 0.5% FC; and the contoured endwall of the contoured passage (e) 0.5% FC.
3.2.2 Adiabatic Effectiveness Measurements

Adiabatic wall temperatures were obtained from Infrared (IR) measurements on the endwall floors of the three configurations discussed previously. Experiments were performed with a temperature difference between the freestream and coolant flow of approximately 25°C to reduce measurement uncertainty. To minimize conduction error, the measurement endwall was made of a 2.54cm thick plate of low-density closed-cell polyurethane foam, which has a low thermal conductivity (0.0287 W/m-K). In addition, the foam endwall was painted flat black to enable good resolution of the surface temperatures with the IR camera.

An inframetrics P20 IR camera was used to acquire the spatially-resolved adiabatic temperatures on the bottom endwall. To capture the entire endwall surface, measurements were taken at 14 different viewing locations, which is a method similar to that used by previous researchers [40-43]. Five images were taken at each location and averaged to produce the final image. From a camera distance of 55cm, each picture covered an area that was 24cm by 18cm with the viewing area being divided into 320 by 240 pixel locations resulting in a spatial integration of 0.75 mm (0.39 FC hole diameters).

Surface temperatures captured by the IR camera were post calibrated using directly measured temperatures on the endwall by thermocouples placed along the endwall. Each image captured by the IR camera enclosed at least two calibration thermocouples which were shared with neighboring viewing locations. Images were post-calibrated by determining the emissivity and background temperature of the image through matching of the image temperatures with the acquired thermocouple measurements. In general, the thermocouple and calibrated images agreed within 0.5°C. Typical emissivity and
background temperature values were 0.96 and 55°C respectively. The background temperature corresponded to the freestream temperature of the flow as the surrounding environment was allowed to reach steady state. After calibration of the images, the data was exported to an in-house MATLAB program which assembled the individual images into a single map of the entire endwall.

A one-dimensional conduction correction as described by Ethridge et al. [44] was applied to all adiabatic effectiveness measurements. The correction involved measuring the endwall surface effectiveness with no coolant flow. A correction value of $\eta_0 = 0.1$ was measured within the vane passage. At the passage throat the correction value was found to be $\eta_0 = 0.05$. Upstream of the leakage slot a correction value of $\eta_0 = 0.15$ was measured. The endwall upstream of the leakage slot was made of medium density fiberboard with a thermal conductivity (0.13 W/m-K) higher than that of the foam making up the cooled surface, resulting in larger conduction losses.

### 3.2.3 Uncertainty Analysis

An uncertainty analysis was performed on the measurements of adiabatic effectiveness using the partial derivative method described by Moffat [45]. A precision uncertainty for the adiabatic surface temperatures was determined by taking the standard deviation of six measurement sets of IR camera images. Each image set consisted of five images as mentioned previously. Based on a 95% confidence interval the precision uncertainty of the measurements was $\pm 0.2^\circ$C. The bias uncertainty for an image was taken as the root-sum-square of the thermocouple bias uncertainty ($\pm 0.2^\circ$C) and the average deviation of the calibrated images from the thermocouples ($\pm 0.5^\circ$C). In this way,
a bias uncertainty of ±0.54°C was determined. Combining the bias and precision uncertainties, a total uncertainty of ±0.58°C was estimated for the IR surface temperature measurements. The uncertainty in adiabatic effectiveness was then found based on the partial derivative of $\eta$ with respect to each temperature in the definition and the total uncertainty in the measurements. An uncertainty of $\partial\eta = ±0.02$ was calculated over the range $\eta = 0.03$ to 0.7.

3.3 Results and Discussion

The results from the experiments will be presented as contours and line plots of adiabatic effectiveness on the three following endwalls: the flat endwall of the planar passage; the flat endwall of the contoured passage; and the contoured endwall of the contoured passage. A comparison will be made among the three endwalls concerning the effect of leakage flow on endwall cooling effectiveness. The performance of the film cooling pattern will also be investigated among all three endwalls.

3.3.1 Effects of Endwall Contouring on Leakage Flow

Adiabatic effectiveness experiments were performed over a range of leakage and film cooling flow rates. To isolate the effect of endwall contouring on the upstream leakage flow only those experiments with the lowest film cooling flow rate are considered. In this way the influence of the film coolant on the upstream leakage flow is minimized. Figure 3.6 compares contours of adiabatic effectiveness among the three endwalls at various leakage flow rates with a film coolant flow rate of FC = 0.3%.
For each endwall in this study the ejection of leakage flow from the upstream slot is concentrated near the suction side leading edge. This can be attributed to the relatively low static pressure on the endwall near the suction side. Lynch and Thole [27], in comparison to a study performed by Knost and Thole [31], showed that moving the leakage slot further upstream away from the influence of the vane, led to more uniform coolant ejection. Another leakage flow pattern that is common among all conditions in Figure 3.6 is the sweeping of the leakage flow from the pressure to the suction side. This pattern is caused by the endwall cross-flow and passage vortex, entraining coolant and sweeping it to the suction side of the vane.

Figure 3.6 shows that increasing the leakage flow from the upstream slot results in increased coolant coverage for each endwall. As the leakage flow rate increases the pressure difference between the slot plenum and the local static pressure at the exit of the slot increases. This allows the coolant flow to overcome the relatively high static pressure near the vane leading edge.

As mentioned previously the surface upstream of the leakage slot was made of a material with a thermal conductivity approximately 4.5 times that of the polyurethane foam on the downstream surface. As observed in Figure 3.6, this led to effectiveness values around 0.1 directly upstream of the leakage slot where the driving temperature difference was high.

Heat transfer measurements by Thrift et al. [35] on the passage endwalls highlight the importance of the path taken by the leakage flow. Results indicate that heat transfer is largest near the suction side of the vane leading edge. In addition, the heat transfer in this region is shown to increase with increasing leakage flow rate.
Figure 3.6. Comparison of adiabatic effectiveness contours between the three endwalls for (a) 0.3% FC and 0.3% leakage flow, (b) 0.3% FC and 0.5% leakage flow, and (c) 0.3% FC and 0.7% leakage flow.
A slight reduction in coolant coverage from the upstream leakage flow is observed for the flat endwall of the contoured passage relative to the flat endwall of the planar passage. Although each slot plenum receives the same percentage of coolant flow, the endwall static pressure in the case of the flat endwall of the contoured passage is such that the slot plenum pressure is significantly lower. A reduction in slot plenum pressure for the flat endwall of the contoured passage results in slightly less coolant coverage at each leakage flow rate relative to the flat endwall of the planar passage as shown in Figure 3.6.

The contoured endwall of the contoured passage shows the largest coolant coverage from the upstream slot at each leakage flow rate in comparison to the flat endwalls in both the planar and contoured passages. This can be attributed to the reduced coolant ejection angle for the upstream slot in the contoured endwall. Figure 3.2 shows that the upstream leakage slot in the contoured endwall is ejecting coolant at an angle of 74° relative to the contoured endwall. Ejecting coolant at a reduced angle reduces the penetration depth of the leakage flow into the mainstream, resulting in better coverage of the endwall surface. Predictions of the three-dimensional flow field were performed with all three endwalls by Thrift et al. [35] to investigate the horseshoe vortex. To illustrate the difference in penetration depth between the leakage flow issuing from the flat endwall of the planar passage and the contoured endwall of the contoured passage, Figure 3.7 provides streamtraces along the vane stagnation plane obtained from the CFD simulations at a leakage flow of 0.75%. As shown in Figure 3.7, the injection of coolant results in the separation of the approaching flow and the subsequent formation of a horseshoe vortex as the stagnating boundary layer flow is driven toward the endwall and back upstream. The
coolant penetrates less into the freestream for the contoured endwall as a result of the reduced spanwise injection component, causing a smaller separation region thus resulting in a smaller horseshoe vortex. This finding is similar to those computational results provided by Y.-L. Lin et al. [22], who also showed increased coolant coverage from a 90° upstream slot on the contoured endwall relative to the flat endwall. As in this study, Y.-L Lin et al. [22] kept the slot orientation at 90° relative to the main flow direction, resulting in a reduced ejection angle for their sloped wall case of 45°. As was discussed previously, the relative close location of the upstream slot to the vane stagnation results

Figure 3.7. Streamtrace predictions along the vane stagnation plane (x-z) with 0.75% leakage flow for the (a) flat endwall of the planar passage and (b) contoured endwall of the contoured passage.
in leakage flow being ejected near the suction side of the vane leading edge. In addition, the high endwall static pressure at the vane stagnation can lead to ingestion of hot mainstream flow into the upstream slot. Hot mainstream flow is detrimental to the upstream leakage slot and the corresponding supply plenum.

To investigate the severity of this ingestion among the three endwalls, temperature measurements were recorded from the thermocouples measuring the slot flow temperature near the exit of the slot. The resulting measurements are presented in Figure 3.8 as non-dimensional temperatures across the cascade pitch for each endwall configura-

![Figure 3.8](image)

**Figure 3.8.** Comparison of non-dimensional leakage flow temperatures across the pitch of the upstream leakage slot at different leakage flow rates.
-tion at a leakage flow of 0.3% and 0.7%. Figure 3.8 shows that the contoured endwall has less flow ingestion than the flat endwalls of the planar and contoured passages. The flat endwall of the planar passage shows a slight improvement in mainstream ingestion relative to the other flat endwall at the lowest leakage flow rate. With an increase in leakage flow to 0.7% however, the flat endwall of the contoured passage remains the only endwall to experience ingestion into the upstream slot at stagnation. The consequence of flow ingestion can be observed in Figure 3.6c at the vane stagnation. Non-zero effectiveness levels downstream of the slot at vane stagnation for the flat endwall of the planar passage and the contoured endwall indicate that leakage flow was ejecting from the slot. For the flat endwall of the contoured passage however, no leakage flow is present at the vane stagnation.

As mentioned previously, predictions of the three-dimensional flow field were performed with all three endwalls by Thrift et al. [35]. Included in the simulations were predictions of adiabatic wall temperatures from leakage flow. Figure 3.9 presents the predicted adiabatic effectiveness levels on each endwall at a comparatively low and high leakage flow rate. The predicted adiabatic effectiveness levels display the same trends that were observed experimentally for the leakage flow. The majority of the coolant is ejecting near the suction side leading edge with the coolant sweeping from the pressure to the suction side of the passage. In addition, the predictions show that the coolant coverage and effectiveness levels increase with leakage mass flow rate. Also captured by the simulations is the improvement in coolant performance from the leakage flow for the contoured endwall in comparison to the flat endwalls of the planar and contoured passages.
3.3.2 Effects of Endwall Contouring on Film Cooling

As in the previous section, to isolate the effect of endwall contouring on the film cooling, only those experiments with the lowest leakage flow rate of 0.3% are considered. This is to minimize the effect that leakage flow has on the film cooling.

Figure 3.10 compares contours of adiabatic effectiveness among the three endwalls at various film cooling flow rates. Figure 3.10 shows that the coolant coverage from the
pressure side film cooling holes is shifted from within the passage toward the passage throat with an increase in film cooling flow. As the mainstream flow approaches the passage throat it accelerates, reducing the local blowing ratios. As the film cooling flow rate is increased however, the local blowing ratio for each film cooling hole increases as shown previously in Figure 3.5. Larger blowing ratios combined with increased film coolant ejection leads to an increase in coolant coverage near the passage throat.

Figure 3.10 shows that the film coolant coverage is poorest for the contoured endwall at each film cooling flow rate. To visualize these results in a more quantitative manner, values were extracted from the endwall data along the paths of inviscid streamlines obtained from a CFD prediction of the passage flowfield using FLUENT [37]. The inviscid streamlines originate at the midspan of the vane at two pitch locations corresponding to $y/P = 0.25$ and 0.5 as shown in Figure 3.10a. The streamlines pass directly through the endwall area influenced by the pressure side film cooling holes. Figures 3.11-3.13 present the adiabatic effectiveness levels along these inviscid streamlines for those contours presented in Figure 3.10.

Figure 3.11a and 3.11b show the effectiveness values for the contoured endwall are well below those for the flat endwall of the planar passage at the lowest film cooling flow rate along both the 0.25P and 0.5P streamlines respectively. While not nearly as severe, the effectiveness values for the flat endwall of the contoured passage are also slightly below those for the flat endwall of the planar passage. Figure 3.11b shows that the contoured endwall effectiveness levels are initially higher than those on the flat endwalls of the planar and contoured passages. Recall that the 0.5P streamline passes through the region influenced by the leakage flow. As discussed previously, the contoured endwall
Figure 3.10. Comparison of adiabatic effectiveness contours between the three endwalls for (a) 0.3% FC and 0.3% leakage flow, (b) 0.5% FC and 0.3% leakage flow, and (c) 0.7% FC and 0.3% leakage flow.
Figure 3.11. Comparison of adiabatic effectiveness levels between the three endwalls for 0.3% FC and 0.3% leakage flow, sampled along inviscid streamlines released from (a) 25% pitch and (b) 50% pitch.
Figure 3.12. Comparison of adiabatic effectiveness levels between the three endwalls for 0.5% FC and 0.3% leakage flow, sampled along inviscid streamlines released from (a) 25% pitch and (b) 50% pitch.
Figure 3.13. Comparison of adiabatic effectiveness levels between the three endwalls for 0.7% FC and 0.3% leakage flow, sampled along inviscid streamlines released from (a) 25% pitch and (b) 50% pitch.
shows improved cooling performance from leakage flow in comparison to the flat endwalls.

With an increase in film cooling to 0.5%, the disparity between the effectiveness levels on the contoured endwall and the flat endwall of the planar passage is reduced as shown in Figure 3.12a and 3.12b. Effectiveness values for the contoured endwall, however, are still below the flat endwall results at 0.5% FC, especially approaching the passage throat. In addition, the effectiveness values between the flat endwalls are very similar. Figure 3.12b highlights the initial improvement in effectiveness level from the leakage flow for the contoured endwall.

With a further increase in the film cooling flow to 0.7%, the difference in effectiveness levels between the contoured endwall and the flat endwalls continues to get smaller as shown in Figure 3.13a and 3.13b. The contoured endwall continues to exhibit poorer effectiveness values compared to the flat endwalls. Contrary to the results at the lowest film cooling flow rate, the effectiveness values on the flat endwall of the contoured passage are now slightly higher than those on the flat endwall of the planar passage.

The flat and contoured endwalls of the contoured passage are subject to different flow conditions. Relative to the planar passage, the flat endwall of the contoured passage is subject to the highest flow acceleration and the lowest static pressure. This is a result of the spanwise area contraction through endwall contouring. The flat endwall of the contoured passage is similar to a flat plate with a favorable pressure gradient along the streamwise direction. Schmidt and Bogard [46] and Teekaram et al. [47] investigated the effect of a favorable pressure gradient on the adiabatic effectiveness downstream of a
single row of film cooling holes. Both studies indicated negligible changes in the adiabatic effectiveness between the zero and non-zero pressure gradient cases in the ejection range where the jets were either fully attached or detached. For the range in which the jets were beginning to detach however, the studies indicated a significant increase in adiabatic effectiveness downstream of the ejection row. In the current study, the use of laid back, fan shaped film cooling holes makes it unlikely that the coolant flow is beginning to separate. As such, it is not unexpected that the flat endwall results of the contoured passage are similar to the flat endwall of the planar passage.

The same is not true for the contoured endwall, which in addition to the streamwise acceleration is also subject to a second flow condition, streamline curvature. As the flow approaches the contoured passage the streamlines converge, not only in the pitchwise direction, but along the span in response to the area contraction as well. Near the contoured endwall the streamlines must turn toward midspan as they approach and impinge on the endwall. The impingement of the mainstream flow occurs in the upstream regions where the flow is still mostly uniform. As shown in Figure 3.5, the blowing ratios for those holes near the pressure side leading edge are smaller on the contoured endwall than on the flat endwall of the contoured passage. As a result, the cooling performance of the pressure side film cooling holes on the contoured endwall is poor for those holes further upstream as shown in Figure 3.10. The cooling effectiveness increases for those holes further downstream, near the throat of the vane passage.
3.5 Conclusions

Measurements of adiabatic cooling effectiveness were presented on the endwalls of a planar and contoured passage, from leakage flow through a two-dimensional upstream slot and fully expanded film cooling holes. Three slot and film cooling flow rates were tested.

The influence of the vane and endwall secondary flows on upstream slot coolant was evident in the ejection pattern of the leakage flow. Leakage flow ejection was not uniform; instead most of the coolant was ejected near the suction side leading edge of the vane. After ejection, the coolant was abruptly swept from the pressure side of the passage toward the suction side. With slot and film-cooling flows present, there was a large region near the suction side leading edge that was overcooled. The presence of the film cooling holes in this region is redundant as the leakage flow, even at the lowest flow rate, adequately covers the surface. It seems that it would be beneficial to eliminate those film-cooling holes and use the coolant elsewhere, such as providing increased coolant mass flow near the vane leading edge.

Coolant coverage from the upstream leakage slot was largest for the contoured endwall at each flow rate tested. This was attributed to the reduced ejection angle of the leakage flow in the contoured endwall. In addition, ingestion of hot mainstream flow into the slot was reduced for the contoured endwall relative to the flat endwalls. The flat endwall of the contoured passage showed slightly reduced coolant coverage relative to the flat endwall of the planar passage. Consequently, ingestion of mainstream flow into the slot was observed for the flat endwall of the contoured passage even at the highest
leakage flow rate. An increase in coolant coverage was observed on each endwall for an increase in leakage flow.

Contrary to the leakage flow results, film cooling performance was poorest for the contoured endwall. The flat endwall of the contoured passage showed limited improvement over the flat endwall of the planar passage, but this was only at increased leakage flow rates. At the lowest leakage flow the flat endwall of the planar passage provided the best film cooling performance. The reduction in film coolant coverage and effectiveness for the contoured endwall was attributed to impingement and turning of the mainstream flow at the contoured surface, resulting in a reduction of the local blowing ratios near the pressure side leading edge.

Axisymmetric endwall contouring was shown to have a significant impact on cooling performance, both from an upstream slot and film cooling holes. The contoured passage introduced a unique situation where each endwall was subject to increased acceleration relative to the planar case. In addition, the contoured endwall was under the influence of a streamline curvature. Increased acceleration was shown to have a limited impact on the cooling performance as indicated by a comparison between the flat endwalls of the planar and contoured passages. However, streamline curvature and mainstream impingement were shown to be detrimental to film cooling performance on the contoured endwall.
Chapter 4

Effects of an Axisymmetric Contoured Endwall on a Nozzle Guide Vane: Convective Heat Transfer Measurements [35]

4.1 Introduction

The junctions between airfoils and their bounding endwalls serve as formation locations of highly three-dimensional flow structures. These structures are referred to as secondary flows. Secondary flows increase the total pressure loss through a turbine stage and ultimately lead to decreases in overall turbine efficiency. Secondary flows are rotational in nature thus they enhance mixing of the near wall flow with the hot mainstream gases. Mixing increases heat transfer to the turbine components and reduces the effectiveness of any supplied coolant. To improve the durability of the turbine, many researchers have investigated endwall contouring as a means to passively reduce or eliminate secondary flows.

Past researchers have shown that two- and three-dimensional modifications to the endwall surface can result in changes to the local and mainstream flow field that can be beneficial in mitigating secondary flows. Many studies have investigated these changes to the flow field through aerodynamic measurements, indicating lower loss levels for the contoured passage. Few studies, however, have considered the effect of endwall contouring on endwall heat transfer.

To help fill this void, the current paper discusses the effect of axisymmetric endwall contouring on the endwall heat transfer of a low-pressure, vane endwall. The effect of
varying the leakage mass flow rate through an upstream interface slot is also considered as this represents the combustor-turbine interface gap. Note that this paper is part of a larger study that also investigates the effect of endwall contouring on the cooling performance of a film cooled endwall with leakage from an interface slot. Thrift et al. [34] presents adiabatic effectiveness results on the endwalls of the planar and contoured passages over a range of leakage flow rates.

4.2 Experimental and Computational Methods

All experiments were performed in a closed loop, low speed wind tunnel, depicted in Figure 4.1 and previously described by Thrift et al. [34]. This facility included three channels: a primary channel, representing the main gas path; and two symmetric secondary channels, representing the coolant flow paths. In the primary channel, the flow passed through a heat exchanger then through a thermal and flow conditioning section.

Figure 4.1. Depiction of the low speed, closed loop wind tunnel.
containing a bank of heaters followed by a series of screens and flow straighteners. After a contracted section, the primary flow entered the experimental test section where all measurements were performed. Upstream of the thermal and flow conditioning section, a porous plate diverted flow into the secondary channels from the primary flow path. Air in each of the two outer flow paths traveled through secondary heat exchangers. Coolant air was drawn from the upper flow path and into the appropriate slot supply plenum on the attached test section using a 2 hp blower. Note that only the top secondary flow path was used in this study.

The vane test section contained two full nozzle guide vanes and a third partial vane connected to a flexible wall. The vane design was an extrusion of a two-dimensional midspan airfoil geometry scaled up by a factor of 2.4. As described in Thrift et al. [34], the measurement method dictated that the endwall of interest be located on the floor due to optical access. To investigate the flat endwall of the planar passage as well as the endwalls of the contoured passage, three different configurations of the test section were used. Figure 4.2 presents a schematic illustrating the three endwall configurations. Note for the contoured passage the area contraction began 1.25C_{ax} upstream of the cascade inlet with an endwall slope of approximately 16°.

Table 4.1 provides the vane geometry and flow conditions for both the planar and contoured passages. For both the CFD analysis and wind tunnel experiments, the exit Reynolds number based on chord and exit velocity was Re_{exit} = 1.0 \times 10^6. To simulate the leakage interface between a combustor and turbine, a two-dimensional slot was placed upstream of the turbine vane on the bottom endwall. Figure 4.2 documents the loc-
Figure 4.2. Schematic of the (a) planar passage, (b) contoured passage with contour on ceiling, and (c) contoured passage with contour on floor.

The boundary layer entering the cascade was measured at a location $4.25C_{ax}$ upstream of the vane stagnation. At this location the boundary layer was measured to be approximately 17% of the span height for both the planar and contoured passages. The measured turbulence intensity was lower than that typically found in an engine at 1.0%. However, the effect of turbulence was considered to be of secondary interest and was not
Table 4.1. Vane Geometry and Flow Conditions

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<td>Exit Reynolds number (Reex)</td>
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<td>Exit span/chord (Sexit/C)</td>
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</tr>
<tr>
<td>Inlet span/exit span (Sn/Sexit)</td>
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<tr>
<td>Velocity Ratio (Uin/Uexit)</td>
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</tr>
<tr>
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<td>1.7 X 10^5</td>
</tr>
<tr>
<td>Exit Reynolds number (Reex)</td>
<td>1.0 X 10^6</td>
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examined in this study.

Computations were performed assuming incompressible, viscous, low-speed conditions using the FLUENT [37] commercial software package. FLUENT [37] is a pressure-based, incompressible flow solver for unstructured meshes. FLUENT [37] also allows for solution-adaptive mesh refinement to resolve the domain based on $y^+$ and local gradients. Second-order discretization was used to implicitly solve for the Reynolds Averaged Navier Stokes (RANS) equations as well as the energy and turbulence equations under steady flow conditions. A RNG, k-ε model with non-equilibrium wall-functions was used for the turbulence model. Using the same modeling technique, Hermanson and Thole [48] were able to show good agreement between computational predictions and experimental measurements of secondary flows. Knost and Thole [49] also used this technique to predict adiabatic effectiveness levels issuing from an upstream
leakage slot and film cooling holes on the endwall of a nozzle guide vane. A comparison between their computational and experimental measurements indicated that the near wall flow field was reasonably predicted using the model described above.

The computational domain was such that the vane was divided at the stagnation point and the trailing edge with a single flow passage being modeled. For the planar passage, the vane was modeled from the endwall to the mid-span with boundary conditions of no-slip and symmetry, respectively. For the contoured passage it was necessary to model the entire passage span, using no-slip conditions on each endwall. Periodic boundary conditions were placed along the pitchwise boundaries of the computational domains. Along the endwall at the inlet to each CFD domain, the measured boundary layer was extrapolated and applied as part of the velocity boundary condition. In the case of the planar passage the velocity inlet was located 1C upstream of the vane cascade. For the contoured passage the velocity inlet was located 1C upstream of the start of the endwall contouring. An outflow boundary condition was located 1.5C downstream of the vane trailing edge for each passage domain. In addition, a 0.1C extension was added to the exit of each domain to avoid highly skewed cells at the domain exit. To simulate the wind tunnel conditions, the freestream turbulence intensity and dissipation length scale were set to 1% and 0.1m, respectively. A slot plenum was also modeled to supply coolant flow to the upstream leakage slot. The mass flow through the slot was controlled using a mass flow boundary condition placed at the entrance to the supply plenum.

A tri-pave meshing scheme was used to discretize each domain. The resulting mesh for the planar passage consisted of approximately $2.8 \times 10^6$ tetrahedral cells. Conversely, the contoured passage consisted of approximately $4.1 \times 10^6$ tetrahedral cells.
since the entire passage span had to be modeled. Once the domain was created and meshed the entity was imported into FLUENT [37] to begin solving. For this study a solution was computed for 1000 iterations. The grid was then adapted based upon $y^+$ values as well as temperature and velocity gradients. All cells where $30 \leq y^+ \leq 60$ was not true were marked for adaption. Next the maximum temperature and velocity gradients were calculated and any cells with gradients higher than half of the maximum were marked for adaption. After marking, the grid was adapted using the hanging node method. After adaption, the computation was continued for another 1000 iterations. This process was repeated until the computation reached 4000 iterations and the residuals of the continuity equation, $U$, $V$, $W$, energy equation, turbulent kinetic energy ($k$), and turbulent dissipation ($\varepsilon$) were changing by less than $1 \times 10^{-3}$. To evaluate whether the results were converged, the lift coefficient and area-averaged endwall effectiveness were monitored after each iteration. The monitored parameters typically converged after two grid adaptions. This indicated that the solution was adequately converged and grid dependent to an acceptably low level. For a more detailed review of the CFD study as well as a comparison between experimental and predicted adiabatic effectiveness results see Appendix A.

### 4.2.1 Secondary Flow Analysis

Computations were performed to compare the differences between the horseshoe vortex among the three endwalls under consideration over a range of leakage flow rates. The velocity vectors of these vortices, which will be referred to as the secondary flow vectors, were determined by transforming the predicted local velocities ($U$, $V$, and $W$ in
Figure 4.3) into the mean flow direction within the vane leading-edge, stagnation plane ($V_s$ and $V_z$). The transformation quantifies the deviation of the local velocities from the inviscid flowfield. This method of analysis has been previously used by both Kang and Thole [50] and Hermanson and Thole [48] to produce secondary flow plots upon planes of interest within a planar passage.

![Diagram of planar vane passage](image)

**Figure 4.3.** (a) Overhead view of the planar vane passage with the initial transform angle in the x-y plane and (b) Side view of the planar vane passage with the final transform angle in the stagnation plane where secondary flow vectors are plotted.

For a planar passage, the transformation was based on the two-dimensional flow field at midspan. For the contoured passage, however, this method of analysis had to be expanded as symmetry no longer existed. To obtain the inviscid flow field an additional computation was performed, using the same technique described in the previous section,
but with inviscid flow. The transformation for the contoured passage was then based on the inviscid turning angles obtained from the inviscid solution all along the stagnation plane under no leakage flow conditions. The secondary flow vectors are plotted using the in-plane components at the vane stagnation ($V_s$ and $V_z$). For a detailed explanation on the derivation of secondary flow vectors see Appendix B.

4.2.2 Heat Transfer Measurements

Heat transfer tests were performed on three surfaces with leakage flow from an upstream slot: the flat endwall of the planar passage, the flat endwall of the contoured passage, and the contoured endwall of the contoured passage. The leakage flow from the slot was measured using a laminar flow element located in the supply line from the upper coolant plenum on the wind tunnel to the slot plenum on the vane test section. The leakage mass flow was set as a percentage of the mass flow entering a single passage. The leakage flow percentage was varied from 0.25% to 1.0%. A constant heat flux surface was provided on the endwall in the form of a thin endwall heater. The heater consisted of a serpentine inconel pattern, encapsulated in Kapton, with a thin copper layer on the side serving as the flow side. The heater was attached to a low thermal conductivity foam endwall to minimize conduction losses. The heater covered the entire endwall, from immediately downstream of the slot to the trailing edge of the second full vane as shown in Figure 4.3.

Experiments were conducted by providing a constant current to the heater and operating the freestream and leakage flow temperatures at room temperature. It was important that the freestream and leakage flow be at the same temperature to ensure that
the heat flux was solely due to convection. Type E calibration thermocouples, embedded in the foam endwall, were placed in thermal contact with the bottom surface of the heater using thermal epoxy. During testing, the lowest temperature difference between the endwall thermocouples and the freestream was approximately 10°C to minimize uncertainty. Once the endwall temperatures reached a steady state value, the thermocouple data was recorded and a set of 5 infrared (IR) thermography images were captured at each measurement location. An inframetrics P20 IR camera was used to acquire the spatially-resolved temperatures on the endwall. With a distance of 55cm between the camera and endwall, each IR image covered an area that was 24cm by 18cm with the viewing area being divided into 320 by 240 pixel locations resulting in a spatial integration of 0.75mm. Once the images were calibrated and transformed, the contour of endwall temperatures could be used to back out heat transfer coefficients with a known heat flux. The heat transfer coefficients were then put into non-dimensional form in terms of Nusselt Number based on axial chord.

The total power supplied by the heater was not totally convected to the freestream. A minimal part of the heat was lost to the room by way of conduction and to the surroundings as radiation. The percentage of the radiation heat flux loss was typically between 6-18% of the total supplied heat flux. The percentage of conduction heat flux loss was typically between 0.6-1.5%. Note that the conduction and radiation losses varied locally. Areas of lower heat transfer, higher temperatures, experienced more heat flux loss through both conduction and radiation due to a larger driving temperature difference. In addition, lateral conduction in the copper layer of the heater was
determined to be insignificant compared to convection given the small integration size over which the IR camera averages the surface temperatures [51].

4.2.3 Uncertainty Analysis

An uncertainty analysis was performed on the measurements of heat transfer using the partial derivative method described by Moffat [45]. Uncertainty in Nusselt numbers was dominated by the uncertainty in surface temperature measurements. For those measurements, a precision uncertainty of approximately ±0.25°C was estimated from the standard deviation of six IR image measurements based on a 95% confidence interval. A bias uncertainty of ±0.7°C was determined in the same way as for adiabatic effectiveness measurements previously described by Thrift et al. [34]. Overall uncertainty in Nusselt numbers was $\partial Nu = \pm 7.5 \ (3\%)$ at a $Nu$ value of 250 and $\partial Nu = \pm 34.4 \ (5.6\%)$ at a $Nu$ value of 620.

4.3 Results and Discussion

The results from the experiments will be presented as contours and line plots of Nusselt number on the three endwalls under consideration: the flat endwall of the planar passage, the flat endwall of the contoured passage, and the contoured endwall of the contoured passage.
4.3.1 Effects of Endwall Contouring with No Leakage

A baseline experiment was performed with no leakage flow. The leakage slot was sealed off and the transition over the slot was smoothed over. These results are presented in terms of Nusselt number contours in Figure 4.4 for each endwall.

Figure 4.4 shows that there is a large area of low heat transfer within the vane passages; however, at the vane stagnation the heat transfer is relatively high. The increase in heat transfer at the vane stagnation is due to the presence of the horseshoe vortex which forms and separates near the vane leading edge. The predicted formation of the horseshoe vortex is shown in Figure 4.5 which compares the secondary flow vectors and total pressure contours between each endwall at vane stagnation.

**Figure 4.4.** Comparison of Nusselt number contours between the three endwalls with no leakage flow.
The heat transfer is reduced not only within the vane passage but at the vane stagnation for both endwalls in the contoured passage relative to the planar passage, as shown in Figure 4.4. The contoured endwall shows the largest reduction in heat transfer levels, both within the passage and at the vane stagnation. Endwall contouring gives a contraction in the area of the passage, providing an additional acceleration that is superimposed on the acceleration from the vane passage. Increased acceleration through a favorable pressure gradient thins the endwall boundary layer approaching the vane casca-

![Image](image.png)

**Figure 4.5.** Secondary flow vectors with no leakage flow at the vane stagnation plane for the (a) flat endwall of the planar passage, (b) flat endwall of the contoured passage, and (c) contoured endwall of the contoured passage.
-de thereby minimizing the horseshoe vortex as shown in Figure 4.5b-4.5c. In addition to weakening the horseshoe vortex, Figure 4.5c shows that the horseshoe vortex separates closer to the vane stagnation for the contoured endwall as a result of the acceleration. Consequently, heat transfer is reduced due to the reduction in secondary flows in the contoured passage.

It is important to mention that while inlet velocities were smaller for the contoured passage, the reduced velocities were not enough to account for the reduction in heat transfer observed in the contoured passage. This was determined by comparing the ratio of Nusselt numbers between the planar and contoured passage to the ratio of Reynolds number raised to the 0.8 power as dictated by the turbulent flat plate correlation for Nusselt number. The ratio of Nusselt numbers was shown to be sufficiently larger than the Reynolds number ratio. The comparisons of these ratios indicated that the difference in inlet velocities was not the reason why the endwalls of the contoured passage experienced lower heat transfer than the planar passage.

To visualize the heat transfer results in a more quantitative manner, measured values were extracted from the endwall data along the paths of inviscid streamlines obtained from a prediction of the passage flowfield using FLUENT [37]. The streamlines are inviscid as they originate at the midspan of the vane at two pitch locations corresponding to y/P = 0.5 and 0.75 as shown in Figure 4.4a. In addition, measured heat transfer values were extracted across the pitch of the vane passage at an axial location 0.35C_{ax} downstream of the vane leading edge as shown in Figure 4.4b.

Figure 4.6 compares the measured Nusselt number values between the three endwalls at no leakage flow and 1.0% leakage flow along the 0.5P and 0.75P streamlines.
respectively. Figure 4.6 shows that the heat transfer levels are high directly downstream of the slot compared to the levels within the passage for all three endwalls at no leakage flow. The heat transfer levels decrease within the passage before increasing as the flow accelerates near the passage throat. The relatively large Nusselt numbers at the start of the streamlines for the no leakage case can be attributed to the thin thermal boundary, as the heat transfer surface begins directly downstream of the slots location. The contoured passage has lower heat transfer levels than the planar passage along the entire length of the streamlines for the no leakage flow case, with the contoured endwall experiencing the lowest heat transfer levels.

To capture the extent at which heat transfer levels are reduced for the endwalls of the contoured passage, Figure 4.7 gives the Nusselt number augmentation values along the same inviscid streamlines illustrated in Figure 4.4. Figure 4.7 shows that relative to the flat endwall of the planar passage, the flat endwall of the contoured passage experiences heat transfer reductions as large as 13% and 16% along the 0.5P and 0.75P streamlines respectively. On average, heat transfer levels are reduced by 7% and 9% along the respective inviscid streamlines. The contoured endwall of the contoured passage experiences even larger heat transfer reductions than the corresponding flat endwall, with maximum heat transfer reductions of 22% and 28% along the 0.5P and 0.75P streamlines respectively. In addition, the average reductions in heat transfer levels along the inviscid streamlines are 16% and 15% respectively for the contoured endwall of the contoured passage.

The effect of the endwall contouring can be also seen in Figure 4.8 which presents the measured Nusselt number values along the pitch of the passage at 0.35C_{ax} downstream of
Figure 4.6. Comparison of measured Nusselt numbers between the three endwalls with no leakage flow and 1.0% leakage flow, sampled along inviscid streamlines released from (a) 50% pitch and (b) 75% pitch.
**Figure 4.7.** Comparison of Nusselt number augmentations with no leakage flow and 1.0% leakage flow, extracted along inviscid streamlines released from (a) 50% pitch and (b) 75% pitch.
Figure 4.8. Comparison of measured Nusselt numbers between the three endwalls along the pitch of the vane passage at $0.35C_{ax}$ with no leakage flow and 1.0% leakage flow.

the vane leading edge for all three endwalls with no leakage flow and 1.0% leakage flow. Figure 4.8 shows that the heat transfer levels near the pressure side of the vane are not influenced by endwall contouring, as the pressure side region is separated from approaching near wall flow by the passage vortex. Near the suction side, however, the heat transfer increases for each endwall, with the contoured passage experiencing the lowest heat transfer levels with no leakage flow.

Figure 4.9 presents the Nusselt number augmentation values along the same pitchwise line used to produce Figure 4.8 with no leakage flow and 1.0% leakage flow. Figure 4.9
shows that the heat transfer is generally lower for the endwalls of the contoured passage relative to the planar passage for the no leakage flow case. On average the heat transfer is reduced by 7% and 14% along the pitchwise line for the flat and contoured endwalls respectively. As described earlier, the reduction in strength of the horseshoe vortex and the subsequent passage secondary flows as a result of freestream acceleration leads to a reduction in endwall heat transfer for the contoured passage.

Figure 4.9. Comparison of Nusselt number augmentations along the pitch of the vane passage at 0.35C_{ax} with no leakage flow and 1.0% leakage flow.
4.3.2 Effects of Leakage Flow

To investigate the effect of leakage flow on endwall heat transfer, heat transfer tests were performed over a range of leakage flows from 0.25% to 1.0%. Figure 4.10 compares the results between each endwall for the lowest leakage flow of 0.25%. The results indicate that the leakage flow increases heat transfer levels near the suction side of the vane leading edge for each endwall. Thrift et al. [34] showed that the majority of the leakage flow is ejected and swept near the suction side leading edge. The interaction of the leakage flow with the mainstream increases the local heat transfer along the path of the coolant.

![Figure 4.10. Comparison of Nusselt number contours between the three endwalls with 0.25% leakage flow.](image)

At the vane stagnation a reduction in the heat transfer is observed for each endwall relative to the respective no leakage flow case. This relative reduction in heat transfer at
the vane leading edge can be attributed to a reduction in size and strength of the horseshoe vortex. At the lowest leakage flow rate there is significant ingestion of mainstream gas into the upstream slot at stagnation for each endwall. Ingestion of mainstream gas into the upstream slot was confirmed through temperature measurements within the upstream slot by Thrift et al. [34] and also predicted using CFD. The ingestion into the slot removes the incoming endwall boundary layer resulting in the formation of a new boundary layer with minimal distance to form a full horseshoe vortex relative to the no slot flow configuration. A smaller horseshoe vortex results in sequentially weaker secondary flows through the passage, consequently reducing heat transfer.

Figure 4.11 compares the Nusselt number contours between each endwall at a leakage flow rate of 0.5%. The ejection of more leakage flow leads to an increase in heat transfer.

![Figure 4.11](image)

**Figure 4.11.** Comparison of Nusselt number contours between the three endwalls with 0.5% leakage flow.
near the suction side leading edge for each endwall. In addition, the heat transfer at the vane stagnation returns to the levels observed with no slot flow. As indicated by Thrift et al. [34], mainstream ingestion into the slot was reduced but not completely eliminated at a leakage flow rate of 0.5%.

At leakage flows above 0.5%, Thrift et al. [34] showed that ingestion was eliminated across the entire pitch of the leakage slot. With an increase in the leakage flow to 0.75% the heat transfer near the suction side leading edge continues to increase for each endwall as shown in Figure 4.12. In addition to the shrinking of the relatively low heat transfer area within the passage, an increase in the heat transfer at the vane stagnation is observed at each endwall relative to the respective low leakage flow cases. These trends in heat transfer levels continue to intensify as shown in Figure 4.13 with a subsequent increase in the leakage flow to 1.0%.

![Figure 4.12. Comparison of Nusselt number contours between the three endwalls with 0.75% leakage flow.](image)

**Figure 4.12.** Comparison of Nusselt number contours between the three endwalls with 0.75% leakage flow.
The relatively high heat transfer levels at the vane stagnation and the subsequent reduction in size of the low heat transfer area within the passage indicates an intensification of the passage secondary flows relative to the low leakage flow cases. The ejection of coolant at the vane stagnation promotes the formation of the horseshoe vortex. Figure 4.14 presents the secondary flow vectors superimposed on the total pressure contours at the vane stagnation plane for each endwall at a leakage flow rate of 1.0%.

![Figure 4.13.](image.png)

**Figure 4.13.** Comparison of Nusselt number contours between the three endwalls with 1.0% leakage flow.

Figure 4.14 shows that the ejecting coolant promotes the separation of the incoming endwall boundary layer, leading to an intensification of the horseshoe vortex relative to the no leakage flow configuration shown in Figure 4.5. The consequence of intensifying the horseshoe vortex is the dramatic increase in the Nusselt numbers at the vane stagnati-
Figure 4.14. Secondary flow vectors with 1.0% leakage flow at the vane stagnation plane for the (a) flat endwall of the planar passage, (b) flat endwall of the contoured passage, and (c) contoured endwall of the contoured passage.

-on relative to lower leakage flow cases. In addition, the relatively low heat transfer region within the vane passages shows an increase in heat transfer for each endwall.

These findings are similar to those by Kost and Nicklas [28] and Nicklas [29] who made aerodynamic and thermodynamic measurements within a linear turbine cascade with a transonic flow field utilizing leakage flow from an upstream slot. Kost and Nicklas [28] showed that the horseshoe vortex was strengthened by the ejection of
coolant from the slot. This was attributed to the location of the slot, being positioned in the region of the saddle point just 0.2C_{ax} upstream of the vane cascade. Nicklas [29] observed that the higher intensity of the horseshoe vortex led to increased heat transfer coefficients at the endwall in proximity of the leading edge. In a similar experiment Kost and Mullaert [30] investigated the effect of moving the slot further upstream to 0.3C_{ax} upstream of the vane cascade. Results showed that the intensification of the horseshoe vortex could be avoided by moving the slot further upstream. It was suggested that the saddle point is a sensitive region where coolant ejection should be avoided.

The quantitative effect of leakage flow on each endwall can be seen by investigating the line plots presented in Figures 4.6 and 4.8. Figure 4.6 shows that each endwall experiences an increase in heat transfer along the path of the inviscid streamlines relative to the respective no leakage flow case. The introduction of 1.0% leakage flow increases the average heat transfer along the 0.5P streamline by 14%, 22%, and 28% for the flat endwall of the planar passage, the flat endwall of the contoured passage, and the contoured endwall of the contoured passage respectively. Correspondingly, an average increase in heat transfer of 24%, 31% and 39% is observed along the 0.75P streamline with the introduction of 1.0% leakage flow. Note that the introduction of leakage flow is most detrimental to endwall heat transfer for the contoured passage relative to the planar passage. The contoured endwall of the contoured passage experiences the largest increase in heat transfer with the introduction of 1.0% leakage flow along the inviscid streamlines.

Similar to the inviscid streamlines, Nusselt number values along the pitchwise line presented in Figure 4.8 increase for each endwall with 1.0% leakage flow. On average,
the heat transfer is increased by 19%, 17%, and 26% for the flat endwall of the planar passage, the flat endwall of the contoured passage, and the contoured endwall of the contoured passage respectively. Again, the largest increase in heat transfer along the pitchwise line is observed on the contoured endwall with the introduction of 1.0% leakage flow.

To further investigate the effect of leakage flow on the endwall heat transfer, Figure 4.15 presents the Nusselt numbers along the pitch of the passage at 0.35C_{ax} downstream of the vane leading edge for all three endwalls over the entire range of leakage flow rates. Figure 4.15 shows that the heat transfer levels near the pressure side of the vane are mostly unaffected by the introduction of leakage flow. As discussed previously, the pressure side region is separated from the upstream flow conditions by the passage vortex. Near the suction side, however, the heat transfer increases with leakage flow for each endwall. The increase in heat transfer near the suction side can be attributed to the interaction between the leakage flow and the near wall flow.

Near the midpitch location, at the edge of the mixing zone between the leakage and near wall flow, the flow interaction does not dominate the endwall heat transfer. This is shown in Figure 4.15 near Y/P = 0.5 where the heat transfer levels are lower than the respective no leakage flow case for the smaller leakage flow rates at all three endwalls. As mentioned previously, ingestion of mainstream flow into the slot is observed at each endwall at leakage flow rates below 0.75%. Flow ingestion weakens the horseshoe vortex and the subsequent passage secondary flows, reducing the heat transfer.
Figure 4.15 continued on next page…
Figure 4.15. Comparison of Nusselt numbers along the pitch of the vane passage at 0.35Cₘₐₓ over a range of leakage flow rates for the (a) flat endwall of the planar passage, (b) flat endwall of the contoured passage, and (c) contoured endwall of the contoured passage.

4.3.3 Effects of Leakage Flow on the Contoured Passage

The effect of leakage flow on the contoured passage can be seen by comparing the contours in Figures 4.10-4.13. Figures 4.10-4.13 show that the endwalls of the contoured passage continue to experience less heat transfer than the flat endwalls of the planar passage with the introduction of leakage flow. The reduction in heat transfer relative to the planar passage is evident at both the vane stagnation and the suction side leading edge on the endwalls of the contoured passage.
Figure 4.7 shows that the Nusselt number values are generally lowest for the contoured passage along the inviscid streamlines for the 1.0% leakage flow cases compared to the planar passage. In respect to the no leakage flow cases, the reduction in heat transfer relative to the planar passage is lower with the introduction of 1.0% leakage flow. On average, heat transfer levels on the flat endwall of the contoured passage are reduced by 3% and 6% along the 0.5P and 0.75P streamlines respectively. As before with the no leakage flow case, the contoured endwall of the contoured passage experiences even larger heat transfer reductions than the corresponding flat endwall with the introduction of 1.0% leakage flow. The average reduction in heat transfer levels along the inviscid streamlines is 7% and 6% respectively for the contoured endwall of the contoured passage relative to the flat endwall of the planar passage. The relative increase in heat transfer levels for the contoured passage with the introduction of 1.0% leakage flow is also evident across the pitch of the vane passage as shown in Figure 4.9. Although the contoured passage heat transfer results are still generally lower than in the planar passage, the average reduction in heat transfer along the pitchwise line is 8% and 9% for the flat and contoured endwalls respectively.

Figure 4.8 shows that the heat transfer near the pressure side surface is similar among all three endwalls with the introduction of 1.0% leakage flow. This observation suggests that the redevelopment of the endwall boundary layer, downstream of the passage vortex, is similar between all three endwalls. Approaching the suction side, however, heat transfer levels increase dramatically with 1.0% leakage flow. The lowest heat transfer levels occur on the contoured endwall of the contoured passage with the flat endwall of the planar passage showing the largest heat transfer levels.
As discussed in the previous section, the endwall heat transfer can be lower near the midpitch of the vane passage at the 0.35Cax axial location with no leakage flow than with leakage flow. As shown in Figures 4.15a and 4.15b, there is a reduction in heat transfer near the midpitch location for the flat endwalls of the planar and contoured passages relative to the no leakage flow case at leakage flows of 0.25% and 0.5%. For the contoured endwall, however, only the 0.25% leakage flow heat transfer results are lower than the no leakage flow case as shown in Figure 4.15c. Recall that the reduced injection angle for the slot in the contoured endwall results in less ingestion of mainstream flow. The heat transfer results indicate that the coolant ejection on the contoured endwall is significant enough to intensify the secondary flows at the lowest leakage flow compared to the no leakage flow case.

Figure 4.16 presents the heat transfer augmentation values with and without leakage flow along the same pitchwise line used in Figure 4.15 for the flat and contoured endwalls of the contoured passage respectively. Figure 4.16a shows that the augmentation values are generally below unity on the flat endwall of the contoured passage except near the midpitch location. Similarly, Figure 4.16b shows that the heat transfer levels on the contoured endwall are relatively smaller than the planar passage except for the 1.0% leakage flow case near midpitch. Approaching the suction side of the vane passage, Figure 4.16 indicates that heat transfer augmentation becomes increasingly less than unity for both endwalls of the contoured passage. In addition, the augmentation values become similar near the suction side of the vane passage for each endwall in the contoured passage over the range of leakage flows.
Figure 4.16. Comparison of Nusselt number augmentations along the pitch of the vane passage at $0.35C_{ax}$ over a range of leakage flow rates for the (a) flat endwall of the contoured passage and (b) contoured endwall of the contoured passage.
4.4 Conclusions

Measurements of heat transfer were presented on the endwalls of a planar and contoured passage with and without leakage flow from a two-dimensional upstream slot. Four leakage mass flow rates were tested ranging from 0.25% to 1.0%. CFD simulations were also performed to predict the flow field at the vane stagnation plane. A velocity transformation was used to calculate the secondary flows and highlight the horseshoe vortex.

With and without leakage flow the endwalls of the contoured passage were shown to have lower heat transfer levels than the corresponding flat endwall of the planar passage. In all experiments, the contoured endwall showed the lowest heat transfer levels. Endwall contouring provides a favorable pressure gradient for the incoming flow, thinning the endwall boundary layer approaching the vane cascade. As seen in plots of secondary flow vectors this weakens the horseshoe vortex when no leakage flow is present, reducing heat transfer.

With increasing leakage flow the heat transfer levels were shown to increase near the suction side leading edge for each endwall. Measurements of adiabatic effectiveness by Thrift et al. [34] indicated that the majority of the leakage flow was swept into this region. The interaction between the leakage flow and the approaching near wall flow served to enhance heat transfer. Regarding reduced heat transfer levels on the endwalls of the contoured passage relative to the flat endwall of the planar passage, augmentation values were shown to decrease with the introduction of leakage flow. In addition, the relatively low heat transfer area within the vane passage was shown to shrink in size with increasing leakage flow. This signified the presence of the passage vortex, sweeping
across the passage pitch and increasing heat transfer across the breadth of its path. Near the pressure side surface, however, heat transfer levels were similar among all three endwalls and were mostly unaffected by the introduction of leakage flow. The majority of the endwall near the pressure side is cut off from the influence of the incoming boundary conditions as the passage vortex sweeps the approaching endwall flow into the mainstream. The removal of the endwall flow causes a new boundary layer to form. The heat transfer levels near the pressure side surface suggest that the redevelopment of the endwall boundary layer is similar among the three endwalls.

The relative closeness of the upstream leakage slot to the vane stagnation dictated the interaction between the leakage flow and the formation of the horseshoe vortex. Coincidently, this had a dramatic effect on the endwall heat transfer. In this study the location of the leakage slot served to weaken the formation of the horseshoe vortex at low leakage flow rates, but intensified the horseshoe vortex at high leakage flow rates. The intensification of the horseshoe vortex led to an increase in heat transfer near the leading edge of the vane at stagnation relative to the no leakage flow configuration.
Chapter 5

Effects of Orientation and Position of the Combustor-Turbine Interface on the Cooling of Vane Endwall [52*]

5.1 Introduction

Throughout the history of gas turbine development, thermal efficiencies have continuously been driven up by a desire to increase power output and lower fuel consumption. In the past decade the rising cost of fuel and an increasing worldwide effort to reduce environmental impact have only served to increase the demand for higher thermal efficiencies. Increasing engine thermal efficiency is directly achieved by increasing the turbine inlet temperature. Consequently, this places increased heat loads on turbine components that are already extensively cooled, particularly the first stage turbine vanes. Current cooling methods consist of full coverage film cooling through discrete holes on the surface of the vanes as well as the endwalls. Although it is generally not considered as a part of the cooling design, leakage from the combustor-turbine interface gap can also provide substantial cooling to the endwalls. The current literature lacks a systematic foundation of results related to the many parameters that influence the cooling performance of the combustor-turbine interface gap. As such, the interface gap is an area where improvements can possibly be made.

This paper reports the effects of orientation and position of an upstream leakage slot on the cooling of a low-pressure, nozzle guide vane endwall. In addition, time-resolved flowfield measurements within the vane stagnation plane are presented. By

* In press

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simultaneously varying the slot width, the effects of varying the mass flow rate through the upstream leakage slot while maintaining the momentum flux ratio is also considered.

5.2 Experimental Methods and Benchmarking

Experiments were performed in a low speed, closed loop wind tunnel, depicted in Figure 5.1 and previously described by Thrift et al. [34]. The flow was driven through the wind tunnel by a 50 hp fan controlled by a variable frequency drive. After passing the fan the flow was turned 90° and then passed through a finned-tube heat exchanger. The primary heat exchanger circulated temperature controlled water at approximately 10°C to remove the initial heat supplied to the flow by the fan.

![Diagram of the wind tunnel](image)

**Figure 5.1.** Depiction of the low speed, closed loop wind tunnel.

The flow was then turned another 90° before encountering a porous plate with circular holes providing a 75% reduction in open flow area. The plate, positioned only over the main flow path, diverted flow to the two outer secondary flow paths where the flow would be used as coolant. Note that only the top secondary flow path was used in this study. Flow in each of the secondary flow paths traveled through secondary finned-
tube heat exchangers supplied with the same water used in the primary heat exchanger. The secondary heat exchangers provided additional cooling to the secondary flows before passing into respective plenums. The temperature of the secondary flow could be controlled by adjusting the amount of water passing through the secondary heat exchangers through needle values located upstream of each heat exchanger. As only the top secondary flow was used, the valve to the lower secondary heat exchanger was always closed. During experiments the secondary flow was cooled to approximately room temperature at 25°C. Flow was drawn from the upper plenum and into the appropriate leakage coolant plenum on the attached test section using a 2 hp blower.

After the split, the primary flow continued down the center path where it encountered an electrical resistance heater bank capable of supplying 55kW of heat. Using a small percentage of the heater bank capacity, the main flow was heated to approximately 50°C to increase the temperature difference between the main flow and coolant flow. After the heater bank, the main flow passed through a series of screens used for flow straightening and then into a contracted straight flow section. The contraction reduced the flow area from 1.11m$^2$ to 0.62m$^2$ through rounded inlets after which the flow area was constant. At the exit of this section was the experimental test section where all measurements were performed. The test section incorporated a 90° bend to assist in the turning of the flow through the vane cascade. Air exiting the test section was turned by a final 90° elbow before encountering the fan and completing the closed-loop.

The vane test section was a two-dimensional, linear vane cascade as illustrated in an over-head view schematic in Figure 5.2. The test section contained two full nozzle guide vanes and a third partial vane connected to a flexible wall to maintain the desired
pressure distribution along the three vanes. Constructed of low-density closed cell polyurethane foam, the vane design was a spanwise extrusion of a two-dimensional midspan vane geometry. The vanes were scaled up by a factor of 2.4 from engine size to achieve high measurement resolution. A description of the nozzle guide vane parameters is given in Table 5.1.

Figure 5.2. Schematic of the linear vane cascade with the leakage slot at the nominal location and the TRDPIV setup.
Table 5.1. Vane Geometry and Flow Conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scaling factor</td>
<td>2.4</td>
</tr>
<tr>
<td>Scaled vane chord (C)</td>
<td>46cm</td>
</tr>
<tr>
<td>Axial chord/chord (C&lt;sub&gt;ax&lt;/sub&gt;/C)</td>
<td>0.49</td>
</tr>
<tr>
<td>Pitch/chord (P/C)</td>
<td>1.01</td>
</tr>
<tr>
<td>Midspan/chord (0.5S/C)</td>
<td>0.6</td>
</tr>
<tr>
<td>Inlet/exit Reynolds number (Re&lt;sub&gt;in&lt;/sub&gt;/Re&lt;sub&gt;exit&lt;/sub&gt;)</td>
<td>2.0 x 10&lt;sup&gt;5&lt;/sup&gt;, 1.0 x 10&lt;sup&gt;6&lt;/sup&gt;</td>
</tr>
<tr>
<td>Inlet, exit angle (α&lt;sub&gt;in&lt;/sub&gt;/α&lt;sub&gt;exit&lt;/sub&gt;)</td>
<td>0°, 79°</td>
</tr>
<tr>
<td>Inlet, exit Mach number (Ma&lt;sub&gt;in&lt;/sub&gt;/Ma&lt;sub&gt;exit&lt;/sub&gt;)</td>
<td>0.022, 0.12</td>
</tr>
</tbody>
</table>

To simulate leakage flow from the combustor turbine interface gap a two-dimensional slot was placed on the bottom endwall, upstream of the vane stagnation as illustrated in Figure 5.2. The upstream slot was interchangeable so that the flow metering width, slot orientation, and slot location could be easily changed. Note that the upstream slot is at the nominal location in Figure 5.2, located at x = -0.17C<sub>ax</sub> from the vane stagnation.

To accurately model the characteristics of the leakage flow, momentum flux ratio was matched between the engine and experiments. Based on a realistic engine pressure ratio between the leakage plenum and freestream, a momentum flux ratio of I = 2.8 was determined. Unlike the engine, the wind tunnel operates with a density ratio of approximately 1. So to achieve a large momentum flux ratio required a very small slot width to meet the proper slot coolant exit velocity. In an engine, the leakage mass flux is only altered through changes in the slot width since the pressure ratio is maintained constant. The constant pressure ratio results in a constant momentum flux ratio. In the current experiments, three upstream slot widths were chosen to model the interface gap so that a constant momentum flux ratio of I = 2.8 could be achieved for several different leakage mass flux conditions. As shown in Table 5.2 a slot width of 3.3mm, 2.3mm, and 1mm were considered for mass flux ratios of 1.0%, 0.7%, and 0.3% respectively.
Table 5.2. Slot Flow Parameters

<table>
<thead>
<tr>
<th>Slot Width (mm)</th>
<th>MFR (%)</th>
<th>Momentum Flux Ratio, I</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.3</td>
<td>1.0</td>
<td>2.8</td>
</tr>
<tr>
<td>2.3</td>
<td>0.7</td>
<td>2.8</td>
</tr>
<tr>
<td>1</td>
<td>0.3</td>
<td>2.8</td>
</tr>
</tbody>
</table>

Experiments were also performed with a slot orientation of 90° and 45° as indicated in Figure 5.2. Each slot had a flow length-to-axial chord ratio of 0.17. Subsequently, this gives a changing flow length-to-width ratio of 11.7, 16.8, and 38.6 for the 3.3mm, 2.3mm, and 1mm slot widths respectively. The relatively large flow length-to-width ratios for each slot width, however, suggest that the flow becomes fully developed before exiting the slot. Finally, the effects of slot location were considered by moving the slot further upstream and downstream from the nominal location, to x = -0.34C_{ax} and -0.05C_{ax} respectively.

The boundary layer entering the cascade was measured at a location 4.25C_{ax} upstream of the vane stagnation by Thrift et al. [34] using the same inlet conditions. Table 5.3 lists the turbulent inlet boundary layer parameters, which were maintained throughout this study. The measured turbulence intensity was lower than that typically found in an engine at 1.0%. The effect of turbulence, however, was not considered in this study as past studies by the authors have evaluated this effect [36].

Table 5.3. Inlet Turbulent Boundary Layer Characteristics

| Boundary layer thickness/midspan (δ/0.5S) | 0.34   |
| Displacement thickness/midspan (δ’/0.5S)  | 0.038  |
| Momentum thickness/midspan (θ/0.5S)       | 0.03   |
| Shape factor (δ’/θ)                       | 1.3    |
| Momentum thickness Reynolds number (Reθ)  | 4245   |
5.2.1 Mainstream and Coolant Flow Settings

Before performing any experiment, the inlet velocity distribution across the cascade pitch and the vane pressure distributions along the midspan were verified. The inlet velocity distribution was measured approximately 0.8C upstream of the cascade inlet at 50% span height. Measurements were taken at 8 locations spanning from ±0.75P about the center vane stagnation. The inlet velocity range across the width of the cascade varied less than 5% from the pitch averaged mean for all experiments. Static pressure measurements around the circumference of the vane at midspan were compared with predictions from a computational study. The computational study was previously performed by Thrift et al. [34] for incompressible, viscous, low-speed conditions using FLUENT [37]. A detailed description of the computational method and domain used for generating the predicted pressure distribution is provided by Thrift et al. [35]. As shown previously by Thrift et al. [34], the measured and predicted pressure distributions agreed indicating that the inviscid flowfield around each vane was matched to the predicted curve.

The mass flux issuing from the upstream leakage slot was measured using a laminar flow element located within the supply pipe to the slot plenum. The leakage coolant issuing from the upstream slot and entering each passage was then defined as a prescribed percentage of a single passages mass flow rate. As shown previously in Table 5.2, coolant mass flux ratios ranged from 0.3% to 1.0%. With the known mass flux and leakage slot width, an average momentum flux ratio could be calculated according to Equation 5.1 below.
The vane span height was maintained throughout this study. The slot width, however, had to be reduced simultaneously with the mass flux ratio to maintain a constant momentum flux ratio as indicated in Equation 5.1 and shown in Table 5.2. As mentioned previously the slots were individual, interchangeable components which were machined to the desired widths. The slot widths were verified with the use of calipers and gage blocks to within approximately ±0.1mm.

\[ I = \frac{MFR^2 S^2}{w^2} \]  \hspace{1cm} (5.1)

5.2.2 Adiabatic Effectiveness Measurements

Spatially resolved, adiabatic wall temperatures were obtained from infrared (IR) measurements of the endwall. Adiabatic effectiveness experiments were performed at steady state conditions with a temperature difference between the freestream and leakage coolant of approximately 25°C to reduce measurement uncertainty. To ensure the measured endwall temperatures were adiabatic, the endwall was made of a 2.54cm thick plate of low density closed cell polyurethane foam, which has a very low thermal conductivity (0.03 W/mK). The foam endwall was painted black to maintain a high emissivity on the endwall surface thus providing good resolution of the surface temperatures. Type-E thermocouples were placed throughout each vane passage for calibration of the IR images.

An Inframetrics P20 IR camera was used to capture the spatially-resolved adiabatic temperatures over the entire endwall. The ceiling of the test section contained 14
viewing ports, distributed across both vane passages to allow unobstructed access of the IR camera to the entire bottom endwall. At each viewing location the IR camera was placed perpendicular to the endwall surface at a distance of approximately 55cm. Based on this distance and the camera viewing angles, each IR image covered an area that was 24cm by 18cm. The resolution of the camera was 320 X 240 pixels resulting in a spatial integration of 0.75mm. To reduce uncertainty, five images were taken at each of the 14 different locations and averaged to produce the final image at each location. Note that each single image was also an average of 16 frames taken by the camera.

Images were post-calibrated by determining the emissivity and background temperature of the image through matching of the IR captured surface temperatures with the acquired thermocouple measurements. Although the thermal conductivity of the foam endwall was low, it was necessary that a small conduction correction still be used. A one-dimensional conduction correction as described by Ethridge et al. [44] was applied to all adiabatic effectiveness measurements. The correction involved measuring the endwall surface effectiveness with no coolant flow. A correction value of \( \eta_o = 0.1 \) was measured within the vane passages. Upstream of the leakage slot, however, a correction value of \( \eta_o = 0.15 \) was measured. The endwall upstream of the leakage slot was made of medium density fiberboard with a thermal conductivity (0.13 W/m-K) higher than that of the foam making up the cooled surface, resulting in slightly larger conduction losses.

Using the partial derivative method described by Moffat [45] an uncertainty analysis was performed on the measurements of adiabatic effectiveness based on the uncertainties associated with the measured temperatures. Based on a 95% confidence interval the precision uncertainty of the IR temperature measurements was ±0.2°C. The bias
uncertainty for an image was taken as the root-sum-square of the thermocouple bias uncertainty (±0.2°C) and the average deviation of the calibrated images from the thermocouples (±0.5°C) resulting in a bias uncertainty of ±0.54°C. Combining the bias and precision uncertainties, a total uncertainty in adiabatic effectiveness was then found to be $\hat{\eta} = ±0.03$ over the range $\eta = 0.03$ to 0.7.

5.2.3 Particle Image Velocimetry Measurements

The stagnation plane flowfield upstream of the vane was recorded using a high-image-density, time resolved, digital particle image velocimetry (TRDPIV) system. Figure 5.2 illustrates the experimental setup used when performing TRDPIV. TRDPIV is a non-intrusive, laser-optical measurements technique based on the illumination and tracking of particles which follow the flowfield [53-57]. The TRDPIV system consisted of a high frequency double pulsed laser, high speed CMOS camera, and controller that synchronized the timing between the laser and camera. The dual cavity 15W Nd:YAG laser was capable of firing at 10kHz per laser cavity. The laser produced double pulses of green light at a wavelength of 527nm with a pulse duration less than 180ns and an energy level of 10mJ/pulse. The CMOS camera was capable of recording images at 2kHz with a spatial resolution of 1024x1024 pixels. Flowfields were captured using a camera sampling frequency of 1kHz for 3 seconds resulting in the recording of 3000 image pairs. Time averaged flowfields were calculated using all 3000 image pairs. The time delay between laser pulses and the resulting image pairs was adjusted for each run to obtain a bulk particle displacement of approximately 7 pixels. Di-Ethyl-Hexyl-Sebakan
(DEHS) seeder particles with an approximate mean diameter of 1µm were introduced directly upstream of the wind tunnel blower using a Laskin nozzle.

The raw images were corrected for the small off normal viewing angle and then vector fields were produced with cross-correlation using a standard cyclic FFT-based algorithm [58]. A decreasing, multi-pass processing technique was employed with a single pass at an interrogation window size of 64x64 pixels and then a double pass at an interrogation window size of 32x32 pixels with 50% overlap resulting in a final vector spacing of 16x16 pixels. In some cases it was necessary to use a final interrogation window size of 16x16 pixels with 50% overlap resulting in a vector spacing of 8x8 pixels to capture smaller flow features. Multi-pass and final pass post processing was performed using a 4-pass regional median filter to remove spurious vectors.

To qualify the capabilities of the TRDPIV system, flowfield measurements were performed in the stagnation plane with no leakage slot present to observe the dynamics of the unsteady HSV. The particular dynamics of the HSV is a well studied phenomena by researchers as early as Devenport and Simpson [59] using LDV and as recent as Praisner and Smith [60] and Sabatino and Smith [61] using PIV. Although these studies investigated the HSV upstream of a streamlined cylinder, the studies still serve as valid baselines for comparison. Figure 5.3 presents instantaneous flowfields of the HSV transitioning from the dominate flow mode to a secondary flow mode. The first contour plot shows the clock-wise rotating HSV at an axial position of approximately $x/C_{ax} = -0.07$. Directly upstream of the HSV is a secondary vortex with the opposite sense of rotation followed by a small tertiary vortex with the same rotation direction as the HSV.
Figure 5.3. Instantaneous flowfield vectors and contours of vorticity in the stagnation plane with no leakage flow showing the transition between two flow modes.
This mode dominates the flowfield for the majority of the time only to be interrupted by the quasi-periodic switching to a second flow mode. As shown in Figure 5.3b the dominate mode is interrupted when the secondary vortex is moved away from the endwall and an inrush of fluid is observed behind the HSV, displacing it upstream. A short time later, Figure 5.3c shows that the size of the HSV is reduced while being displaced further upstream and closer to the endwall. This quasi-periodic switching is identical to that observed by Praisner and Smith [60] and Sabatino and Smith [61].

Although Devenport and Simpson [59] were unable to capture the entire time-resolved flowfield using LDV, they were able to comment on the bi-modal switching of the near wall velocities. For comparison, Figure 5.4 presents probability density functions of the offset streamwise velocity at two different spanwise locations taken at the average axial position of the HSV core, \( x/C_{ax} = -0.08 \). The first location, \( z/0.5S = 0.0017 \) is below the average HSV core and shows a bimodal velocity distribution indicative of the switching between flow modes. In Figure 5.4b, above the vortex core at \( z/0.5S = 0.027 \), only a single velocity peak is observed. The velocity histograms presented in Figure 5.4 are very similar to those presented by Devenport and Simpson [59]. Devenport and Simpson [59] commented that the double-peaked histogram implies that the near wall velocity has two preferred states which were deemed the backflow and zeroflow modes. In addition, Devenport and Simpson [59] showed that the turbulent fluctuations were very high near the vortex region as a result of the structures unsteady motion. Radomsky and Thole [36] also identified increased streamwise fluctuations in the region of the HSV. The two flow modes represent the quasi-periodic switching observed by Praisner and
Smith [60] and Sabatino and Smith [61] and shown in Figure 5.3. The observed dynamics of the HSV compare well with the results of past researchers.

**Figure 5.4.** Histogram of velocity probability density for no leakage flow at $x/C_{ax} = -0.08$ and (a) $z/S = 0.0017$ and (b) $z/S = 0.027$. 

The observed dynamics of the HSV compare well with the results of past researchers.
5.3 Results and Discussion

A number of experiments were performed for this study with the most representative results for the effects of slot orientation and slot location being presented. Integrated in with the presentation of these results will also be a discussion on the effects of mass flux ratio. Although the effects of mass flux ratio for a constant momentum flux ratio have been documented by past researchers, a presentation of these results is helpful in supporting precedent conclusions.

5.3.1 Effects of Slot Orientation

To study the effects of slot orientation, two separate slot angles of 90° and 45° were considered with both being located at x/Cax = -0.17, upstream of the vane stagnation. Experiments were performed at a constant momentum flux ratio for several mass flux ratios. As shown previously in Table 5.2, three slot widths were considered for this study. Slot widths of 3.3mm, 2.3mm, and 1mm were studied for MFR’s of 1.0%, 0.7%, and 0.3% respectively. By reducing the slot width and mass flux ratio simultaneously the momentum flux ratio could be maintained at approximately 2.8 as according to Equation 5.1. Maintaining the momentum flux ratio instead of the mass flux ratio represents a realistic constraint on turbine designers. Within an engine, the pressure difference between the coolant and the exit pressure at the endwall remains constant. Maintaining the same pressure ratio results in a constant momentum flux ratio given that pressure difference scales with velocity squared.
Figure 5.5 compares adiabatic effectiveness levels between the 90° and 45° slot orientations at the nominal location for several different mass flux ratios. Inspection of Figure 5.5 shows that at each mass flux ratio the adiabatic effectiveness levels are substantially higher for the 45° slot orientation. The improvement in cooling performance for the 45° slot is likely related to the ejection angle being closer aligned to the plane of the

**Figure 5.5.** Comparison of adiabatic effectiveness contours at several different mass flux ratios with $l = 2.8$ between the (a) 90° and (b) 45° slot orientations at $x/C_{ax} = -0.17$. 
endwall. To understand how the ejected coolant alters the stagnation plane flowfield, Figure 5.6 presents time-averaged flowfield results for both the 90° and 45° slot orientations with MFR = 1.0% and I = 2.8.

**Figure 5.6.** Average flowfield vectors, streamlines, and contours of vorticity in the stagnation plane with MFR = 1.0% and I = 2.8 for the (a) 90° and (b) 45° slot orientations at $x/C_{ax} = -0.17$. 
The stagnation plane flowfields show vastly different flow patterns between the 90° and 45° slot orientations. In the case of the 90° slot, all of the coolant ejects into the spanwise direction. As indicated in Figure 5.6a, the ejection of coolant perpendicular to the vane endwall results in the separation of the incoming boundary layer. The static pressure gradient, formed along the span of the vane as a result of the stagnating boundary layer, forces the separated leakage flow to turn toward the endwall to form the leading edge vortex. The time-resolved results indicate that the ejected coolant sustains the large leading edge vortex as the structure never dissipates. In addition to the large leading edge vortex, a small tertiary vortex with the same direction of rotation is also present directly upstream of the ejected coolant. Unlike the large vortex structure, the tertiary vortex exhibits some unsteady dynamics, dissipating and then reforming. For the majority of the time, however, the tertiary structure is present as it shows up in the time-averaged results presented in Figure 5.6a. Note that the time-averaged tertiary vortex in Figure 5.6a is similar to that seen in the instantaneous flowfields with no leakage flow presented in Figure 5.3.

For the 45° slot orientation, no permanent vortex structure is present. As shown in Figure 5.6b, the coolant is ejected along the endwall. Ejecting coolant along the endwall plane fills in the approaching boundary layer in the low velocity region near the wall. Thickening the boundary layer in the near wall region produces a static pressure gradient near the vane-endwall junction that is away from the endwall, opposite of that seen for the 90° slot orientation. Proof for the presence of this pressure gradient can be seen in Figure 5.6b as a region of positive vorticity where the endwall flow begins to turn up toward midspan on approach to the vane surface. As a result of the strong turning of the
flow toward midspan near the vane stagnation, an intermittent vortex structure is occasionally formed. The formation of this structure occurs where strong inflection points exist in the time-averaged streamlines presented in Figure 5.6b. These strong inflection points indicate where the spanwise direction of the velocity is reversed, setting up a scenario where a counter-clockwise rotating vortex can form.

Figure 5.7 presents three sequential, instantaneous flowfields to illustrate the formation of these vortical structures. The first flowfield in Figure 5.7 depicts a small vortex at $z/0.5S = 0.12$. The second flowfield shows that a short time later the single vortex near the vane surface is gone and appears to be replaced by a strong downwash of fluid from $z/0.5S = 0.15$ to 0.05. Contrarily, an increase in the upwash of fluid near the vane-endwall junction is observable. Subsequently, the final flowfield shows that the interaction of the two opposing flows near the vane surface results in the formation of a counter-clockwise rotating vortex at approximately the same position observed in the first time instance. In addition, a vortex is also formed near the vane-endwall junction where the turning of fluid toward the midspan is the greatest. Although Figure 5.7 indicates small vortical structures forming at both $z/0.5S = 0.03$ and 0.12, the structures do not appear to convect up the surface of the vane. Instead, the structures were created along the inflection points within $-0.05 < x/C_{ax} < 0$ and $0.03 < z/0.5S < 0.15$ and were quickly dissipated or moved out of the stagnation plane.

To observe the effect of slot orientation on the stagnation flowfield at a lower MFR, Figure 5.8 presents time-averaged flowfield results for both the 90° and 45° slot orientations with MFR = 0.7% and $I = 2.8$. Comparison of the flowfield results in Figures 5.6 and 5.8 show that the overall flow structure is very similar between the two
Figure 5.7. Instantaneous flowfield vectors, streamlines, and contours of vorticity in the stagnation plane with MFR = 1.0% and I = 2.8 for the 45° slot orientation at x/C_{ax} = -0.17 for three time instances.
Figure 5.8. Average flowfield vectors, streamlines, and contours of vorticity in the stagnation plane with MFR = 0.7% and $I = 2.8$ for the (a) 90° and (b) 45° slot orientations at $x/C_{ax} = -0.17$.

MFR cases. For the 90° orientation, the ejecting coolant lifts off the approaching endwall flow and is then rolled up into a permanent vortical structure. Likewise, flow ejecting from the 45° slot travels along the endwall resulting in a slight upwash of fluid near the vane-endwall junction but no permanent vortex.
Comparison of Figures 5.6a and 5.8a show that the size of the vortex for the 90° orientation is much smaller for the MFR = 0.7% case. Subsequently, the maximum vorticity magnitude is larger. The axial position of the vortex center is approximately the same between the two MFR cases at $x/C_{ax} = -0.11$, but the spanwise position decreases from $z/0.5S = 0.035$ to 0.03 with a reduction in MFR from 1.0% to 0.7%. For the 45° slot, a reduction in MFR results in a weaker upwash of fluid near the vane-endwall junction as seen in a comparison of Figures 5.6b and 5.8b. Subsequently, the severity of streamline curvature near the vane stagnation is smaller resulting in the formation of weaker counter-rotating vortices.

To further explore the effects of reduced MFR, Figure 5.9 presents the time-averaged flowfield results for the 90° slot with MFR = 0.3% and $I = 2.8$. Comparison of Figure 5.6a, 5.8a, and 5.9 show that reducing MFR results in a consistent reduction in the size of

![Figure 5.9. Average flowfield vectors, streamlines, and contours of vorticity in the stagnation plane with MFR = 0.3% and $I = 2.8$ for the 90° orientation at $x/C_{ax} = -0.17$.](image)
the leading edge vortex. Although the axial position remains at approximately \( \frac{x}{C_{ax}} = -0.11 \) for all three MFR cases, the spanwise position of the vortex center is moved closer to the endwall with decreasing MFR. At MFR = 0.3\% the vortex is centered at approximately \( \frac{z}{0.5S} = 0.025 \), as compared to \( \frac{z}{0.5S} = 0.035 \) and 0.03 for MFR = 1.0\% and 0.7\% respectively.

The improvement in endwall cooling performance associated with the 45° slot orientation is an obvious result as there is less mixing between the coolant and hot mainstream flow as indicated in the flowfield results. What is not apparent is the qualitative improvement in endwall cooling effectiveness of the 45° slot with respect to the 90° orientation. To quantify the improvements associated with the 45° slot, area averages of the endwall effectiveness values are presented in Figure 5.10 for the contours presented in Figures 5.5. The area over which the averages were performed extended from the downstream edge of the leakage slot to the passage throat and across the passage pitch from one vane stagnation to the other. The averaging area is depicted in Figure 5a on the 1.0\% MFR contour. Ejecting coolant closer to the plane of the endwall results in a large increase in area averaged effectiveness. Figure 5.10 indicates a percent difference increase in area averaged effectiveness of approximately 177\% and 129\% for MFR’s of 0.7\% and 1.0\% respectively between the 90° and 45° slot orientations.

In addition to the cooling effectiveness improvements associated with the 45° slot, Figure 5.5 shows that there is little change in the overall range of effectiveness levels with a reduction in mass flow for both slot orientations. The coolant, however, is shown to penetrate progressively less into the vane passage with a reduction in mass flow. For
the 90° slot, local effectiveness levels downstream of the slot near the suction side leading edge are shown to increase slightly with reduced mass flow. As indicated in Figures 5.6a, 5.8a, and 5.9, the coolant penetrates less into the freestream with each subsequent reduction in MFR, improving the near slot cooling effectiveness. Local effectiveness levels near the downstream edge of the slot for the 45° orientation, however, show little change.

Figure 5.10 indicates that the area averaged effectiveness increases with mass flux ratio for both slot orientations. Qualitatively, the 45° slot realizes a percent difference increase in area averaged effectiveness of 7% with an increase in MFR from 0.7% to 1.0%. Although providing poorer cooling performance, the 90° slot orientation sees a
greater increase in area averaged effectiveness with increasing MFR. The 90° slot realizes a percent difference increase in area averaged effectiveness of 41% and 29% with an increase in MFR from 0.3% to 0.7% and 0.7% to 1.0% respectively.

5.3.2 Effects of Slot Position

The effects of the slot position on the cooling performance and stagnation plane flowfield were studied by moving the 90° slot further upstream and downstream from the nominal location to $x/C_{ax} = -0.34$ and -0.05 respectively. Figure 5.11 compares adiabatic effectiveness contours between the three upstream locations for the 90° slot with MFR = 1.0% and $I = 2.8$.

![Figure 5.11. Comparison of adiabatic effectiveness contours for the 90° slot orientation at three different locations for MFR = 1.0% and $I = 2.8$.](image)
Figure 5.11 shows that there is little difference in the overall range in effectiveness level between the slot at the nominal location and the upstream location. For the downstream location, however, an improvement in the local cooling performance of the leakage slot is observed. The relatively close proximity of the leakage slot to the vane stagnation makes it likely that some coolant is washed down the surface of the vane and onto the endwall improving the effectiveness near the suction side leading edge. To appreciate why these differences in cooling performance occur, Figure 5.12 presents time-averaged flowfield results for the $90^\circ$ slot at the further upstream and downstream locations with MFR = 1.0% and $I = 2.8$.

At a location $x/C_{ax} = -0.34$ upstream of the vane stagnation, Figure 5.12a indicates that the high momentum leakage flow lifts off the incoming boundary layer and forms a permanent vortex structure. In addition, the strong back flow underneath the vortex results in a small counter clockwise recirculation zone between the ejecting coolant and main vortex at $x/C_{ax} = -0.029$. Comparisons of the leading edge vortex at a slot location of $x/C_{ax} = -0.17$ and -0.34 indicate a similar yet stretched vortex structure as shown in Figures 5.6a and 5.12a. The further upstream slot location is subject to a weaker spanwise pressure gradient allowing leakage flow to travel further downstream before being turned toward the endwall and into the leading edge vortex. The extended penetration of leakage flow downstream results in a lengthened vortex structure and subsequently a weaker core vorticity magnitude. Figures 5.6a and 5.12a show that the vortex center is located approximately $0.12C_{ax}$ and $0.06C_{ax}$ from the downstream edge of the leakage slot at the further upstream and nominal location respectively. The larger distance between the vortex structure and the ejecting coolant allows room for the small
Figure 5.12. Average flowfield vectors, streamlines, and contours of vorticity in the stagnation plane with MFR = 1.0% and I = 2.8 at a slot location of (a) $x/C_{ax} = -0.34$ and (b) $x/C_{ax} = -0.05$.

Recirculation zone mentioned previously to form.

Most important to the endwall effectiveness level, however, is the penetration depth of the leakage coolant into the mainstream. The penetration depth determines the amount of contact between the coolant and mainstream and subsequently how effective
the leakage flow will be in cooling the endwall. Figures 5.6a and 5.12a show similar penetration depths of the leakage coolant to a spanwise location of approximately $z/0.5S = 0.09$. Also note that the spanwise location of the vortex center is similar between the nominal and further upstream slot locations at $z/0.5S = 0.035$. The similarity between the vortex structures at the further upstream and nominal location correspond to the similarity between endwall effectiveness values.

At the further downstream location, $x/C_{ax} = -0.05$, the ejecting coolant has a high enough momentum to cause the separation of the approaching boundary layer as shown in Figure 5.12b. Unlike the other two upstream locations, however, a very strong spanwise pressure gradient and a large exit static pressure greatly reduce the penetration of the leakage flow into the mainstream. The ejecting coolant is quickly turned toward the endwall to form a small vortex with a relatively high vorticity magnitude compared to that seen at the other two locations. The leakage coolant interacts less with the hot mainstream flow, improving cooling effectiveness directly downstream of the slot and near the suction side leading edge. As shown in Figure 5.12b, the vortex center is located at approximately $z/0.5S = 0.01$ as compared to $z/0.5S = 0.035$ for the further upstream locations. Figure 5.12b also indicates the presence of a small tertiary vortex forming upstream of the ejecting coolant similar to that seen previously when the slot was at the nominal location.

Similar to the area averages performed to assess the performance of slot orientation on the endwall cooling performance, similar area averages were taken over the endwall to quantify the effects of slot location. Figure 5.13 presents the area averaged effectiveness
values for the three slot locations shown in Figure 5.11. Note that the averaging area for the further upstream and downstream locations began directly downstream of the leakage slot across one passage pitch, extending to the passage throat as indicated in Figure 5.5a. There is only a small percent difference increase in area averaged effectiveness of 4% between the nominal and further upstream locations. At the closer location, however, a percent difference increase of approximately 30% is realized relative to the nominal location.
5.4 Conclusions

Adiabatic effectiveness measurements on the flat endwall of a vane cascade and stagnation plane flowfield measurements were presented to study the leakage flow from a two-dimensional upstream slot simulating the combustor-turbine interface gap. Results for two different slot orientations of 90° and 45° were presented at a nominal upstream location. In addition, results were presented for the 90° slot orientation at two other upstream locations, one further upstream and one further downstream from the nominal location. Several leakage mass flux ratios were considered while maintaining a constant momentum flux ratio. The orientation and position of the combustor-turbine interface gap was shown to have a significant impact on endwall cooling performance and stagnation plane flowfields.

At the nominal upstream location, the 45° slot orientation provided a substantial improvement in the endwall cooling performance over the 90° orientation. Effectiveness contours indicated a significant improvement in effectiveness levels at every location within the vane passage. Area averages of effectiveness over a single vane passage showed an improvement of well over 100% for the 45° slot. The stagnation plane flowfields also showed vastly differing flowfields between the 90° and 45° orientations. Ejecting coolant at 90°, perpendicular to the endwall, resulted in the separation of the incoming boundary layer and the subsequent formation of a large leading edge vortex. Coolant ejected from the slot was turned back down toward the endwall and underneath the vortex by the inherent pressure gradient resulting from the stagnation of the incoming boundary. Time-resolved measurements indicated that there were some small unsteady fluctuations in the position of the vortex center, but the ejecting coolant served to
maintain the vortex as the structure never waned. For the 45° orientation, however, no permanent vortex was present. Instead, an upwash of coolant along the vane surface was observed at the vane-endwall junction.

Results for both the 90° and 45° slot at the nominal location indicated little change in the overall range of measured effectiveness levels with a reduction in mass flux ratio. Reducing the mass flux ratio reduced the penetration of the ejecting coolant into the vane passage effectively lowering the local effectiveness values near the pressure side and toward the passage throat. For the 90° orientation, the size of the leakage flow induced vortex was reduced with MFR but the axial location of the vortex core was maintained. Flowfield results for the 45° slot showed a weaker upwash of fluid along the vane stagnation with a reduction in MFR.

Moving the 90° slot also had an overall effect on the cooling. While the most upstream position was shown to have little impact on the resulting endwall cooling performance, the position closest to the vane showed an influence. Moving the 90° slot closer to the vane stagnation was shown to improve local and area averaged effectiveness. Unlike the two further upstream locations, the size of the induced vortex was greatly reduced while the vorticity intensity increased.
Chapter 6

Impact of the Combustor-Turbine Interface Slot Orientation on the Durability of a Nozzle Guide Vane Endwall [62*]

6.1 Introduction

The large range of temperatures experienced by the first stage of the turbine section requires that clearance slots be placed between the combustor and turbine section to allow for thermal expansion. To prevent potentially destructive hot gases from being ingested into these interface slots, the feed cavities are pressurized with relatively cool air bled from the compressor. Although generally overlooked in comparison to film cooling holes, leakage flow from the combustor-turbine interface slot can provide essential cooling to the endwalls of the first stage, nozzle guide vane. The ability of the leakage flow to reduce the heat load to the endwall is critical as the coolant is a parasitic drain on the compressor, possibly reducing the overall efficiency of the engine. A net benefit to engine efficiency can be gained, however, if the performance of the leakage flow in reducing the endwall heat load allows for increased turbine inlet temperatures.

One way to improve the performance of the leakage flow is through the modification of the interface slot orientation. This paper is the first to report the effects of interface slot orientation on the net heat flux reduction experienced by a low pressure, nozzle guide vane endwall in the presence of leakage flow. Thermal measurements are complemented with time-resolved flowfield measurements in the vane stagnation plane. The effects of

* In review
interface slot orientation are investigated for a high momentum flux ratio typical of that seen in an engine and a low momentum flux ratio as more commonly seen in the literature.

6.2 Experimental Methods

All thermal and velocity measurements were performed in a low speed, closed loop wind tunnel, depicted in Figure 6.1 and previously described by Thrift et al. [52]. Driving the flow through the wind tunnel was a 50hp fan with a 50Hz variable frequency drive. After passing through the fan the flow was turned 90° before encountering a finned-tubed heat exchanger used to remove the initial heat supplied to the flow by the fan.

Figure 6.1. Depiction of the low speed, closed loop wind tunnel.

After passing through the primary heat exchanger, the flow was turned by another 90° elbow before being split into three separate flow paths. A porous plate positioned over
the main flow path, diverted some flow to the two outer secondary flow paths to serve as leakage coolant. The flow in each of the secondary flow paths traveled through secondary finned-tube heat exchangers before being passed into respective plenums. The secondary heat exchangers provided additional cooling. Flow was drawn from the upper plenum and into the appropriate leakage coolant plenum on the attached test section using a 2 hp blower. The leakage plenum on the test section supplied coolant to the interface slot. During all experiments the secondary flow was cooled to approximately room temperature at 25°C.

The freestream temperature that was set depended on the type of thermal measurement being performed. Adiabatic effectiveness and heat transfer measurements were made under steady state conditions with a freestream temperature of approximately 50°C and 25°C respectively. For adiabatic effectiveness experiments, the heated freestream flow and the cooled leakage flow simulated the hot core flow and leakage coolant of the engine. Using a temperature difference of approximately 25°C reduced the measurement uncertainty associated with adiabatic effectiveness. For heat transfer experiments, the temperatures of the freestream and leakage coolant were matched to within 0.25°C to ensure that the measured endwall heat transfer was solely due to convection.

After the flow split, the mainstream flow continued down the center of the wind tunnel and through an electrical resistance heater bank. After the heater bank, the main flow passed through a series of fine wire screens and honeycomb panel used for turbulence dampening and flow straightening respectively. The mainstream flow was then passed into a contracted straight flow section through rounded inlets. At the exit of
the straight flow section was the experimental test section containing the linear vane cascade and measurement endwall.

The vane test section was a two-dimensional, linear vane cascade as illustrated in a schematic in Figure 6.2. The cascade consisted of two full nozzle guide vanes and a third partial vane connected to a flexible wall. The profile of the flexible wall was adjusted to maintain the desired pressure distribution along the three vanes. The vane design was a spanwise extrusion of a two-dimensional midspan vane geometry. The van-

Figure 6.2. Schematic of the linear vane cascade with the different leakage slot orientations and the TRDPIV setup.
-es were scaled up by a factor of 2.4 from engine size to achieve high measurement resolution. A description of the nozzle guide vane parameters and flow conditions is given in Table 6.1.

The boundary layer entering the cascade was measured at a location 4.25\(C_{ax}\) upstream of the vane stagnation by Thrift et al. [34] using the same inlet flow conditions in Table 6.1. Table 6.2 lists the turbulent inlet boundary layer parameters, which were maintained throughout this study. The measured turbulence intensity was lower than that typically found in an engine at 1.0%. The effect of turbulence, however, was not considered in this study as past studies by the authors have evaluated this effect [36].

<table>
<thead>
<tr>
<th>Table 6.1. Vane Geometry and Flow Conditions</th>
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<tr>
<td>Scaling factor</td>
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<tr>
<td>Scaled vane chord (C)</td>
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<tr>
<td>Axial chord/chord ((C_{ax}/C))</td>
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<tr>
<td>Pitch/chord (P/C)</td>
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<tr>
<td>Midspan/chord (0.5S/C)</td>
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<tr>
<td>Inlet/exit Reynolds number ((Re_{in}/Re_{exit}))</td>
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<tr>
<td>Inlet/exit angle ((\alpha_{in}/\alpha_{exit}))</td>
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<tr>
<td>Inlet/exit Mach number ((Ma_{in}/Ma_{exit}))</td>
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<tr>
<th>Table 6.2. Inlet Turbulent Boundary Layer Characteristics</th>
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<tr>
<td>Boundary layer thickness/midspan ((\delta/0.5S))</td>
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<tr>
<td>Displacement thickness/midspan ((\delta^*/0.5S))</td>
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<tr>
<td>Momentum thickness/midspan ((\theta/0.5S))</td>
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<tr>
<td>Shape factor ((\delta^*/\theta))</td>
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<tr>
<td>Momentum thickness Reynolds number ((Re_{\theta}))</td>
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Leakage flow from the combustor turbine interface was simulated using a two-dimensional slot placed on the bottom endwall, upstream of the vane stagnation as illustrated in Figure 6.2. To investigate the effects of slot orientation, experiments were
performed with orientations of $90^\circ$, $65^\circ$, $45^\circ$, and $30^\circ$. As indicated in Figure 6.2, each slot had a flow length-to-width of 11.7. Also note that the downstream edge of each slot was located at $x = -0.17C_{ax}$ from the vane stagnation.

The mainstream conditions were verified before performing every experiment. The inlet velocity across the range of the cascade was found to vary less than 5% from the pitch averaged mean for all experiments. Static pressure measurements were made around the circumference of each vane at midspan and compared with predictions from a computational study. The computational study was previously performed by Thrift et al. [35] for incompressible, viscous, low-speed conditions using FLUENT [37]. Similar to that shown previously by Thrift et al. [52], the measured and predicted pressure distributions agreed indicating that the inviscid flowfield around each vane was matched to the predicted curve. In addition, mainstream temperature measurements were made in the spanwise direction and were found to never vary more than $0.75^\circ C$ from the mainstream temperature average.

Two different leakage coolant momentum flux ratios were investigated for the present study. Based on a realistic engine pressure ratio between the leakage plenum and freestream, a momentum flux ratio of $I = 2.8$ was determined. In addition, a lower momentum flux ratio more typical of that seen in the literature of $I = 0.7$ was investigated. The average momentum flux ratio was set according to Equation 6.1 below.

$$I = \frac{MFR^2(S)^2}{w^2}$$

(6.1)
Equation 6.1 indicates that the leakage coolant was also characterized by the mass flux issuing from the interface slot. A mass flux ratio (MFR) was defined as the ratio of coolant issuing from the upstream slot and entering a single passage to the mainstream mass flow rate through a single passage. The total mass flux issuing from the upstream leakage slot was measured using a laminar flow element located within the supply pipe to the slot plenum. Based on engine realistic conditions, a mass flux ratio of MFR = 1.0% was chosen for the engine realistic momentum flux ratio at I = 2.8.

To achieve the necessary slot coolant exit velocity to maintain I = 2.8 and MFR = 1.0% required a very small slot width as unlike the engine, the wind tunnel operated with a density ratio of approximately 1. Using Equation 6.1, a leakage slot width of w = 3.3mm was determined for the I = 2.8 and MFR = 1.0% condition. Consequently, the low momentum flux ratio condition at I = 0.7 was achieved by lowering the mass flux ratio with the given slot width to MFR = 0.5%. Note that for a given momentum flux ratio the pressure ratio between the slot plenum and vane stagnation was held constant for each injection slot.

6.2.1 Adiabatic Effectiveness Measurements

Spatially resolved, adiabatic wall temperatures were obtained from infrared (IR) measurements of the endwall using an Inframetrics P20 IR camera. To achieve an adiabatic condition, the endwall was made of a 2.54cm thick plate of low density closed cell polyurethane foam, which has a very low thermal conductivity (0.03 W/mK). The foam endwall was painted black to maintain a high emissivity on the endwall surface thus providing good resolution of the surface temperatures.
The test section ceiling contained 14 viewing ports, distributed across both vane passages to allow unobstructed access of the IR camera to the entire bottom endwall. At 55cm from the bottom endwall and with an IR camera resolution of 320 X 240 pixels, the resulting spatial resolution at the endwall was 0.75mm. Used for calibration, 19 type-E thermocouples were placed in the endwall surface. At least two thermocouples were present in each IR image and each image shared at least one common thermocouple with each neighboring image. A set of five images were taken at each of the 14 different viewing locations during an experiment. The surface images were then post-calibrated by determining the emissivity and background temperature of the image through matching of the image temperatures with the acquired thermocouple measurements. In general, the thermocouple and calibrated images agreed within 0.5°C. After calibration, the images were averaged at each location and then exported to an in-house MATLAB program. The program was used to assemble the individual images into a single map of the entire endwall based on the location of mutual thermocouples.

Although the thermal conductivity of the foam endwall was low, it was necessary that a small conduction correction still be used. A one-dimensional conduction correction as described by Ethridge et al. [44] was applied to all adiabatic effectiveness measurements according to the definition of $\eta$ in the nomenclature. The correction involved measuring the endwall surface effectiveness with no leakage slot and the freestream at test condition temperature. A correction value of $\eta_o = 0.1$ was measured within the vane passages. Upstream of the leakage slot, however, a correction value of $\eta_o = 0.15$ was measured because the upstream of the leakage slot was made of medium density fiberboard with a thermal conductivity (0.13 W/m-K).
6.2.2 Heat Transfer Measurements

Heat transfer experiments were performed with the same methodology as adiabatic effective experiments except for two major differences. First, a constant heat flux surface was provided on the endwall in the form of a thin endwall heater. The heater was permanently bonded to a foam surface to minimize conduction losses to the external environment. The heater consisted of a serpentine inconel pattern, encapsulated in Kapton, with a thin copper layer on the flow side. Like the adiabatic surface, the heater surface was painted black to maintain a high emissivity. Second, the mainstream and coolant temperatures were both held at room temperature to ensure that the heat transfer on the surface was solely due to convection.

For all heat transfer experiments, the lowest temperature difference between the endwall thermocouples and the freestream was approximately 10°C to minimize uncertainty in the IR measurement. Once the endwall temperatures reached a steady state value, the thermocouple data was recorded and IR images were captured according to the methodology discussed in the previous section. After an experiment, the images were calibrated and assembled into a single contour of endwall temperatures as before. As with adiabatic effectiveness tests, the thermocouple and calibrated images agreed within 0.5°C. A small percentage of the heat was lost to the surroundings through both radiation and conduction. The percentage of the radiation heat flux loss was typically between 3-16% of the total supplied heat flux. The percentage of conduction heat flux loss was much lower in the range of 0.5-1%.

Using the measurements of adiabatic effectiveness and heat transfer a net heat flux reduction (NHFR) was calculated. The net heat flux reduction makes use of the adiabatic
effectiveness and a non-dimensional metal temperature, $\phi$. As indicated in the nomenclature, the non-dimensional metal temperature accounts for the internal cooling taking place on the non-flow side of the endwall. A typical value for $\phi$ in a turbine engine is $\phi = 0.6$ [63]. A value of $\phi = 0.6$ was assumed for the calculation of all NHFR in the current study.

### 6.2.3 Particle Image Velocimetry Measurements

Flowfield measurements were performed in the stagnation plane of the center vane with a high-image-density, time resolved, digital particle image velocimetry (TRDPIV) system as illustrated in Figure 6.2. To capture instantaneous flow structures, TRDPIV illuminates and tracks very small particles which follow the flowfield. For the current study, Di-ethylhexyl Sebacat (DEHS) seeder particles with an approximate mean diameter of 1µm were introduced into the flow directly upstream of the wind tunnel blower using a Laskin nozzle. A dual cavity 15W Nd:YAG laser capable of firing at 10kHz per laser cavity was used to illuminated the tracer particles.

As shown in Figure 6.2 the laser was located over the center vane. The attached optics spread the laser light into a sheet and redirected it perpendicular to the endwall and along the vane stagnation line. The illuminated particles were captured using a CMOS camera capable of recording digital images at 2kHz with a spatial resolution of 1024x1024 pixels. The time delay between laser pulses was adjusted for each experiment to obtain a bulk particle displacement of approximately 8 pixels. At a camera sampling frequency of 1kHz, 3000 images were captured over 3 seconds for each experiment. The sampling frequency and sampling duration were chosen based on the characteristic fluid
time scale, $\tau_f$. As flowfield measurements of the HSV were being made, the characteristic fluid time scale was based on the boundary layer thickness and the freestream velocity as defined in the nomenclature. A sampling frequency and sampling duration of 1000Hz and 3 seconds corresponded to $12\tau_f$ and $250\tau_f$ respectively ensuring that the flow features would be temporally resolved. Time-averaged flowfields were calculated using all 3000 image pairs.

Captured images were processed using commercially available software [58]. A minimum pixel intensity background subtraction was first performed on the raw images to improve the signal-to-noise ratio. As indicated in Figure 6.2 the camera was at a slight angle to the stagnation plane, typically less than 8°. Using the software, the adjusted raw images were also corrected for the small off normal viewing angle resulting in less than a 1% correction in the axial velocities. After correction, the images were masked to remove the endwall and vane boundaries, replacing data outside the masked region with zero intensity. The images were then processed using a decreasing, multi-pass technique to determine the displacement vectors.

The multi-pass scheme employed a single pass at an interrogation window size of 32x32 pixels followed by two passes at an interrogation window size of 16x16 pixels with 50% overlap resulting in a final vector spacing of 8x8 pixels. Interrogation windows that contained at least 30% masked out area were discarded on the initial pass. For all subsequent passes an interrogation window was only discarded if at least 60% of the window was masked. The displacement vectors were calculated for each interrogation window amongst image pairs using a standard cross-correlation, cyclic FFT-based algorithm. To improve accuracy, a fractional window offset and image mapping through
bilinear interpolation was applied to each interrogation window during intermediate and final passes based on the calculated vector field in the previous pass. Vector validation was performed after each pass using a 4-pass regional median filter with adjustable criteria for the removal and re-insertion of possible spurious vectors. The vector validation scheme was visually checked for several representative instantaneous flowfields to ensure that only spurious vectors were removed.

6.2.4 Uncertainty Analysis

An uncertainty analysis was performed on the parameters of adiabatic effectiveness, heat transfer, and net heat flux reduction using the partial derivative method described by Moffat [45] and is presented in Table 6.3. For the TRDPIV velocity measurements, the uncertainty was calculated based on the average particle displacement, maximum displacement gradient, average particle image density, and signal-to-noise ratio [57, 64].

Table 6.3. Total Measurement Uncertainties

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adiabatic Effectiveness, $\eta$</td>
<td>$\delta\eta = \pm 0.025$ for $\eta = 0.03$ to 0.9</td>
</tr>
<tr>
<td>Heat Transfer, $Nu$</td>
<td>$\delta Nu = \pm 6 \ (2.5%)$ at $Nu = 250$</td>
</tr>
<tr>
<td></td>
<td>$\delta Nu = \pm 96 \ (8%)$ at $Nu = 1200$</td>
</tr>
<tr>
<td>Net Heat Flux Reduction, $NHFR$</td>
<td>$\delta NHFR = \pm 0.08 \ (5%)$ at $NHFR = 1.6$</td>
</tr>
<tr>
<td></td>
<td>$\delta NHFR = \pm 0.06 \ (20%)$ at $NHFR = 0.3$</td>
</tr>
<tr>
<td>Particle Image Velocimetry, $U,W$</td>
<td>$\delta U = 0.16 m/s (2% \ of \ U_\infty)$ for a bulk particle displacement of 8 pixels</td>
</tr>
<tr>
<td>Turbulence Intensity, $Tu$</td>
<td>$\delta Tu = \pm 0.02$ for $Tu = 0.1$ to 0.4</td>
</tr>
</tbody>
</table>

6.3 High Momentum Flux Ratio Results

Fully understanding the impact of interface slot orientation on the overall cooling of the vane endwall requires investigating the reduction in heat flux provided by the injected
coolant. As mentioned previously, the net heat flux reduction can be calculated from the adiabatic effectiveness and heat transfer. Figure 6.3 compares contours of adiabatic effectiveness for the four interface slot orientations. Although limited, injecting coolant at 90° with I = 2.8 provides some cooling to the entire endwall. An increase in the effectiveness over the entire endwall for 90° injection indicates that the coolant is separating and mixing with the near wall flow, creating a relatively cool layer of fluid over the entire endwall. Unfortunately the coolant is degraded through mixing with the hot mainstream flow, leading to relatively low adiabatic effectiveness levels. For the 90° slot, the effectiveness of the leakage coolant is highest near the suction side leading edge. This

![Figure 6.3. Comparison of adiabatic effectiveness contours between four different slot orientations for MFR = 1.0% and I = 2.8.](image)

-veness over the entire endwall for 90° injection indicates that the coolant is separating and mixing with the near wall flow, creating a relatively cool layer of fluid over the entire endwall. Unfortunately the coolant is degraded through mixing with the hot mainstream flow, leading to relatively low adiabatic effectiveness levels. For the 90° slot, the effectiveness of the leakage coolant is highest near the suction side leading edge. This
can be attributed to the relatively low static pressure on the endwall near the suction side leading edge, increasing the local momentum and mass flux ratios.

Reducing the slot orientation angle to 65° provides a substantial improvement to the cooling effectiveness in comparison to the 90° slot. The coolant injected at 65° has a smaller spanwise velocity component than the 90° slot, allowing the coolant to stay closer to the endwall and improve effectiveness. Near the pressure side of the passage where the cooling effectiveness is lowest, however, the 65° slot provides slightly lower effectiveness than that seen for the 90° slot. As to be discussed in more detail later, injecting coolant at 90° and 65° results in the formation of a large leading edge vortex in the stagnation plane. The presence of a leading edge vortex for coolant injected at 90° and 65° ensures that a passage vortex exists for each case. The adiabatic effectiveness results for 90° injection suggest that the coolant was able to successfully mix with the mainstream flow to such a spanwise distance as to overcome the passage vortex, allowing for some cooling at the pressure side of the passage. For the 65° slot, however, penetration of the coolant was reduced thus preventing the coolant from overcoming the passage vortex and cooling the pressure side of the passage.

Although a further reduction in the slot orientation angle to 45° does not provide an increase in the maximum effectiveness level, there is a substantial improvement in the overall spreading of high effectiveness levels in the passage. Specifically, effectiveness levels are increased at stagnation and near the pressure side of the passage, all the way to the trailing edge. The addition of a much stronger axial velocity component when the slot orientation is reduced to 45° allows the coolant to penetrate further into passage. A reduction in the slot orientation to 30° results in a less significant change to the passage
cooling effectiveness than that observed for the reduction from 65° to 45°. The injection of coolant at 30°, however, provides a more uniform distribution of effectiveness directly downstream of the slot. Consequently, a shifting of higher effectiveness values from the suction side leading edge to the vane stagnation results in a slight reduction of cooling effectiveness near the pressure side at the trailing edge in comparison to the 45° slot.

In either case, injecting coolant at 45° and 30° allows for improved adiabatic effectiveness levels near the pressure side of the passage in comparison to the 90° and 65° slots. To be discussed in more detail in the following paragraphs, no time-averaged vortex is formed in the stagnation plane for coolant injected at 45° and 30°. In combination with the improved passage side cooling effectiveness, the removal of the leading edge vortex with 45° and 30° injection suggests that the passage vortex is diminished. In addition to the improved passage side effectiveness, the strong sweeping of the effectiveness levels from pressure to suction side is reduced with 45° and 30° injection as a result of some mitigation of the endwall crossflow. Endwall crossflow is the result of the velocity deficit within the boundary layer, forcing endwall streamlines to follow a smaller radius of curvature through the passage than the freestream to maintain the balance between centripetal acceleration and the passage pressure gradient [5]. Injection along the endwall reduces crossflow by reducing the velocity deficit at the near wall, particularly just downstream of the injection slot.

Figure 6.4 compares contours of heat transfer in the form of Nusselt number for the four leakage slot orientations. Each slot orientation case indicates relatively low heat transfer in the endwall region near the vane pressure side. For the 90° and 65° slots, the sweeping of the low heat transfer region toward the passage throat as high heat transfer
levels approach near the midpitch of the passage supports the existence of a strong passage vortex as suggested previously from the cooling effectiveness results. Similarly,

![Nusselt number contours between four different slot orientations](image)

**Figure 6.4.** Comparison of Nusselt number contours between four different slot orientations for MFR = 1.0% and I = 2.8.

the reduction in size and skew of the low heat transfer region for the 45° and 30° slots supports a mitigation of the passage vortex and endwall crossflow. Beyond the passage throat and along the vane suction side, however, heat transfer values are higher for the 30° and 45° slot orientations in comparison to the 90° and 65° slot orientations.

As shown in the adiabatic effectiveness results, the coolant is concentrated near the endwall just downstream of the passage throat for the 45° and 30° slots indicating little mixing with the hot mainstream flow up to that point. Beyond the passage throat,
however, the effectiveness is reduced corresponding to the increase in heat transfer. The sudden reduction in effectiveness and the increase in heat transfer around the suction side indicate an increase in the near wall mixing. For the 45° and 30° slots this suggests an intensification of the passage vortex beyond the point of attachment which is typically considered to be the minimum pressure point at the passage throat.

Concerning the heat transfer near the leakage slot, the majority of the coolant for the 90° and 65° slots is concentrated near the suction side leading edge. Figure 6.4 indicates that the concentration of coolant near the suction side leading edge results in relatively high heat transfer in the region, particularly for the 65° slot where cooling effectiveness is higher. Contour results for the 45° and 30° slot orientations show a more pitchwise uniform distribution of Nusselt number downstream of the slot similar to that seen in the effectiveness results. Injecting coolant more tangential to the endwall rather than normal to the endwall where separation is likely increases the interaction of the coolant with the near wall flow at the exit of slot, providing a greater potential for heat transfer augmentation. As shown in Figure 6.4, the Nusselt number values downstream of the slot near the midpitch for the 30° and 45° slot orientations are almost twice as high as those seen for the 90° and 65° slots.

To support the observations made from the adiabatic effectiveness and heat transfer contours, Figure 6.5 compares the time-averaged flowfields with overlaid mean streamlines and contours of turbulence intensity in the stagnation plane for the four leakage slot orientations. The stagnation of the impending boundary layer at the vane leading edge results in a pressure gradient toward the endwall as a consequence of the velocity deficit within the boundary layer. Streamlines in Figure 6.5a indicate that the se-
Figure 6.5. Average flowfield vectors, streamlines, and contours of turbulence intensity in the stagnation plane with MFR = 1.0% and I = 2.8 for the (a) 90° (b) 65° (c) 45° and (d) 30° slot orientations.

-eparated coolant injected at 90° is turned back to the endwall to form a leading edge vortex. The time-averaged flowfield indicates that the coolant injected at 90° promotes the large leading edge vortex. The presence of the vortex and the associated mixing as indicated by increased turbulence levels results in high heat transfer at the vane stagnation as shown in Figure 6.4a and poor adiabatic effectiveness as shown in Figure 6.3a. Also note that a small time-averaged, secondary vortex forms upstream of the
injecting coolant which acts as a physical impedance to the approaching boundary layer flow similar to the vane stagnation.

Like the 90° slot orientation, coolant injected at 65° also separates from the endwall resulting in the formation of a time-averaged vortex as shown in Figure 6.5b. With the addition of an axial injection component and a reduced spanwise component, however, the large time-averaged vortex forms further downstream and closer to the endwall. As shown in Figure 6.3b, reducing the penetration depth of the coolant improves the stagnation region adiabatic effectiveness relative to the 90° slot. The high turbulence levels for coolant injected at 65° in comparison to the 90° slot indicate the influence of the approaching flow on the injecting coolant. Coolant injected at 90° acts as impedance to the incoming flow resulting in the formation of a secondary vortex. For the 65° slot, however, the incoming flow was able to frequently disrupt the injecting coolant. As a result of the frequent disruption the vortex formed with 65° injection is less steady than that observed for 90° injection, resulting in increased turbulence. Consequently, endwall heat transfer is also increased in the stagnation region as shown in a comparison of Figures 6.4a and 6.4b. While the approaching flow does turn toward the endwall just upstream of the injecting coolant, the coolant jet was not a significant enough obstruction to result in the regular formation of a secondary vortex, thus no time-averaged secondary structure is present for 65° injection.

Figures 6.5c and 6.5d show that no time-averaged vortex is present in the vane stagnation plane for the 45° and 30° slot orientations. For both the 45° and 30° slot orientations, the time-averaged flowfield indicates that the coolant is injected tangential to the endwall. The injection of high velocity coolant along the endwall plane energizes
the velocity deficit in the approaching boundary layer at the near wall. The stagnation of this boundary layer at the vane results in a static pressure gradient at the vane-endwall junction that is away from the endwall. Figures 6.5c and 6.5d show a strong turning of the flow toward midspan at the vane-endwall junction for both the 30° and 45° slot orientations. As shown in Figure 6.4, the heat transfer close to the vane-endwall junction in the stagnation region is higher for the 90° and 65° slots than the 45° and 30° slots respectively. The increased heat transfer near the vane stagnation in the case of the 90° and 65° slots is a result of downwash impinging on the endwall at the junction of the vane. Downwash at the vane-endwall junction at stagnation, however, is not present for the 45° and 30° slots.

The injection of coolant along the endwall, however, does not completely fill in the velocity deficit region of the approaching boundary layer for the 45° and 30° slots. Coolant injection at 45° and 30° results in the formation of a shear layer along the interface between the high momentum coolant and the low velocity boundary layer flow. Large gradients exist in the streamwise velocity along the interface line of the shear layer causing an increase in turbulence level. The streamwise velocity gradient is stronger for coolant injected at 30° in comparison to the 45° slot because of the larger streamwise injection component, thus the turbulence levels are higher.

Streamlines in Figures 6.5c and 6.5d indicate that the mainstream flow turns toward the endwall on approach to the vane stagnation. The interaction between the opposing flows of the coolant and mainstream at stagnation results in strong inflection points in the time-averaged streamlines. The inflection points indicate where the spanwise direction of the velocity is reversed, setting up a scenario where a counter-clockwise rotating vortex
can form. For the 45° slot, intermittent vortices are formed along the inflection points but are quickly dissipated or moved out of the stagnation plane as no vortex structure is present in the time-averaged flowfield. Stronger upwash in the case of the 30° slot, however, results in the regular formation of a small counter-clockwise rotating vortex centered at z/S = 0.09 at the vane stagnation.

Adiabatic effectiveness and heat transfer measurements with an assumed non-dimensional metal temperature can be combined to produce a net heat flux reduction to the endwall. Figure 6.6 compares contours of net heat flux reduction for the four slot orientations. The injection of coolant for each slot orientation provides a reduction in heat flux.

**Figure 6.6.** Comparison of net heat flux reduction contours between four different slot orientations for MFR = 1.0% and I = 2.8.
flux at every location on the endwall. A large net heat flux reduction is achieved by providing a low fluid temperature at the near wall in the presence of high heat transfer. The heat transfer contours in Figure 4 are similar over much of the passage, excluding the region just downstream of the slot and past the passage throat. Therefore, the distribution of net heat flux reduction is most analogous to the adiabatic effectiveness distributions where differences were larger among the different slot orientations. In the case of the 90° slot, the maximum net heat flux reduction occurs near the suction side leading edge where heat transfer and effectiveness was highest. Although reducing the slot angle to 65° provided only a marginal change in the endwall heat transfer, the substantial improvement in effectiveness resulted in an equally significant reduction in the net heat load experienced by the endwall. A further reduction in the injection slot angle to 45° does not result in as substantial of an increase in the maximum net heat flux reduction level as that seen with the injection slot orientation change from 90° to 65°. Injecting coolant at 45°, however, does greatly improve the net heat flux reduction in the stagnation region and near the pressure side of the passage all the way to the passage throat in comparison to the 90° to 65° slots. Only a marginal change in the net heat flux reduction is observed with a reduction in slot orientation from 45° to 30°, indicating that the benefit of lowering the slot injection orientation has reached a plateau.

To quantify the improvement in heat flux associated with injecting coolant for each slot orientation, area averages of net heat flux reduction are presented in Figure 6.7 for the contours presented in Figure 6.6. The area over which the averages were performed extended from the downstream edge of the leakage slot to the passage throat and across the passage pitch from one vane stagnation to the other. The averaging area is depicted in
Figure 6.7. Area averaged net heat flux reduction between four different slot orientations.

Figure 6.6 on the 90° slot orientation contour. Figure 6.7 indicates that injecting coolant at 90°, 65°, 45°, and 30° results in an area average net heat flux reduction of 36%, 75%, 137%, and 131% respectively.

6.4 Low Momentum Flux Ratio Results

In addition to high momentum flux ratio experiments, results were also obtained for the injection of coolant at a lower momentum flux ratio. As mentioned previously, low momentum flux ratio experiments were performed by reducing the mass flux ratio to MFR = 0.5% thus reducing the momentum flux ratio for the fixed slot width to I = 0.7. Figure 6.8 compares contours of adiabatic effectiveness for the 90°, 65°, 45°, and 30° slot orientations for the low momentum flux ratio condition. Each slot orientation indicates a
strong sweeping of coolant from the pressure side to the suction side of the passage, highlighting the strong influence of the passage vortex and endwall crossflow. Unlike the $I = 2.8$ results, injecting coolant at $I = 0.7$ is insufficient in providing cooling to the endwall near the vane pressure side for each slot orientation.

![Image of adiabatic effectiveness contours]

**Figure 6.8.** Comparison of adiabatic effectiveness contours between four different slot orientations for MFR = 0.5% and $I = 0.7$.

For the $90^\circ$ slot the majority of the coolant is again concentrated near the suction side leading edge. Figure 6.8 shows that the local cooling effectiveness at the suction side leading edge is higher for the $I = 0.7$ case than that seen previously for the $90^\circ$ slot with $I = 2.8$. Reducing the momentum flux ratio reduces the penetration depth of the coolant in
to the mainstream, improving the effectiveness of the coolant near the suction side leading edge. As the local effectiveness is improved for injection at 90° with I = 0.7, reducing the slot orientation to 65° results in only a small improvement in the adiabatic effectiveness in comparison to the same orientation reduction at I = 2.8. A similar maximum cooling effectiveness level was observed for the injection of coolant at 65° for both the high and low momentum flux ratios. The distribution of the maximum effectiveness level, however, covered a larger region of the endwall just downstream of the slot near the suction side leading edge for injection at 65° with I = 0.7. Similar to the 90° slot, reducing the penetration depth of the coolant improves the cooling effectiveness just downstream of the slot while the cooling effectiveness near the pressure side of the passage suffers. Reducing the slot orientation to 45° results in improved coolant coverage in comparison to the 90° and 65° slot orientations for low momentum injection. The overall change in effectiveness, however, is much less significant for the reduction from 65° to 45° for I = 0.7 in comparison to high momentum injection. Similar to the high momentum results, a further reduction in slot orientation to 30° results in only a marginal change in coolant coverage.

The overall influence of slot orientation on effectiveness is less significant at I = 0.7 than what was observed for I = 2.8. Figure 6.9 indicates a similar development for heat transfer in a comparison of Nusselt number contours for the four leakage slot orientations with I = 0.7. The low heat transfer region within the vane passage near the pressure side is shown to be similar in both range and distribution between the four slot orientations. The heat transfer in this region is similar because the injected coolant was unable to reach the endwall near the passage pressure side for each slot orientation. In comparison to the
heat transfer results presented in Figure 6.4 for the I = 2.8 condition, the minimum Nusselt number value within the passage for the low momentum flux ratio cases is shown to be approximately 10% lower. The sweeping of the low heat transfer region from pressure to suction side for each slot orientation also correlates with the sweeping of effe-

![Nusselt number contours for different slot orientations](image)

**Figure 6.9.** Comparison of Nusselt number contours between three different slot orientations for MFR = 0.5% and I = 0.7.

-ctiveness levels indicating the strong influence of the passage vortex and endwall crossflow. Near the suction side leading edge the Nusselt number values are shown to skew toward higher values in the presence of leakage flow for each slot orientation. The distribution of peak Nusselt Number values near the suction side leading edge, however, is shown to diminish in area with a reduction in the slot orientation.
At the stagnation region, Figure 6.9 shows that the Nusselt number values are similar in range for all four slot orientations. For the 90° and 65° slots, however, the higher Nusselt number values are spread over a larger area. To support the stagnation region heat transfer observations, Figure 6.10 presents time-averaged flowfields at the stagnation plane for all four slot orientations with $I = 0.7$. As shown in Figure 6.10a, injecting coolant at 90° results in the formation of a time-averaged vortex. Similar to that observed for the $I = 2.8$ cases, coolant injected at 65° results in the formation of time-averaged vor-

**Figure 6.10.** Average flowfield vectors, streamlines, and contours of turbulence intensity in the stagnation plane with MFR = 0.5% and $I = 0.7$ for the (a) 90° (b) 65° (c) 45° and (d) 30° slot orientations.
-tex that is slightly further downstream as a result of a reduced spanwise injection component and increased axial component. For the 45° and 30° slot orientations, however, a much smaller vortex is formed at the vane-endwall junction. In addition, the coolant interaction with the near wall flow and subsequently the turbulence levels are reduced for each reduction in slot orientation. The disparity in size and turbulence intensity of the leading edge vortex formed between the 90° and 65° slots support the wider distribution of high heat transfer in the stagnation region in comparison to the 45° and 30° slots for low momentum injection.

Comparing to the flowfields shown previously in Figure 6.5 for I = 2.8, the leading edge vortex that forms for coolant injected at 90° is much larger and results in higher turbulence levels than that seen for the low momentum flux ratio condition. For the injection of coolant at 90°, the leading edge vortex that forms for each momentum flux ratio case is centered at \(x/C_{ax} = -0.11\). This is expected as each momentum flux ratio condition is under the influence of the same mainstream Reynolds number and boundary layer. The spanwise position of the vortex center, however, is higher for the I = 2.8 condition at \(z/S = 0.04\) as compared to \(z/S = 0.02\) for the low momentum flux ratio case. A higher injection velocity for the I = 2.8 case allows the coolant to penetrate further into the mainstream before being turned back to the endwall. The time-averaged vortex formed for coolant injected at I = 0.7 for the 65° slot also maintains the same axial location as the I = 2.8 case at \(x/C_{ax} = -0.1\). In contrast to the 90° slot, however, the vortex also maintains the same spanwise position at \(z/0.5S = 0.02\) for the 65° slot. Although the vortex location is similar between the high and low momentum flux ratio cases for the 65° slot, the turbulence intensity generated by the vortex is lower for the I = 0.7 case.
A comparison of the 45° and 30° slot orientation results between the two momentum flux ratio conditions in Figures 6.5 and 6.10 show a more significant augmentation of the leading edge flowfield than that observed for coolant injection at 90° and 65°. Coolant injected at 45° and 30° with I = 2.8, flows along the endwall and is turned up at the vane-endwall junction. In reducing the momentum flux ratio to I = 0.7, however, the velocity deficit within the approaching boundary layer is not sufficiently filled in to counteract the strong turning of flow toward the endwall. As a result, a small leading edged vortex is formed at the vane-endwall junction as shown in Figures 6.10c and 6.10d. Interestingly, the similarity in stagnation plane flowfield between the 45° and 30° slot orientations is reflected in the similarity in stagnation region heat transfer for both momentum flux ratio cases respectively. For the 90° and 65° slot orientations, only the I = 0.7 case shows strong similarities between the stagnation plane flowfield and heat transfer. At I = 2.8 the stagnation region heat transfer was higher for the 65° slot in comparison to the 90° slot, correspondingly the stagnation plane flowfields were also different with injection at 65° resulting in higher turbulence levels.

Injecting coolant at I = 0.7 results in weaker turbulence levels and lower heat transfer in the stagnation region for all four slot orientations in comparison to the I = 2.8 cases. For coolant injected at I = 2.8, the average coolant exit velocity is approximately 68% higher than the freestream velocity. Consequently, the large velocity difference between the injecting coolant and near wall flow leads to large separation regions for the 90° and 65° slots and strong shear layers for the 45° and 30° slots respectively. With low momentum injection, however, the average coolant exit velocity is much closer to the freestream velocity, approximately 17% lower. The smaller velocity disparity between
the injecting coolant and near wall flow for injection at $I = 0.7$ results in smaller separation regions and weaker shear layers, reducing the stagnation plane turbulence for each slot orientation in comparison to the respective $I = 2.8$ case. Expectantly, the stagnation heat transfer for coolant injection at $I = 0.7$ compared to $I = 2.8$ is lower for each slot orientation as shown previously in Figures 6.4 and 6.9.

Figure 6.11 compares contours of net heat flux reduction between the four slot orientations for $I = 0.7$. A comparison of Figure 6.11 to the effectiveness contours in Figure 8 shows that the net heat flux was only reduced where coolant was present. Subse-

![Figure 6.11. Comparison of Net Heat Flux Reduction contours between four different slot orientations for MFR = 0.5% and I = 0.7.](image)

-quently, the net heat flux was slightly increased on the pressure side of the passage for each slot orientation as no coolant was present. For the 90° slot, a reduction in
momentum flux ratio from \( I = 2.8 \) to \( I = 0.7 \) results in a slight increase in peak net heat flux reduction. This is a result of the effectiveness near the suction side leading edge being higher for the \( I = 0.7 \) condition when coolant is injected at \( 90^\circ \). For the \( 65^\circ, 45^\circ, \) and \( 30^\circ \) slot orientations, however, the peak net heat flux reduction is slightly lower between the \( I = 2.8 \) and \( I = 0.7 \) cases. Although the maximum effectiveness was similar between the high and low momentum flux cases for the \( 65^\circ, 45^\circ, \) and \( 30^\circ \) slots, the heat transfer was much lower for the \( I = 0.7 \) cases resulting in a reduction in peak net heat flux reduction.

The area averaged, net heat flux reduction for each slot orientation at \( I = 0.7 \) is plotted on Figure 6.7 for comparison to the previously presented results at \( I = 2.8 \). Figure 6.7 indicates that injecting coolant at \( 90^\circ, 65^\circ, 45^\circ, \) and \( 30^\circ \) results in an area average net heat flux reduction of 33\%, 54\%, 79\%, and 81\% respectively. Reducing the momentum flux ratio from \( I = 2.8 \) to \( I = 0.7 \) results in a percent difference reduction in area averaged net heat flux reduction of 8\%, 28\%, 42\% and 38\% for the \( 90^\circ, 65^\circ, 45^\circ, \) and \( 30^\circ \) slot orientations respectively. The average net heat flux reduction is much lower for the \( I = 0.7 \) results in comparison to the \( I = 2.8 \) results for several reasons. The first is related to the reduction in coolant mass flow from MFR = 1.0\%, to MFR = 0.5\% for the lower momentum flux ratio case. Recall that the momentum flux ratio was reduced by reducing the MFR of the coolant through the fixed slot width. As shown in previous studies [26-27, 52], the effect of lowering the coolant mass flux ratio is to reduce the spreading of high effectiveness values within the passage while only marginally altering the maximum effectiveness value. Second, the large velocity difference between the injecting coolant and the near wall flow results in higher turbulence levels for the \( I = 2.8 \) case in
comparison to injection at $I = 0.7$ where the velocity difference is smaller. As a result, endwall heat transfer is higher for high momentum injection in comparison to low momentum injection for each respective slot orientation. As discussed previously, large net heat flux reductions are achieved at locations with high effectiveness and heat transfer levels. Reduced effectiveness levels and lower heat transfer within the passage for injection at $I = 0.7$ compared to $I = 2.8$ combines to produce lower area averaged, net heat flux reduction.

In addition, reducing the slot orientation is shown to increase the average net heat flux reduction by a greater extent for the high momentum flux ratio condition. For coolant injected at $I = 0.7$, reducing the slot orientation results in a fairly linear increase in area averaged net heat flux down to a slot orientation of $45^\circ$, below which the results plateau. For high momentum injection, however, an exponential increase in area averaged net heat flux reduction is observed down to a slot orientation of $45^\circ$. The exponential increase in area averaged net heat flux reduction is owed to the improvement between the $90^\circ$ and $65^\circ$ slots and the $45^\circ$ and $30^\circ$ slots respectively for injection at $I = 2.8$ in comparison to the $I = 0.7$ case. As discussed previously, a fundamental difference in the endwall adiabatic effectiveness, heat transfer, and stagnation plane flowfields exists between the $90^\circ$ and $65^\circ$ slots and the $45^\circ$ and $30^\circ$ slots respectively at $I = 2.8$. For low momentum injection, however, thermal and velocity field measurements indicate fewer differences among the four slot orientations.
6.5 Conclusions

Thermal and velocity measurements were performed in a nozzle guide vane passage to investigate the cooling performance of a 90°, 65°, 45°, and 30° interface slot orientation. Two leakage flow conditions were considered, an engine realistic momentum ratio and a low momentum flux ratio.

Coolant injected at the engine realistic momentum flux ratio provided some cooling effectiveness to the entire endwall for each slot orientation. Systematically reducing the slot orientation was shown to improve cooling effectiveness and coverage uniformity, particularly near the pressure side of the passage and trailing edge down to a slot orientation of 45°. A final reduction in slot orientation to 30° did not provide any substantial improvement over coolant injected at 45°. The absence of a strong sweeping in the effectiveness values from the pressure side to the suction side of the passage for the 45° and 30° slots indicated that the passage vortex and endwall crossflow were diminished. Simultaneously lowering the momentum and mass flux ratio restricted the injected coolant to the suction side of the passage indicating the strong influence of secondary flows for each slot orientation.

Similar to the adiabatic effectiveness results, heat transfer patterns were shown to be characteristically different between the 90° and 65° slots and the corresponding 45° and 30° orientations at the engine realistic momentum flux ratio. For the 90° and 65° orientations, the highest heat transfer occurred in the stagnation region. A leading edge, time-averaged vortex was formed in the stagnation plane when coolant was injected at both 90° and 65°, considerably augmenting the local turbulence and endwall heat transfer. For coolant injected at 45° or 30° no large time-averaged vortex was present,
instead a strong upwash of fluid at the vane-endwall junction was observed. Injecting coolant along the plane of the endwall produced strong shear layers and subsequently high turbulence levels at the endwall directly downstream of the slot. As a result, very high heat transfer values were observed downstream of the interface slot along the entire passage pitch instead of just at the stagnation region. Reducing the momentum flux ratio reduced the velocity difference between the injecting coolant and near wall flow, substantially reducing the mixing and the resulting turbulence levels. Subsequently, endwall heat transfer levels were reduced for each slot orientation. Although weaker in intensity a time-averaged vortex was still formed in the stagnation plane for both the $90^\circ$ and $65^\circ$ slot orientations at low momentum injection. For the $45^\circ$ and $30^\circ$ slot orientations, the injection of low momentum coolant was insufficient in counteracting the strong turning of flow toward the endwall leading to the formation of a small vortex at the vane-endwall junction.

With full coolant coverage and high endwall heat transfer, injecting coolant at the engine realistic momentum flux ratio reduced the net heat flux to the endwall at each point within the vane passage for all four slot orientations. Reducing the interface slot orientation, however, significantly improved the overall reduction in net heat flux with area averaged, net heat flux reductions as high as 137%. In combination with lower endwall heat transfer owing to reduced turbulence levels, the absence of coolant near the pressure side of the passage resulted in lower area averaged net heat flux reduction for the reduced momentum and mass flux ratio case. While reducing the interface slot orientation for low momentum injection improved the net heat flux reduction, the highest area average was lower than for higher momentum injection at only 80%.
Although an angled slot requires more axial space than a perpendicular slot, the substantial improvement in endwall cooling provided by reducing the slot orientation may outweigh the possible small increase in engine length. The current paper has shown that injecting coolant at a high momentum flux ratio of $I = 2.8$ and MFR = 1.0% with an orientation of $45^\circ$ would provide the largest reduction in heat load to the vane endwall while minimizing the axial space required by the slot.
Chapter 7

Influence of Flow Injection Angle on a Leading-Edge Horseshoe Vortex [65*]

7.1 Introduction

The use of obstructions protruding from an endwall into a crossflow has applications ranging from internal cooling channels to external turbine airfoils. The particular shape of the obstruction can vary widely between different applications. A common geometry among many protruding features, however, is the junction of a rounded leading-edge obstruction as in the case of a pin fin within a channel or a nozzle guide vane in a gas turbine engine. As the approaching velocity boundary stagnates onto the obstruction, a leading-edge vortex is formed. The leading-edge vortex, commonly referred to as the horseshoe vortex (HSV) convects around the obstruction increasing the heat transfer on the endwall at the junction. While an increase in heat transfer is desired for a pin fin within a cooling channel, an increase in heat transfer for a gas turbine airfoil can be detrimental to the component life. In each case, however, the unsteady nature of the HSV and the subsequent propagation of the structure around the obstruction also lead to pressure losses.

One method of augmenting the heat transfer on the endwall in the junction region is by injection flow through the endwall upstream of the obstruction. In addition, flow injection augments the leading-edge flowfield and the subsequent formation of the HSV. An important parameter influencing the effects of the injecting flow is the angle of

* In review
injection relative to the endwall. Only a limited number of studies, however, have investigated the effects of injection angle on the heat transfer and flowfield within the junction.

This paper reports the effects of injection angle from a two-dimensional slot on the endwall heat transfer in the junction region of a rounded leading-edge vane. Heat transfer measurements are complemented with time-resolved flowfield measurements in the stagnation plane of the junction. The effects of injection angle are investigated for both high momentum flux and low momentum flux flows exiting the slot.

7.2 Experimental Facility

All experiments were conducted in a low-speed, closed-loop wind tunnel facility depicted in Figure 7.1. Driving the flow through the wind tunnel was a variable speed fan. Downstream of the fan, the flow was turned by a 90° elbow before being passed through the primary, finned-tube heat exchanger used to remove the heat supplied to the

Figure 7.1. Depiction of the low speed, closed loop wind tunnel.
flow by the fan. After being turned by another 90° elbow, the air was split into three flow paths.

The flow which passed through the upper and lower passages entered into respective plenums and could be used as injection fluid while flow passing through the center passage served as the mainstream flow. A fraction of the flow was drawn from the upper plenum only and into a plenum that supplied the injectant to the slot using a variable speed blower.

During all experiments the mainstream and injection flows were maintained at a constant temperature at 25°C to ensure that the heat transfer coefficients were reported for one driving temperature, wall minus freestream as indicated in the nomenclature. Downstream of the flow split, the mainstream passed through a series of screens used for flow straightening, and then into a contracted straight flow section with a rounded inlet. At the exit of the straight flow section was the experimental test section where all measurements were performed. The test section incorporated a linear vane cascade. Air exiting the test section was turned by a final 90° elbow before encountering the fan and completing the closed-loop.

The test section consisted of a two-dimensional, linear vane cascade as illustrated in an over-head schematic in Figure 7.2. The two passage cascade contained two full nozzle guide vanes and a third partial vane connected to a flexible wall. The vane design was a vertical extrusion of a two-dimensional midspan geometry. Two adjustable bleed boards were also positioned on either side of the two passage cascade. Note that flow passing by
Figure 7.2. Schematic of the linear vane cascade with the different injection slot angles and the TRDPIV setup.

The bleed board in the outer corner of the cascade was routed to the exit of the test section, downstream of the cascade to maintain the closed loop. A description of the vane parameters and the freestream flow conditions are given in Table 7.1. Note that the freestream turbulence intensity was maintained at 1.0% for this study. Table 7.1 lists an effective diameter of the vane leading-edge which was determined by matching the predicted inviscid approach velocity within the stagnation plane of the vane to the inviscid approach velocity within the stagnation plane of a circular cylinder [66].
Table 7.1. Vane Geometry and Flow Conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scaling factor</td>
<td>2.4</td>
</tr>
<tr>
<td>Effective leading-edge diameter (d_{eff})</td>
<td>100 cm</td>
</tr>
<tr>
<td>Scaled vane chord (C)</td>
<td>46 cm</td>
</tr>
<tr>
<td>Axial chord/chord (C_{ax}/C)</td>
<td>0.49</td>
</tr>
<tr>
<td>Pitch/chord (P/C)</td>
<td>1.01</td>
</tr>
<tr>
<td>Quarter-span/chord (0.25S/C)</td>
<td>0.3</td>
</tr>
<tr>
<td>Freestream turbulence intensity, T_{u_{∞}}</td>
<td>1.0%</td>
</tr>
<tr>
<td>Inlet/exit Reynolds number (Re_{in}/Re_{exit})</td>
<td>5.0 \times 10^4, 2.0 \times 10^5</td>
</tr>
<tr>
<td>Inlet/exit angle (α_{in}/α_{exit})</td>
<td>0°, 79°</td>
</tr>
<tr>
<td>Inlet/exit Mach number (Ma_{in}/Ma_{exit})</td>
<td>0.022, 0.12</td>
</tr>
</tbody>
</table>

The inlet velocity was measured 3.7d_{eff} upstream of the vane cascade and was found to vary less than 5% from the pitch averaged mean for all experiments. Static pressure measurement were made around the circumference of each vane at midspan and compared with predictions. As shown previously by Thrift et al. [34], the measured and predicted pressure distributions agreed indicating that the inviscid flowfield around each vane was matched to the predicted curve. The boundary layer entering the cascade was measured at a location 9.6d_{eff} upstream of the vane leading-edge. Table 7.2 lists the turbulent inlet boundary layer parameters, which were maintained throughout this study.

Table 7.2. Inlet Turbulent Boundary Layer Characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boundary layer thickness/quarter-span (δ/0.25S)</td>
<td>0.68</td>
</tr>
<tr>
<td>Displacement thickness/quarter-span (δ’/0.25S)</td>
<td>0.076</td>
</tr>
<tr>
<td>Momentum thickness/quarter-span (θ/0.25S)</td>
<td>0.06</td>
</tr>
<tr>
<td>Shape factor (δ’/θ)</td>
<td>1.3</td>
</tr>
<tr>
<td>Momentum thickness Reynolds number (Re_{θ})</td>
<td>4245</td>
</tr>
</tbody>
</table>

The two-dimensional slot was placed upstream of the vane cascade as illustrated in Figure 7.2. Also shown in Figure 7.2 are the four slot injection angles of 90°, 65°, 45°, and 30° that were investigated. Each slot had a flow length-to-width of 11.7. The
downstream edge of each slot was located 0.39d_{eff} upstream from the geometric stagnation location of the vane. The temperature of the injecting flow was measured using three thermocouples located within the slot plenum. The recorded plenum temperatures were found to vary less than 0.5°C from the average injection fluid temperature for all experiments.

High and low average momentum flux ratios, I = 2.8 and 0.7, were investigated. The flow exiting the slot was highly non-uniform resulting from the effects of the static pressure variation from the vane and as such, average momentum flux ratios are reported. The average momentum flux ratio was calculated according to the equation in the nomenclature. The total mass flux issuing from the upstream leakage slot was measured using a laminar flow element located within the supply pipe to the slot plenum. A mass flux ratio (MFR) was defined as the ratio of injectant issuing from the upstream slot and entering a single passage to the mainstream mass flow rate through a single passage. With a fixed slot width of w = 3.3mm (0.033d_{eff}) an average momentum flux ratio of I = 2.8 was calculated corresponding to MFR = 1.0%. The low momentum flux ratio condition of I = 0.7 was achieved by lowering the mass flux ratio with the given slot width to MFR = 0.5%.

### 7.2.1 Measurement Methods

For the results presented in this paper, heat transfer coefficients on the endwall at the junction were measured. In addition, time-resolved flowfield measurements were also performed in the junction region within the stagnation plane. Heat transfer measurements were made by capturing steady state endwall temperatures using an Infrared (IR) camera.
with a constant heat flux boundary condition on the endwall. The vanes were constructed of a low thermal conductivity foam and were considered to be adiabatic. The constant heat flux surface was achieved with a thin heater permanently bonded to a low thermal conductivity foam plate to reduce conduction loses to the external environment. The heater consisted of a serpentine inconel pattern, encapsulated in Kapton, with a thin copper layer (~37µm) on the flow side. Lateral conduction as a result of the copper layer was determined to be insignificant compared to convection given the small integration size over which the IR camera averages the surface temperatures [51]. The heater surface was painted black to maintain a high emissivity on the endwall surface thus providing good resolution of the surface temperatures. The heater covered the entire endwall, from immediately downstream of the slot to the trailing edge of the second full vane as shown by the heater outline in Figure 7.2.

Steady state conditions were typically reached after four hours of operation. Once at steady state an Inframetrics P20 camera was placed at 90° to the endwall. The resolution of the camera was 320 X 240 pixels resulting in a spatial integration of 0.75mm. Two thermocouples embedded in the endwall surface were located in the IR picture to calibrate each image. At steady state, a set of five images were taken at the leading-edge of the center vane as indicated in Figure 7.2 and the corresponding surface thermocouple temperatures were recorded. Once captured, the 5 IR images were calibrated using the directly measured surface temperatures. Typical emissivity and background temperature values were in the range of 0.93-0.96 and 25°C-30°C respectively. The background temperature corresponded to the approximate mainstream temperature of the flow as the surrounding environment was allowed to reach steady state. Note that in calculating the
heat transfer coefficient, the total power supplied to the heater was corrected to account for conduction and radiation losses. The percentage of the radiation heat flux loss was typically between 3-16% of the total supplied heat flux. The percentage of conduction heat flux loss was typically between 0.5-1.0%.

Flowfield measurements were made in the stagnation plane of the center vane using high-image-density, time-resolved, digital particle image velocimetry (TRDPIV). TRDPIV is a non-intrusive, indirect, whole field, laser-optical measurement technique based on the illumination and tracking of groups of particles which follow the flowfield [53-57]. For the TRDPIV technique used in this study, a double frame of images was captured over a short time period using a high speed CMOS camera and a high frequency, double pulsed laser to illuminate the seeder particles. Di-Ethyl-Hexyl-Sebacat (DEHS) seeder particles with an approximate mean diameter of 1µm were introduced directly upstream of the wind tunnel blower using a Laskin nozzle. The CMOS camera recorded images with a resolution of 1024x1024 pixels which based on the field of view resulted in approximately 11 pixels/mm. As shown in Figure 7.2 the laser was located over the center vane. The attached optics included a spherical lens to concentrate the laser light, a cylindrical lens to spread the laser into a thin sheet, and a 45° mirror to redirect the light sheet perpendicular to the endwall and along the vane stagnation. The dual cavity 15W Nd:YAG laser produced double pulses of green light at a wavelength of 527nm with a pulse duration less than 180ns and an energy level of 10mJ/pulse.

Flowfields were captured using a camera sampling frequency of 1kHz for 3 seconds resulting in the recording of 3000 image pairs. The sampling frequency and sampling duration were chosen based on the characteristic fluid time scale, $\tau_f$. The characteristic
fluid time scale was based on the boundary layer thickness and the freestream velocity. A sampling frequency and sampling duration of 1kHz and 3 seconds corresponded to $12\tau_f$ and $250\tau_f$ respectively ensuring that the flow features were temporally resolved. The time delay between laser pulses and the resulting image pairs was adjusted for each experiment to obtain a bulk particle displacement of approximately 8 pixels. Once captured, images were processed using commercially available software [58]. A minimum pixel intensity background subtraction was performed on the raw images to improve the signal-to-noise ratio. As indicated in Figure 7.2 the camera was at a slight angle to the stagnation plane, typically less than 8°. Using the software, the adjusted raw images were also corrected for the small off normal viewing angle. A viewing angle of 8° results in a less than 1% correction in the streamwise velocities.

After correction, the images were masked to remove the endwall and vane boundaries, replacing data outside the masked region with zero intensity. The images were then processed using a decreasing, multi-pass technique to determine the displacement vectors. The multi-pass scheme employed a single pass at an interrogation window size of 32x32 pixels followed by two passes at an interrogation window size of 16x16 pixels with 50% overlap resulting in a final vector spacing of 8x8 pixels. Interrogation windows that contained at least 30% masked out area were discarded on the initial pass. For all subsequent passes an interrogation window was only discarded if at least 60% of the window was masked. The displacement vectors were calculated for each interrogation window amongst image pairs using a standard cross-correlation, cyclic FFT-based algorithm. To improve accuracy, a fractional window offset and image mapping through bilinear interpolation was applied to each interrogation window during
intermediate and final passes based on the calculated vector field in the previous pass. Vector validation was performed after each pass using a 4-pass regional median filter with adjustable criteria for the removal and re-insertion of possible spurious vectors. The vector validation scheme was visually checked for several representative instantaneous flowfields to ensure that only spurious vectors were removed.

### 7.2.2 Measurement Uncertainties

An uncertainty analysis was performed on the measurements of heat transfer using the partial derivative method described by Moffat [45]. Uncertainty in Stanton numbers was dominated by the uncertainty in surface temperature measurements. For those measurements, a precision uncertainty of approximately ±0.2°C was estimated from the standard deviation of the five IR images measurements based on a 95% confidence interval. The bias uncertainty was taken as the root-sum-square of the thermocouple bias uncertainty (±0.2°C) and the average deviation of the calibrated images from the thermocouples (±0.5°C) resulting in a bias uncertainty of ±0.54°C. Combining the bias and precision uncertainties, a total uncertainty in Stanton number was found to be \( \partial St = \pm 0.0005 \) (5%) at \( St = 0.01 \) and \( \partial St = \pm 0.0018 \) (9%) at \( St = 0.02 \). Note that higher uncertainties are associated with higher Stanton numbers due to smaller temperature differences between the mainstream and heater surface.

Uncertainty in the TRDPIV velocity measurements was calculated based on the average particle displacement, maximum displacement gradient, average particle image density, and signal-to-noise ratio [57, 64]. The uncertainty in the instantaneous velocity measurement was found to be approximately \( \partial U = 0.16 \text{m/s} \) (2% of \( U_\infty \)) for a bulk particle
displacement of 8 pixels. Expanding the instantaneous velocity uncertainty to the turbulence intensity as defined in the nomenclature based on the partial derivative method, an uncertainty in turbulence intensity was calculated to be $\Delta Tu = \pm 0.02$ for $Tu = 0.1$ to 0.4.

7.3 Time-Averaged Results

Baseline conditions without an injection slot were measured to compare to those results with injection upstream of the vane leading-edge. Figures 7.3a-7.3e presents the contours of Stanton number on the endwall in the junction region. For the baseline condition shown in Figure 7.3a, near the upstream edge of the heater, located 0.39$d_{eff}$ from the vane stagnation, the endwall heat transfer shows the effects of an unheated starting length. At the stagnation, however, an expected increase in the endwall heat transfer is observed as a result of the junction flow.

To compare the heat transfer measurements with previously published data, Figure 7.4 gives the augmentation of the endwall heat transfer for both circular cylinders and leading-edge vane geometries with no upstream injection slot present. For the cylindrical data, the endwall heat transfer augmentation is calculated using the heat transfer coefficients along the stagnation line approaching the leading-edge divided by the Stanton numbers along the same streamwise position but with no cylinder present. For the vanes, the heat transfer augmentation is calculated by dividing the heat transfer coefficients along the stagnation line by those along the midpitch line to minimize the influence from the vanes. Concerning the vane data, Figure 7.4 indicates fairly good agreement in endwall heat transfer between the studies. Away from the stagnation the
slightly higher augmentation reported by Kang et. al [51] can be attributed to the boundary layer thickness that was approximately 50% thinner than that of the current study. Very close to the stagnation (x/d<sub>ef</sub> < -0.1), however, the augmentation from the

![Diagram of heat transfer](image)

**Figure 7.3.** Endwall heat transfer in the junction region (a) with no injection slot and for injection angles of b) 90°, c) 65°, d) 45°, and e) 30° with I = 2.8 and MFR = 1.0%.
current study is higher than that measured by Kang et. al [51] as a result of an effective leading-edge diameter that was approximately 70% larger than that used in the current study [67]. For the circular cylinder data, fair agreement with the vane studies is also observed except very close to the stagnation where Ireland and Jones [68] provide the only circular cylinder data point. Ireland and Jones [68] used a short circular cylinder within an internal channel subject to a developed flow resulting in much higher augmentation at the stagnation than that observed in the vane studies.
The stagnation of the approaching velocity boundary layer onto the leading-edge of the vane results in a static pressure gradient along the leading-edge that drives the flow towards the endwall. The flow is reversed towards the upstream along the endwall resulting in the formation of a time-averaged HSV as shown in Figure 7.5a. Figure 7.5b

**Figure 7.5.** Comparison of the (a) time-averaged flowfield vectors with overlaid streamlines and contours of turbulence intensity in the stagnation plane to the (b) endwall heat transfer along the stagnation line with no injection slot.
presents the associated line plot of endwall heat transfer along the stagnation line which is highlighted in Figure 7.3a. The stagnation line heat transfer shows an increase near the stagnation which coincides with the impingement of the mainstream flow as indicated by the streamlines. In addition to the high turbulence levels generated by the HSV, Figure 7.5a indicates the time-averaged separation or saddle point where the near wall flow begins to reverse upstream at approximately \( x/d_{\text{eff}} = -0.35 \). The time-averaged separation location of the impinging boundary layer agrees well with past studies for circular cylinder and symmetric airfoil type junction flows which collectively report a separation location in the range of \( 0.3 < |x/d_{\text{eff}}| < 0.5 \) [59, 68, 70-74].

### 7.3.1 Effects of Slot Angle for High Momentum Injection

To determine the effects of high momentum slot injection on the heat transfer and flowfield in the leading-edge region, high momentum air (\( I = 2.8, \text{MFR} = 1.0\% \)) was injected upstream of the vanes through a two-dimensional slot over a range of injection angles. Figures 7.3b-7.3e present contours of Stanton number on the endwall at the junction for injection angles of 90°, 65°, 45°, and 30°. In comparison to the baseline case with no injection slot in Figure 7.3a, high momentum injection for each slot angle results in a substantial increase in endwall heat transfer. Concerning the cases with injection, there is a fundamental difference in the distribution of endwall heat transfer between the 90° and 65° slots as compared with the 45° and 30° slots respectively. For high momentum injection at 90° and 65° the endwall heat transfer near the stagnation is similar between the two injection angles and higher than that observed for the 45° and 30° slots. Upstream of the stagnation, however, endwall heat transfer is higher for the
65° slot in comparison to the 90° slot. In contrast, near the exit of the injection slot the endwall heat transfer for the 45° and 30° slots is similar and higher than that observed for the 90° and 65° slots.

To understand the differences in endwall heat transfer, Figures 7.6a-7.6d present the time-averaged flowfields with streamlines overlaid with contours of turbulence intensity in the stagnation plane of the vane for injection angles of 90°, 65°, 45°, and 30°. Below each flowfield, Figures 7.6e-7.6h present the respective endwall heat transfer along the stagnation line in comparison to the baseline case for injection angles of 90°, 65°, 45°, and 30°. In comparison to having no injection slot, high momentum injection at 90° and 65° results in the formation of a much larger HSV while injection at 45° and 30° shows an absence of a HSV. The strong shear layer and the subsequent high turbulence levels resulting from injecting high momentum fluid along the endwall for the 45° and 30° slots are also shown to provide high heat transfer. Regardless, for each slot angle with high momentum injection, turbulence levels are above that observed with no injection and the heat transfer along the stagnation line is also higher than the baseline.

A reduction in the injected flow penetration depth and a subsequent reduction in the size of the time-averaged vortex are evident in comparing the 90° and 65° slots. A comparison of Figures 7.6a and 7.6b shows that the center of the time-averaged vortex forms closer to the endwall for the 65° slot. The addition of a streamwise injection component for the 65° slot as compared to the 90° slot also drives the vortex closer to the vane stagnation. Injection at 65° results in higher turbulence levels than that observed for the 90° slot thereby giving higher endwall heat transfer. Note that a small secondary vortex forms upstream of the fluid injected at 90° as a result of the physical blockage to
Figure 7.6. Comparison of time-averaged flowfield vectors with overlaid streamlines and contours of turbulence intensity in the stagnation plane for injection angles of a) 90°, b) 65°, c) 45°, and d) 30° to the respective endwall heat transfer along the stagnation line for injection angles of e) 90°, f) 65°, g) 45°, and h) 30° with $l = 2.8$ and MFR = 1.0%.
the approaching boundary layer for the injection. For 65° injection, the measurements do not indicate a resolvable secondary vortex upstream of the injection.

In contrast to the 90° and 65° injection cases, 45° and 30° injection has a much stronger streamwise component. The stronger streamwise component results in the high momentum fluid being injected along the endwall where it convects upward along the surface of the vane eliminating the downwash and subsequent impingement of the boundary layer flow onto the endwall. As a result of this differing flow pattern, the endwall heat transfer at the vane stagnation is lower in the case of the 45° and 30° slots relative to the 90° and 65° slots where downwash is present. Note that stronger upwash as a result of an increased streamwise injection component for the 30° slot in comparison to the 45° slot results in the regular formation of a small counter-clockwise (CCW) rotating vortex on the vane surface. At the exit of the injection slot for the 90° and 65° slots there is a small, low velocity region between the injecting flow and the vortex. As a result of the higher velocities in the 45° and 30° slot cases, there is an increase in the heat transfer near the exit of the injection slot relative to the 90° and 65° slots.

7.3.2 Effects of Slot Angle for Low Momentum Injection

In addition to high momentum injection upstream of the vane, experiments were also conducted at a low momentum flux ratio (I = 0.7, MFR = 0.5%). Figures 7.7a-7.7e present contours of Stanton number on the endwall at the junction for the baseline case and for injection angles of 90°, 65°, 45°, and 30° at the reduced momentum flux ratio. The general distribution of endwall heat transfer that was observed for the baseline case and high momentum injection cases persists for reduced momentum flux injection. That
Figure 7.7. Endwall heat transfer in the junction region (a) with no injection slot and for injection angles of b) 90°, c) 65°, d) 45°, and e) 30° with $I = 0.7$ and MFR = 0.5%.

is, the endwall heat transfer is initially high at the exit of the injection slot as a result of an unheated starting length and gradually decreases on approach to the vane up to the stagnation where heat transfer increases. Unlike the endwall heat transfer with high momentum injection, however, heat transfer levels in the junction region when low
momentum injection is present are within the same range as that observed with no injection slot.

To understand the similarities in heat transfer between low momentum injection and the case without an injection slot present, the stagnation plane flowfields accompanied with the respective endwall heat transfer along the stagnation line are presented in Figures 7.8a-7.8h for the reduced momentum flux ratio. In contrast to high momentum injection, low momentum injection results in the formation of an HSV for each injection angle. Low momentum injection at 90° and 65° results in the formation of a time-averaged vortex that is of the same approximate size, location, and turbulence level as formed at the corner of the junction that also produces turbulence levels in the same range as those observed in the baseline. The presence of an HSV and the associated turbulence drive the endwall heat transfer in the junction region thus the heat transfer levels and distributions are similar between low momentum injection at each slot angle and the case with no injection slot.

Comparing to the flowfields shown previously for high momentum injection, the leading edge vortex that forms for 90° injection is much larger and results in higher turbulence levels than that observed for the reduced momentum flux case. For 90° injection, the leading-edge vortex that forms for each momentum flux ratio case is centered at $x/d_{eff} = -0.25$. This is expected as each momentum flux ratio condition is under the influence of the same mainstream Reynolds number and boundary layer. There is a reduction in the vertical position of the vortex core for the 90° slot, however, with a reduction in momentum flux. The vortex formed for low momentum injection at 65° also maintains the same streamwise location as the high momentum flux case. At both mome-
Figure 7.8. Comparison of time-averaged flowfield vectors with overlaid streamlines and contours of turbulence intensity in the stagnation plane for injection angles of a) 90°, b) 65°, c) 45°, and d) 30° to the respective endwall heat transfer along the stagnation line for injection angles of e) 90°, f) 65°, g) 45°, and h) 30° with I = 0.7 and MFR = 0.5%.
ntum flux ratios the vortex formed from 65° injection maintains the same vertical position. For low momentum injection at 45° and 30° the velocity deficit within the approaching boundary layer is not fully energized thereby resulting in the HSV. In comparison to the baseline case, the HSV formed for low momentum injection at 45° and 30° is pushed closer to the vane stagnation.

In general, low momentum injection results in weaker turbulence levels and lower endwall heat transfer in the junction region for all four slot angles in comparison to the high momentum injection. At high momentum injection, the slot-averaged injection velocity is approximately 70% higher than the freestream velocity. Consequently, the large velocity difference between the injecting fluid and near wall flow leads to large separation regions for the 90° and 65° slots and strong shear layers for the 45° and 30° slots respectively. With low momentum injection, however, the slot-averaged injection velocity is approximately 17% lower than the freestream. The smaller velocity disparity between the injecting fluid and near wall flow for injection at low momentum results in smaller separation regions and weaker shear layers thereby reducing the stagnation plane turbulence for each slot angle in comparison to high momentum injection. Expectantly, the endwall heat transfer at the junction for low momentum injection compared to high momentum injection is lower for each slot angle.

7.4 Time-Resolved Flowfield Results

The particular dynamics of the HSV has been measured by the current authors [52, 75] for external and internal flows as well as by Praisner and Smith [60] using time-resolved, PIV. In these past studies, the HSV was characterized as having a bimodal flow
pattern as the outer wall fluid impinged at stagnation, turned upstream, and interacted with the near wall fluid of the approaching boundary layer. Figures 7.9a and 7.9b present instantaneous flowfields with no injection. As identified by the structure of the flowfield, in the first mode the reverse near wall flow passes under the HSV and travels upstream over a small secondary vortex, which may or may not be present, to eventually sweep into a tertiary vortex. The first mode persists until it is interrupted by a quasi-periodic sw-

\[ W_i/U_\infty \]

\[ Z/0.25S \]

\[ U_i/U_\infty \]

\[ t/\tau_f - 0 \]

\[ t/\tau_f = 0.64 \]

\[ X/\text{d}_{eff} \]

\[ W_i/U_\infty \]

\[ Z/0.25S \]

\[ U_i/U_\infty \]

\[ t/\tau_f - 0 \]

\[ t/\tau_f = 0.64 \]

\[ X/\text{d}_{eff} \]

**Figure 7.9.** Instantaneous flowfield vectors with overlaid streamlines in the stagnation plane with no injection slot showing (a) the first flow mode and (b) the second flow mode experienced by the HSV.
-itching to a second mode. In the second mode, the reverse near wall flow stops passing in to the tertiary vortex as the outer wall fluid penetrates to the endwall surface behind the HSV forcing the structure further upstream. The oblong shaped distribution of the high turbulence contours shown in Figure 7.5a about the time-averaged position of the vortex core highlights the movement of the HSV during the quasi-periodic switching between flow modes.

7.4.1 Effects of Slot Angle for High Momentum Injection

To quantify the instantaneous strength of the vortices for the various injection conditions, Figures 7.10a-7.10d present contours of probability density function (PDF), as defined in the nomenclature, based on swirling strength that isolates rotation within a flowfield. Swirling strength is defined as the positive imaginary component of the complex eigenvalue of the local two-dimensional, velocity-gradient tensor. The PDF contours were generated using a threshold for swirling strength based on the inverse square of the time required for the high momentum injectant to reach the leading-edge of the vane along the streamwise direction. The swirling strength threshold, as defined in the nomenclature, was found to provide an accurate representation of the time-resolved, rotational nature of the flowfield. Changing the threshold criteria by an order of magnitude did not result in the discovery or elimination of any time-resolved features among injection cases. Only the scale of the PDF was altered by changing the threshold criteria.

As shown in Figure 7.10b, high momentum injection at 90° results in a high PDF level with a uni-model distribution centered on the time-averaged position of the vortex
Figure 7.10. Probability density function contours of swirling strength for a threshold of \((U_i/0.39d_{eff})^2\) in the stagnation plane with (a) no injection slot and for injection angles of b) 90°, c) 65° and d) 45° with \(I = 2.8\) and MFR = 1.0%. 

\[ Z/0.25S \]
as shown in Figure 7.6a. The high PDF level for injection at 90° highlights the strength of the instantaneous vortex in comparison to the baseline case in Figure 7.10a where PDF levels are substantially lower and are distributed in an oblong pattern. Figure 7.10c indicates that injection at 65° generally results in a weaker vortex than that observed for 90° injection as the peak PDF level is lower. Similar to the 90° slot, however, PDF levels for 65° injection are substantially higher than in the baseline case. Figure 7.10c also shows the infrequent production of a secondary vortex upstream of the 65° injecting coolant which was not resolved in the time-average.

To investigate the stability of the vortices, Figures 7.11a-7.11d present histogram contours of the streamwise velocities along vertical profiles for the baseline case as well as for high momentum injection at 90°, 65°, and 30°. Note that histogram bin sizes corresponding to approximately 5% of the freestream velocity were used. For Figures 11a-11c the profiles are located at a streamwise position corresponding to the center of the time-averaged vortices. The histograms indicate the steadiness of the vortex by the spreading of the contour levels at a given distance from the endwall. In Figures 11a-11c the presence of the clock-wise rotating vortex is evident in the positive streamwise velocities far from the endwall and negative velocities close to the endwall. As expected, the largest peak velocities are associated with high momentum injection at 90° while the smallest peak velocities are associated with the baseline case. The range of streamwise velocities for high momentum injection at 90° is similar to those of the baseline case indicating the relative stability of the large vortex. High momentum injection at 90° washes the boundary layer flow into the mainstream allowing the instantaneous vortex to remain stable.
Figure 7.11. Histogram contours of streamwise velocity in the stagnation plane with the overlaid time-averaged velocity profiles for (a) the baseline case with no injection and for injection angles of b) 90°, c) 65° and d) 30° with \( I = 2.8 \) and MFR = 1.0%.

For high momentum injection at 65° shown in Figure 7.10c, however, the instantaneous vortex exhibits a more unsteady nature than that observed for both 90° injection and the baseline case. For 65° injection, the approaching boundary layer flow is able to frequently disrupt the injecting fluid leading to substantial fluctuation and distortion of the instantaneous vortex. As shown in Figure 7.11c, the range of instantaneous velocities observed for 65° injection is much larger than that shown for
both injection at 90° and the baseline case. A relatively unsteady vortex for high momentum injection at 65° results in the higher peak turbulence levels that were shown previously in Figure 7.6b.

To provide further evidence of the time-resolved dynamics of the vortex produced by 65° injection, Figures 7.12a-7.12c present several instantaneous flowfields in the stagnation plane of the vane. Typically, the injecting flow is oriented at approximately 65° as shown in Figure 7.12a resulting in the formation of a fairly circular vortex. Intermittently the approaching boundary layer flow is able to disrupt the vertical penetration depth and angle of the injectant, forcing the vortex closer to the endwall and translating the distorted structure further downstream. Figures 7.12a-7.12c indicate that the boundary layer flow is able to frequently rush into the region just downstream of the vortex and the vane. The time-averaged result of the turning of boundary layer streamlines into the region near the vane stagnation is higher turbulence levels for 65° injection in comparison to the 90° slot as shown previously in Figure 7.6.

In contrast to the time-averaged results in Figure 7.6c, Figures 7.10c captures the production of a secondary vortex on the vane surface along the stagnation line for injection at 45° at a height of z/S = 0.22. Note that injection at 30° results in a very similar PDF distribution to that shown for the 45° slot. Stronger upwash in the case of high momentum injection at 30°, however, produces larger vertical velocities on the vane stagnation resulting in earlier separation. Earlier separation on the vane stagnation for injection at 30° results in a vortex on the surface of the vane that is located closer to the endwall than that observed for 45° injection as shown previously in Figure 7.6d. A comparison of Figures 7.6c and 7.10c for the 45° slot shows that high PDF contours are
Figure 7.12. Instantaneous flowfield vectors with overlaid streamlines in the stagnation plane showing the evolution of the vortex for injection at 65° with $I = 2.8$ and $MFR = 1.0\%$. 
effectively bounded by the time-averaged inflection points of the velocity. The inflection points indicate the shear layer below which the injecting fluid undergoes a strong interaction with the boundary layer flow. Upwash at the corner of the junction leads to a frequent production of high swirling strength. A turning of the near wall flow toward midspan results in the formation of strong CCW rotating vortices near the corner of the junction.

Although weaker in strength than the intermittent vortices formed near the corner of the junction, CCW rotating vortices also form along the inflection points in the time-averaged streamlines for injection at both 45° and 30°. Figures 7.13a-7.13c present instantaneous flowfields in the stagnation plane of the vane for high momentum injection at 30° indicating the formation of small CCW rotating vortices. The intermittent vortices that form convect along the shear layer and up the surface of the vane to either conglomerate into the vortex that is present on the vane surface or move out of the stagnation plane. In addition to the intermittent vortices, the instantaneous flowfields indicate a strong upwash along the vane surface acts as impedance to the approaching boundary layer moving the subsequent downwash further upstream. As a result, the boundary layer flow is driven into the shear layer often disrupting the strong streamwise flow along the endwall. The disruption to the near wall flow either results in a compression of the near wall streamlines or a small separation region as shown in Figure 7.13c.

While the high momentum injectant is able to remain attached to the endwall for the 45° and 30° slots, the magnitude of the streamwise velocities are subject to substantial fluctuation as a result of the disturbances caused by the intermittent vortices and downw-
Figure 7.13. Instantaneous flowfield vectors with overlaid streamlines in the stagnation plane showing the formation and translation of intermittent vortices along the shear layer and the resulting disruption of the endwall flow for injection at 30° with I = 2.8 and MFR = 1.0%.
-ash. Figure 7.11d presents a histogram of the streamwise velocity for the 30° slot along a vertical profile located at $x/d_{eff} = -0.23$ corresponding to the midpoint between the time-averaged vortices that was evident for the 90° and 65° cases. Although confined near the endwall, the spreading of streamwise velocities is similar to that shown for the 65° slot resulting in similar peak turbulence levels. Note that the spreading of streamwise velocities for the 45° slot was only slightly less than that shown for the 30° slot with the average peak velocity being lower as a result of a weaker streamwise injection component for the 45° slot.

7.4.2 Effects of Slot Angle for Low Momentum Injection

For low momentum injection, Figures 7.14a-7.14c present PDF contours of swirling strength for injection angles of 90°, 65°, and 45° using the same threshold criteria as in Figure 7.10 for high momentum injection for comparison. Similar to high momentum injection at 90°, a uni-modal distribution is observed for low momentum injection at 90° although peak PDF levels are lower indicating a weaker instantaneous vortex. Figure 7.14a also shows the infrequent production of a small secondary vortex upstream of the 90° slot that was not evident in the time-averaged results. Similar to that seen for high momentum injection, reducing the injection angle to 65° results in a reduction in peak PDF level in comparison to low momentum injection at 90° as shown in Figure 7.14b. A secondary peak in PDF level downstream of the 65° slot highlights the intermittent formation of a small secondary vortex between the injecting fluid and the primary vortex. A large distance between slot and vortex in addition to a weak instantaneous vortex for low momentum injection at 65° allows for the frequent formation of a small secondary
Figure 7.14. Probability density function contours of swirling strength for a threshold of $(U_i / 0.39d_{eff})^2$ in the stagnation plane for injection angles of a) 90°, b) 65° and c) 45° with $I = 0.7$ and MFR = 0.5%.
Figure 7.15. Instantaneous flowfield vectors with overlaid streamlines in the stagnation plane showing the dissipation of a small secondary vortex formed between the injecting fluid and the primary vortex for injection at 65° with I = 0.7 and MFR = 0.5%.
vortex. Figures 7.15a-7.15c present instantaneous flowfields indicating the formation and dissipation of the secondary vortex for low momentum injection at 65°. When present, the secondary vortex is moved downstream by the injecting fluid where it is dissipated by the primary vortex as the two structures rotate in opposing directions. The existence of the secondary vortex, however, does not result in a significant displacement of the primary vortex.

Figures 7.16a-7.16c present histogram contours of the streamwise velocities along vertical profiles for low momentum injection at 90°, 65° and 30°. Note that for Figures 7.16a and 7.16b the vertical profiles are located at a streamwise position corresponding to the center of the time-averaged vortices. Similar to high momentum injection, low momentum injection at 90° results in a range of streamwise velocities that is very similar to the baseline case. In contrast to high momentum injection at 65°, however, the range of streamwise velocities is small and in the same range as that observed for the baseline case and injection at 90° for low momentum injection at 65°.

The PDF contours of swirling strength in the stagnation plane for low momentum injection are very similar between the 45° and 30° slots. As such Figure 7.14c presents only the contour for low momentum injection at 45°. As expected the peak PDF value is concentrated at the corner of the junction where the time-averaged results indicate the regular formation of a small vortex. The interaction between the downwash and the flow injected along the endwall, however, often results in the removal of the corner vortex for the 45° and 30° slots. Figures 7.17a-7.17d present instantaneous flowfields for low momentum injection at 30° showing how the corner vortex is reduced in size by the downwash to the point where the structure is removed.
Figure 7.16. Histogram contours of streamwise velocity in the stagnation plane with the overlaid time-averaged velocity profiles for injection angles of a) 90°, b) 65° and c) 30° with $I = 0.7$ and MFR = 0.5%.
Figure 7.17. Instantaneous flowfield vectors with overlaid streamlines in the stagnation plane showing the interaction between the endwall flow and the downwash resulting in the removal of the corner vortex for injection at 30° with $I = 0.7$ and MFR = 0.5%.
Further upstream from the corner vortex, the endwall flow remains relatively stable as shown in Figure 16c which presents a PDF of streamwise velocities along the same vertical profile used for high momentum injection at 30°. The range of streamwise velocities is much smaller than those observed for high momentum injection at 30°. Note that low momentum injection at 45° results in a similar range of streamwise velocities as low momentum injection at 30°. For the 45° and 30° slots at low momentum injection the endwall flow is not subject to intermittent vortices formed along the shear layer as in the high momentum flux case allowing the magnitude of the streamwise velocities to remain relatively stable.

7.5 Conclusions

Experiments were conducted to investigate the effects of flow injection angle on the stagnation plane flowfield and associated endwall heat transfer upstream of a rounded leading-edge and flat wall junction. A range of injection angles for a two-dimensional slot were considered for both high and low momentum injection.

Injecting high momentum flow upstream of the junction provided an increase in endwall heat transfer for each injection angle relative to the no injection slot case. A time-averaged vortex turning toward the endwall was formed in the stagnation plane for both 90° and 65° injection, considerably augmenting the local turbulence levels and endwall heat transfer. While injection at 90° produced a vortex that remained stable, the vortex formed from injection at 65° was frequently disrupted by the boundary layer flow. The disruption to the injecting flow distorted the vortex, forcing it closer to the endwall
and further upstream. The unsteadiness of the vortex for 65° injection produced higher turbulence levels and subsequently higher endwall heat transfer than 90° injection.

For high momentum injection at 45° and 30° no time-averaged vortex on the endwall was measured. The injecting flow for the 45° and 30° slots resulted in strong shear layers along the endwall and an upwash of fluid at the corner of the junction. In several instances, counter-clockwise rotating vortices were intermittently formed along the shear layer. The vortices convected along the shear layer where the structures were either moved out of the stagnation plane or were merged into a vortex that was periodically present on the leading-edge surface. The secondary vortices disrupted the endwall flow leading to high turbulence levels and subsequently high endwall heat transfer relative to the no injection case.

Similar to the stagnation plane flowfields, the endwall heat transfer distributions were shown to be characteristically different between injection at 90° and 65° and at 45° and 30° for high momentum injection. Downwash at the junction and the subsequent impingement of the boundary layer flow on the endwall provided increased heat transfer near the stagnation for injection at 90° and 65° in comparison to injection at 45° and 30° as no downwash was present. In contrast, heat transfer levels were higher near the exit of the injection slot for 45° and 30° in comparison to injection at 90° and 65°. A small region between the injecting flow and the time-averaged vortex formed from 90° and 65° injection was characterized by low velocities resulting in low heat transfer near the slot exit in comparison to injection at 45° and 30° where the flow was attached.

Reducing the injection momentum flux reduced the velocity difference between the injectant and near wall flow, substantially reducing the mixing and the resulting
turbulence levels. Subsequently, endwall heat transfer levels were reduced for each slot angle relative to high momentum injection. Although smaller in size and weaker in intensity than that observed for high momentum injection, a relatively steady vortex was formed in the stagnation plane for injection at both 90° and 65° with low momentum. Low momentum injection at 45° and 30° was mostly unable to counteract the strong turning of flow toward the endwall leading to the frequent formation of a vortex at the corner of the junction. The corner vortex, however, was consistently forced toward the leading-edge stagnation where the structure was dissipated. The vortex was reformed, however, as the reverse flow along the endwall separated near stagnation. Low momentum injection for each slot angle resulted in turbulence levels in the same range as those observed with no injection slot, as a result the endwall heat transfer was similar.
Chapter 8

Overview of Conclusions

This dissertation was concerned with the influence of two realistic features of a first stage, nozzle guide vane passage on the cooling of the vane endwalls. Cooling of endwalls is complicated as a result of secondary flows which arise from the stagnation of the velocity boundary layer at the leading-edge of the vanes. Endwall contouring and coolant injection from the combustor-turbine interface have the potential to alter secondary flows and subsequently the performance of any coolant injected along the endwall.

The axisymmetric contouring of one endwall in the form of a linear slope was used to simulate the transition from the combustor to the turbine section over the first stage vanes. Results from an axisymmetric contoured passage were compared to those of a planar passage in the presence of coolant injection from both an upstream interface slot and discrete film cooling holes. Relative to a planar passage, the endwalls of the contoured passage are subject to a greater streamwise acceleration. In addition, the contoured endwall experiences a turning of streamlines near the leading-edge of the contour as a result of flow blockage. Past studies in the available literature indicated that secondary flows and the resulting passage aerodynamic losses are reduced for an axisymmetric contoured passage in comparison to a planar passage as a result of increased freestream acceleration. The work presented in this dissertation focused on the effects of axisymmetric contouring on the cooling performance of discrete holes and the
interface slot in comparison to a planar passage as this analysis does not exist in the available literature.

The performance of coolant injection from discrete film cooling holes was the poorest for the contoured endwall. As the freestream flow approaches the contoured endwall it must impinge and turn near the leading-edge of the contour resulting in higher static pressures on the contoured endwall relative to the flat endwall. Higher endwall static pressures reduce the local blowing ratios of discrete holes. In addition, the impingement of the freestream flow breaks up the coolant jet injecting from the discrete holes. Lower local blowing ratios and coolant jet disruption contribute to the poor performance of coolant injected from discrete film cooling holes on the contoured endwall. The cooling performance of discrete holes on the flat endwalls of the planar and contoured passages was similar showing that increased streamwise acceleration has little impact on the performance of coolant injected at the endwall.

Coolant coverage from the interface slot was the largest for the contoured endwall at each injection flow rate tested in comparison to the flat endwalls of the planar and contoured passages. The orientation of the interface slot in each endwall was maintained perpendicular to the inlet flow direction. As a result of the fixed interface slot orientation, the interface slot in the contoured endwall was at a reduced injection angle relative to the interface slot in the flat endwalls. With an endwall contour corresponding to a 16° slope, the interface slot injected coolant at an angle of 74° relative to the contoured endwall in comparison to 90° for the flat endwalls. Improved coolant coverage from the interface slot on the contoured endwall can be attributed to this reduced injection angle. The performance of coolant injected from the interface slot was similar between the flat
endwalls of the planar and contoured passages indicating again that the additional streamwise acceleration has little effect on the injecting coolant.

Without coolant injection, the endwalls of the contoured passage were shown to have lower heat transfer levels than the corresponding flat endwalls of the planar passage. The additional streamwise acceleration as a result of the contoured passage provided a larger favorable pressure gradient for the incoming flow than the planar passage. A larger favorable pressure gradient resulted in a thinner boundary layer at the vane stagnation as the endwall contour began upstream of the vane cascade. A thinner boundary layer does not inherently result in lower heat transfer. The resulting formation of a smaller and weaker horseshoe vortex leading to weaker passage secondary flows as a result of the thinner boundary layer, however, does lead to lower endwall heat transfer in the contoured passage.

With the introduction of coolant injected from an upstream interface slot, the endwalls of the contoured passage continued to experience lower heat transfer levels than the flat endwall of the planar passage. Relative to the cases without coolant injection, however, the interaction of coolant with the near wall flow served to increase endwall heat transfer. Cooling effectiveness results indicated that coolant injected from the interface slot was concentrated near the suction side leading-edge of the passage as a result of the non-uniform endwall pressure distribution imposed by the vanes. Correspondingly, heat transfer levels on each endwall in the presence of coolant injection were increased near the suction side leading edge relative to the cases with no cooling. The sweeping of the injected coolant across the passage from pressure to suction side for each endwall signified the presence of endwall crossflow and the passage vortex. The
passage secondary flows prevented the coolant injected from the interface slot from reaching the pressure side of the passage. As a result the heat transfer levels near the pressure side of the passages were similar among all three endwalls and were unaffected by the coolant injection.

Coolant injection from a two-dimensional slot placed upstream of the vane passage was used to simulate the interface between the combustor and turbine section. A comparison of the results between the axisymmetric contoured and planar passage indicated that small changes to the orientation of the interface slot can result in significant changes to the endwall cooling. Seeking further understanding of the interface slot coolant injection, the influence of the injection flow rate, position, and orientation of the interface slot on the secondary flows and cooling of the first stage vane endwalls in a planar passage was also investigated.

Concerning the position of a perpendicular interface slot, the most upstream position located at 34% axial chord upstream of the vane was shown be similar in performance to the nominal position at 17% axial chord upstream. The coolant distribution and effectiveness levels were similar between the upstream and nominal interface slot positions relative to the slot. Within the passage, however, effectiveness levels were lower for coolant injected from the further upstream position as the coolant had more distance to mix with the freestream flow and degrade. Moving the interface slot further downstream to 5% axial chord from the vanes improved the local cooling effectiveness, particularly near the suction side leading-edge of the passage in comparison to the further upstream positions.
The similarity in cooling performance between the nominal and further upstream positions is a result of similar secondary flows. Inspection of the stagnation plane flowfields at the vane leading-edge showed that the injecting coolant resulted in the separation of the incoming flow and the subsequent formation of a horseshoe vortex. For the nominal and further upstream positions the horseshoe vortex that was formed was very similar. For the interface slot located closest to the vane cascade, however, the horseshoe vortex was smaller in size but greater in intensity than that formed for the two further upstream positions. At the vane stagnation the local coolant injection rate from the interface slot is smaller for the 5% axial chord position due to the large endwall static pressure near the vane. Reducing the local injection rate reduced the penetration depth of the injected coolant at stagnation, subsequently reducing the separation region and the size of the horseshoe vortex. A smaller horseshoe vortex resulted in less mixing of the coolant and freestream allowing the coolant to provide higher effectiveness levels on the endwall near the suction side leading-edge of the vane.

Concerning the interface slot orientation, at high injection flow rates a reduction in the interface slot orientation from perpendicular to $65^\circ$ was shown to improve cooling effectiveness and coverage by directing the coolant more along the endwall instead of into the freestream where it is useless in cooling the endwall. An additional reduction from $65^\circ$ to $45^\circ$ provided a further increase in cooling effectiveness level, particularly near the pressure side of the passage and trailing edge. A final reduction in slot orientation to $30^\circ$ did not provide any substantial improvement over coolant injected at $45^\circ$. Coolant injected at $45^\circ$ and $30^\circ$ was shown to provide uniform coverage of the
endwall in comparison to the 90° and 65° interface slots where a strong sweeping of effectiveness levels from the pressure to the suction side of the passage was observed.

Similar to the cooling effectiveness results, endwall heat transfer patterns were shown to be characteristically different between the 90° and 65° interface slots and the corresponding 45° and 30° orientations at high injection flow rates. For the 90° and 65° orientations, the highest endwall heat transfer occurred in the stagnation region. A large horseshoe vortex was formed in the stagnation plane for both 90° and 65° injection, considerably augmenting the local turbulence levels and endwall heat transfer in comparison to when no interface slot was present. The presence of a horseshoe vortex and the subsequent formation of the passage vortex resulted in the strong sweeping of the coolant from the pressure to the suction side of the passage. Coolant injected at 45° or 30° did not result in the formation of a horseshoe vortex, instead the coolant was injected along the endwall resulting in a strong upwash of fluid at the vane-endwall junction. The injection of high velocity coolant at 45° and 30° served to fill in the velocity boundary layer resulting in a reduction of endwall crossflow and the subsequent sweeping of the coolant. The absence of a strong sweeping in both heat transfer and cooling effectiveness contours from the pressure side to the suction side of the passage for the 45° and 30° interface slots indicated that the passage vortex and endwall crossflow were diminished. Injecting coolant along the plane of the endwall produced strong shear layers and subsequently high turbulence levels near the exit of the interface slot. As a result, high endwall heat transfer values were observed downstream of the interface slot along the entire passage pitch instead of just at the stagnation region as was the case for 90° and 65° orientations.
Lowering the injection flow rate restricted the injected coolant to the suction side of the passage indicating the strong influence of secondary flows for each interface slot orientation. Reducing the injection flow rates also reduced the velocity difference between the coolant and near wall flow, substantially reducing the mixing and the resulting turbulence levels. Subsequently, endwall heat transfer levels were reduced for each interface slot orientation relative to large injection flow rate results. Although smaller in size and weaker in intensity than that observed for the large injection rates, a horseshoe vortex was formed in the stagnation plane for low flow rate injection at both 90° and 65° as a reduced injection rate led to a reduced separation region. Unlike the high flow rate results, low flow rate injection at 45° and 30° was unable to sufficiently fill in the low velocity region of the boundary layer near the endwall. As a result, the static pressure gradient at the vane stagnation was still directed toward the endwall which prevented the flow from washing up the surface of the vane leading to the frequent formation of a horseshoe vortex at the corner of the vane-endwall junction. Coolant injection for each interface slot orientation at the lower injection flow rate resulted in a horseshoe vortex and turbulence levels similar to that observed with no interface slot, as a result the endwall heat transfer was similar.

8.1 Recommendations for Future Work

Concerning the axisymmetric contouring of a vane endwall, only one endwall slope was considered in this study. Results indicated that the additional streamwise acceleration imparted by the contour allowed for lower endwall heat transfer in the contoured passage by reducing the size of secondary flows. A further increase in streamwise acceleration
from a larger endwall slope may result in a further reduction in secondary flows and subsequent endwall transfer for the contoured passage relative to the planar passage. The contouring of the endwall, however, was shown to drastically reduce the performance of coolant injected from discrete film cooling holes particularly near the leading-edge of the contour. To counteract this effect while still benefiting from the reduced heat transfer as a result of contouring the start of the endwall contour could be moved downstream of the passage inlet. While there are studies that exist in the available literature for axisymmetric contours starting downstream of the passage inlet, these studies do not include coolant injection from discrete holes located in the contoured endwall.

In this dissertation, only a perpendicular interface slot was used to investigate the effects of interface slot position. Results indicated that placing the interface slot very close to the vanes reduced the size of the horseshoe vortex while subsequently improving the effectiveness of the injected coolant. While moving the interface slot closer to the vanes provided improvement for a perpendicular slot, what effect would this have on an interface slot with a reduced orientation?

Reducing the orientation of the interface slot was shown to drastically improve the cooling of the vane endwall as with high injection flow rates the horseshoe vortex was removed by filling in the low velocity region of the boundary layer near the endwall. Filling in the low velocity region of the boundary layer near the endwall reverses the static pressure gradient at the vane stagnation resulting in upwash at the vane-endwall junction instead of downwash which turns upstream at the endwall and separates to form the horseshoe vortex. At large orientations, however, the injection of coolant was shown to increase the size of the horseshoe vortex relative to the case where no interface slot
was present as the injecting slot coolant resulted in a larger separation region than would be present with no slot injection. Although some inference about the passage secondary flows can be made from the inspection of cooling effectiveness and heat transfer contours, without flowfield measurements the nature of the passage vortex and endwall crossflow can not be characterized. Time-resolved measurements of the flowfields within the passage would provide an additional understanding of the endwall thermal results. Of particular interest would be the behavior of the passage vortex in the case where no horseshoe vortex is present as a result of coolant injection from the interface slot along the endwall. Does the passage vortex still exist? If so, how has the removal of the horseshoe vortex affected the growth of the passage vortex? Further understanding of the influence of the horseshoe vortex on the formation of the passage secondary flows can lead to better design features which allow for improved cooling of the vane endwall.
References


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Appendix A

Comparison of Predicted and Measured Adiabatic Effectiveness Results

Computations were performed using FLUENT to predict and compare to experimental adiabatic effectiveness measurements from both the planar and axisymmetric passage configurations. All simulations were three-dimensional and were run under incompressible, low-speed, viscous conditions. The simulations employed the RNG k-ε turbulence model with non-equilibrium wall functions. The Reynolds Average Navier Stokes (RANS) equations as well as the energy and turbulence equations were solved implicitly using a pressure based solver under steady flow conditions.

The first step in the computation process was constructing a model domain for each passage configuration using Gambit. Figures A.1-.A.3 show the domain used for the planar passage, as well as the domain used to study both the flat and sloped endwalls of the axisymmetric passage respectively. Also shown in these figures are the boundary conditions that were applied to each surface. The vane was divided at the dynamic stagnation point and trailing edge, and a single passage was modeled with periodic boundaries specified in both the leading and trailing planes. Only half of the planar passage was modeled using a symmetry condition at midspan while the entire passage span had to be modeled for the axisymmetric passage configurations. No slip was imposed on the vane surfaces, slot plenum walls, and passage endwalls. The inlet to the domain was specified as a velocity inlet. In the case of the planar passage the velocity inlet was located one chord upstream of the vane. For the axisymmetric configurations the velocity inlet was located one chord upstream of the start of the contraction. An
outflow boundary condition was located 1.5C downstream of the vane trailing for all endwall cases. In addition, a 0.1C extension was added to the exit of each domain to avoid highly skewed cells at the domain exit. The final boundary condition was a mass flow inlet on the bottom of the leakage slot plenum.

**Figure A.1.** Domain and boundary conditions modeled for the planar passage.
Figure A.2. Domain and boundary conditions modeled for the axisymmetric passage with the sloped ceiling.
A tri-pave meshing scheme was used to discretize each domain. The resulting mesh for the planar passage consisted of approximately $2.8 \times 10^6$ tetrahedral cells. Conversely, the axisymmetric configurations consisted of approximately $4.1 \times 10^6$ tetrahedral cells since the entire passage span had to be modeled. Once the domain was created and meshed the entity was imported into FLUENT to begin solving. For this

Figure A.3. Domain and boundary conditions modeled for the axisymmetric passage with the sloped floor.
study a solution was computed for 1000 iterations. The grid was then adapted based upon $y^+$ values as well as temperature and velocity gradients. All cells where $30 \leq y^+ \leq 60$ was not true were marked for adaption. Next the maximum temperature and velocity gradients were calculated and any cells with gradients higher than half of the maximum were marked for adaption. After marking, the grid was adapted using the hanging node method. After adaption the computation was continued for another 1000 iterations. The adaption resulted in a large increase in the residuals during the first few iterations as seen in Figure A.4 which represents a typical residual plot.

![Figure A.4. Typical residual plot.](image)

This process was repeated until the computation reached 4000 iterations and the residuals were no longer changing. To evaluate whether the results were converged, the lift coefficient and area-average endwall effectiveness were monitored after each
iteration. As seen in Figure A.5, which represents a typical convergence plot, these values were relatively constant even after two grid adaptions. This indicated that the solution was converged.

![Convergence Plot](image)

**Figure A.5.** Typical plot of lift coefficient and area-average effectiveness versus the number of iterations.

The first passage configuration to be studied computationally was the planar passage. A comparison of endwall adiabatic effectiveness levels between the prediction and the experimental measurements at different slot leakage flows is shown in Figures A.6-A.9. The predicted adiabatic effectiveness levels showed good agreement with measured values. Both predictions and measurements indicated that coolant coverage increased with leakage flow. In addition, the predicted values showed that the adiabatic
effectiveness directly downstream of the leakage slot is highest in the case of the 0.5% leakage flow.

The next endwall configuration to be considered was the flat endwall of the axisymmetric passage. Adiabatic effectiveness experiments performed with the axisymmetric passage included coolant injection from both the leakage slot and from discrete film cooling holes. For comparison, Figures A.10-A.12 show the predicted adiabatic effectiveness levels in contrast to the measured values at the lowest film cooling rate of 0.3% for a given slot leakage flow. Note that experimental results from the outside vane passage, where the least film cooling holes were present, were used for comparison to the predicted values. The predicted adiabatic effectiveness levels showed fairly good agreement with measured values. The presence of the film cooling holes in the measured values served to increase the local effectiveness levels as well as the coolant coverage.

Finally, the sloped endwall of the axisymmetric passage was investigated. Here again for the purpose of comparison, Figures A.13-A.15 show the predicted effectiveness levels in contrast to the measured values at the lowest film cooling rate for a given leakage flow. With the consideration of film cooling, inspection of Figures A.13-A.15 indicates that there is good agreement between predicted and measured values.
Figure A.6. Comparison between predicted and measured adiabatic effectiveness contours for the planar passage at 0.25% leakage flow.

Figure A.7. Comparison between predicted and measured adiabatic effectiveness contours for the planar passage at 0.5% leakage flow.
Figure A.8. Comparison between predicted and measured adiabatic effectiveness contours for the planar passage at 0.75% leakage flow.

Figure A.9. Comparison between predicted and measured adiabatic effectiveness contours for the planar passage at 1.0% leakage flow.
Figure A.10: Comparison between predicted and measured adiabatic effectiveness contours on the flat endwall of the axisymmetric passage at 0.25% and 0.3% leakage flow respectively.

Figure A.11: Comparison between predicted and measured adiabatic effectiveness contours on the flat endwall of the axisymmetric passage at 0.5% leakage flow.
Figure A.12: Comparison between predicted and measured adiabatic effectiveness contours on the flat endwall of the axisymmetric passage at 0.75% and 0.7% leakage flow respectively.

Figure A.13: Comparison between predicted and measured adiabatic effectiveness contours on the sloped endwall of the axisymmetric passage at 0.25% and 0.3% leakage flow respectively.
Figure A.14: Comparison between predicted and measured adiabatic effectiveness contours on the sloped endwall of the axisymmetric passage at 0.5% leakage flow.

Figure A.15: Comparison between predicted and measured adiabatic effectiveness contours on the sloped endwall of the axisymmetric passage at 0.75% and 0.7% leakage flow respectively.
Appendix B
Calculation of Secondary Flow Vectors

To visualize the secondary flow patterns associated with the HSV a method was used where secondary flow vectors are presented as in plane deviations from the midspan or inviscid streamlines. Fluid flow at the midspan of the passage is assumed as the reference condition because no viscous effects are present. Variations from this reference flow path are plotted as secondary flows. In this dissertation the secondary flow vectors were plotted in the leading-edge, stagnation plane of the vane at the vane-endwall junction. The first step was to calculate the flow angle of the inviscid flow at the midspan for rotation about the z-axis as shown in Figure B.1 at several discrete locations corresponding to the axial position of the vectors shown in Figures 4.5 and 4.14 according to the equation below.

\[
\psi_{inv} = \tan^{-1} \left( \frac{V_{inv}}{U_{inv}} \right) \quad (B.1)
\]

After obtaining an inviscid turning angle at each discrete axial location along the midspan, a transform from the local coordinate system to a coordinate system in the plane of the inviscid streamline was performed. This involved using Equation B.2 below at each discrete location within the plane of interest at the vane leading-edge corresponding to the location of the vectors in Figures 4.5 and 4.14. At each spanwise location the local values of \( U \) and \( V \) were used in addition to the value of \( \psi_{inv} \) that corresponded to the
given axial location to calculate the transformed velocity component. Note that a positive angle of rotation is from the x to y-axis.

\[ V'_s = U \cos \psi^{\text{inv}} + V \sin \psi^{\text{inv}} \]  

(B.2)

**Figure B.1.** Schematic showing the calculation of the inviscid flow angle and the transformed velocity component in the x-y plane for the planar passage.

For the planar passage no further transformation was required since at midspan there is no vertical velocity component, \( W \), due to symmetry. For the contoured passage, however, this method of analysis had to be expanded as symmetry no longer existed. To
obtain the inviscid flow field an additional computation was performed, using the same technique described in Appendix A, but with inviscid flow. The transformation for the contoured passage was then based on the inviscid turning angles in the x-z plane obtained from the inviscid solution at each discrete location corresponding to the position of the vectors in Figures 4.5 and 4.14 all along the stagnation plane under no slot leakage flow conditions according to Equation B.3 and illustrated in Figure B.2. Note that this is different from just obtaining a turning angle at the midspan for given axial locations as in the previous step.

\[ \Phi_{\text{inv}} = \tan^{-1} \frac{W_{\text{inv}}}{V_{s',\text{inv}}} \]  

(B.3)

**Figure B.2:** Schematic showing the calculation of the inviscid flow angle and the transformed velocity components in the x-z plane for the axisymmetric passage.
After obtaining the angle at each discrete location within the stagnation plane, the final transformation took place to transform the previous velocity components to those which are in plane with the inviscid streamlines. The local values of \( V_s', W, \) and \( \Phi_{\text{inv}} \) were used at each discrete location within the leading-edge plane to calculate the velocity components as shown in Equations B.4 and B.5 below and illustrated in Figure B.2. Note that a positive angle of rotation is from the z to x-axis.

\[
V_s = V_s'\cos\Phi_{\text{inv}} - W\sin\Phi_{\text{inv}} \tag{B.4}
\]

\[
V_z = V_s'\sin\Phi_{\text{inv}} + W\cos\Phi_{\text{inv}} \tag{B.5}
\]

The above two velocities were plotted as secondary flow vectors. Note that for the planar passage \( V_z = W \) and \( V_s' = V_s \) as \( \Phi_{\text{inv}} = 0 \).
Appendix C

Uncertainty Calculations

Adiabatic Effectiveness Uncertainty Analysis

Adiabatic effectiveness was calculated according to Equation C.1 below.

\[ \eta_{\text{measured}} = \frac{T_\infty - T_{aw}}{T_\infty - T_c} \]  

An uncertainty analysis was performed on the measurements of adiabatic effectiveness using the partial derivative method. The uncertainty in adiabatic effectiveness was found based on the partial derivative of \( \eta \) with respect to each temperature in the definition and the total uncertainty in the measurements. Uncertainty was calculated from the square root of the sum of the squares of each partial-uncertainty product as shown below in Equation C.2.

\[ \hat{\eta} = \sqrt{ \left( \frac{\partial \eta}{\partial T_\infty} \frac{\partial T_\infty}{\partial \eta} \right)^2 + \left( \frac{\partial \eta}{\partial T_{aw}} \frac{\partial T_{aw}}{\partial \eta} \right)^2 + \left( \frac{\partial \eta}{\partial T_c} \frac{\partial T_c}{\partial \eta} \right)^2 } \]  

where \( \frac{\partial \eta}{\partial T_\infty} = \frac{T_{aw} - T_c}{(T_\infty - T_c)^2} \), \( \frac{\partial \eta}{\partial T_{aw}} = \frac{-1}{T_\infty - T_c} \), \( \frac{\partial \eta}{\partial T_c} = \frac{T_\infty - T_{aw}}{(T_c - T_\infty)^2} \)

Table C.1 lists the uncertainties associated with each temperature and the resulting uncertainty in the measured adiabatic effectiveness for \( \eta = 0.03 \) and 0.9. Note that the
uncertainties in the freestream and coolant temperatures are based on the thermocouple uncertainty. In addition to the thermocouple uncertainty, the uncertainty in the adiabatic wall temperature is based on the average deviation of the calibrated IR images from the thermocouples. As shown in Table C.1 an uncertainty of approximately $\delta \eta = \pm 0.025$ was calculated at $\eta = 0.05$ and 0.9.

**Table C.1.** Uncertainty analysis for the measured adiabatic effectiveness

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
<th>Bias Uncertainty</th>
<th>Precision Uncertainty</th>
<th>Total Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_\infty$ (°C)</td>
<td>50</td>
<td>0.2</td>
<td>0.2</td>
<td>0.28</td>
</tr>
<tr>
<td>$T_{aw}$ (°C)</td>
<td>48.75, 27.5</td>
<td>0.54</td>
<td>0.2</td>
<td>0.58</td>
</tr>
<tr>
<td>$T_c$ (°C)</td>
<td>25</td>
<td>0.2</td>
<td>0.2</td>
<td>0.28</td>
</tr>
<tr>
<td>$\eta$</td>
<td>0.05, 0.9</td>
<td>0.023</td>
<td>0.011</td>
<td>0.025</td>
</tr>
</tbody>
</table>

A one-dimensional correction was applied to the adiabatic effectiveness levels provided by Equation C.1 to compensate for the temperature gradient across the boundary layer due to small conduction losses. The equation for obtaining the corrected adiabatic effectiveness is presented below.

$$\eta_c = \frac{\eta - \eta_o}{1 - \eta_o} \quad \text{(C.3)}$$

Note that the conduction correction value was obtained from an adiabatic effectiveness test with no injection coolant present. The resulting effectiveness contour
from that experiment is shown in Figure C.1a. This contour represents the correction factor that was applied to the measured effectiveness values. Figures C.1b and C.1c present two effectiveness contours for the sloped endwall of the asymmetric passage with 0.7% interface slot flow and 0.5% discrete film cooling hole flow. Figure C.1b is an uncorrected effectiveness contour. Figure C.1c has been corrected using the conduction correction in Figure C.1a and the correction method described by Equation C.3. From the uncorrected contour it can be seen that the effectiveness levels near the vane pressure side, where there is no coolant, are greater than 0. Once the correction is applied, however, these surfaces fall within the 0-0.05 effective range. This range indicates those regions with no coolant coverage. The larger effectiveness values represent the contact of coolant with the endwall surface. These areas are less affected by the correction factor.

**Figure C.1.** (a) Conduction correction contour, (b) uncorrected effectiveness contour with 0.7% injection slot flow and 0.5% discrete hole flow, and (c) corrected effectiveness contour with 0.7% injection slot flow and 0.5% discrete hole flow.
An uncertainty analysis was performed on the corrected adiabatic effectiveness using the partial derivative method. The uncertainty in the corrected adiabatic effectiveness was found based on the partial derivative of $\eta_c$ with respect to the measured adiabatic effectiveness and the conduction correction value, $\eta_o$. Uncertainty was calculated from the square root of the sum of the squares of each partial-uncertainty product as shown below in Equation C.4.

$$\partial \eta_c = \sqrt{\left( \frac{\partial \eta_c}{\partial \eta} \right)^2 + \left( \frac{\partial \eta_c}{\partial \eta_o} \right)^2}$$

(C.4)

where

$$\frac{\partial \eta_c}{\partial \eta} = \frac{-1}{\eta_o - 1}, \quad \frac{\partial \eta_c}{\partial \eta_o} = \frac{\eta - 1}{(\eta_o - 1)^2}$$

Table C.2 lists the uncertainties associated with each measurement of adiabatic effectiveness and the resulting uncertainty in the corrected adiabatic effectiveness for $\eta_c = 0$ and 0.89. As shown in Table C.2 an uncertainty of approximately $\partial \eta_c = \pm 0.026$ was calculated at $\eta_c = 0$ and 0.89.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
<th>Bias Uncertainty</th>
<th>Precision Uncertainty</th>
<th>Total Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta$</td>
<td>0.05, 0.9</td>
<td>0.023</td>
<td>0.011</td>
<td>0.025</td>
</tr>
<tr>
<td>$\eta_o$</td>
<td>0.05</td>
<td>0.023</td>
<td>0.011</td>
<td>0.025</td>
</tr>
<tr>
<td>$\eta_c$</td>
<td>0, 0.89</td>
<td>0.024</td>
<td>0.012</td>
<td>0.026</td>
</tr>
</tbody>
</table>
Heat Transfer Uncertainty Analysis

Heat transfer results were presented as Nusselt Number based on axial chord as shown below in Equation C.5.

\[
Nu = \frac{hC_{ax}}{k_{air}} = \frac{q_{conv}}{(T_{wall} - T_{\infty})k_{air}} C_{ax} = \frac{q_{total} - q_{cnd} - q_{rad}}{(T_{wall} - T_{\infty})k_{air}} C_{ax}
\]

The total power put into the heater was not totally convected to the freestream. Some heat was lost to the room in terms of conduction and some was lost to the surroundings as radiation. The percentage of the radiation heat flux loss was typically between 6-18% as shown in Figure C.2a. The percentage of conduction heat flux losses was typically between 0.6-1.5% as shown in Figure C.3a. Note that a percentage range is given becau-

Figure C.2. (a) Heat loss percentage through conduction and (b) heat loss percentage through radiation.
-se the heat flux loss depends on the local heat transfer. Areas of lower heat transfer, higher temperatures, experience more heat flux loss through both conduction and radiation due to a larger driving temperature difference.

An uncertainty analysis was performed on the measurements of heat transfer using the partial derivative method. The uncertainty in Nusselt number was found based on the partial derivative of Nusselt Number with respect to each term in the definition and the total uncertainty in the measurements. For simplicity, the uncertainty was calculated by breaking down each term into the basic measured quantities. The Nusselt number is a function of the heat transfer coefficient and the thermal conductivity of the air. Equation C.6 provides the formulation for the Nusselt number uncertainty.

\[
\partial \text{Nu} = \sqrt{\left( \frac{\partial \text{Nu}}{\partial h} \right)^2 + \left( \frac{\partial \text{Nu}}{\partial k} \right)^2} \tag{C.6}
\]

The thermal conductivity of the air is a function of the temperature. This is a base measured quantity so no further decomposition is required for this term. However, the heat transfer coefficient term must be further broken down. This is shown below in Equation C.7.

\[
\partial h = \sqrt{\left( \frac{\partial h}{\partial T_\infty} \partial T_\infty \right)^2 + \left( \frac{\partial h}{\partial T_w} \partial T_w \right)^2 + \left( \frac{\partial h}{\partial q_{\text{conv}}} \partial q_{\text{conv}} \right)^2} \tag{C.7}
\]
The uncertainty associated with the freestream temperature arises from the uncertainty in the thermocouple measurements. The wall temperature uncertainty is a function of both thermocouple uncertainty and the uncertainty introduced by the IR camera. The convection heat flux is a function of the total supplied heat flux as well as the heat loss through conduction and radiation. The uncertainty of the convection heat loss is found according to Equation C.8.

\[
\delta q_{\text{conv}} = \sqrt{\left( \frac{\partial q_{\text{conv}}}{\partial q_{\text{total}}} \delta q_{\text{total}} \right)^2 + \left( \frac{\partial q_{\text{conv}}}{\partial q_{\text{cnd}}} \delta q_{\text{cnd}} \right)^2 + \left( \frac{\partial q_{\text{conv}}}{\partial q_{\text{rad}}} \delta q_{\text{rad}} \right)^2}
\]  

(C.8)

The final set of decompositions is performed on each heat flux term making up the convection heat flux. The formulations for these three heat fluxes are presented below in Equations C.9-C.11. Note that the conduction accounts for the heat loss through the different layers between the heater and the ambient conditions while the radiation heat loss is between the heater and the inside surroundings of the test section.

\[
q_{\text{total}} = \frac{V_{\text{heater}} I_{\text{heater}}}{A_{\text{endwall}}}
\]  

(C.9)

\[
q_{\text{cnd}} = \frac{T_w - T_{\text{amb}}}{\frac{L_{\text{foam}}}{k_{\text{foam}}} + \frac{L_{\text{MDF}}}{k_{\text{MDF}}} + \frac{L_{\text{board}}}{k_{\text{board}}} + \frac{1}{h_{\text{amb}}}}
\]  

(C.10)
The uncertainty for each heat flux is found according to Equations C.12-C.14 below.

\[
\dot{q}_{\text{rad}}'' = \varepsilon \sigma (T_w^4 - T_\infty^4)
\]  
\(\text{(C.11)}\)

\[
\partial \dot{q}_{\text{total}}'' = \sqrt{\left(\frac{\partial \dot{q}_{\text{total}}''}{\partial \text{V}_{\text{heater}}} \partial \text{V}_{\text{heater}}\right)^2 + \left(\frac{\partial \dot{q}_{\text{total}}''}{\partial \text{I}_{\text{heater}}} \partial \text{I}_{\text{heater}}\right)^2}
\]  
\(\text{(C.12)}\)

\[
\partial \dot{q}_{\text{end}}'' = \sqrt{\left(\frac{\partial \dot{q}_{\text{end}}''}{\partial T_w} \partial T_w\right)^2 + \left(\frac{\partial \dot{q}_{\text{end}}''}{\partial T_{\text{amb}}} \partial T_{\text{amb}}\right)^2}
\]  
\(\text{(C.13)}\)

\[
\partial \dot{q}_{\text{rad}}'' = \sqrt{\left(\frac{\partial \dot{q}_{\text{rad}}''}{\partial T_w} \partial T_w\right)^2 + \left(\frac{\partial \dot{q}_{\text{rad}}''}{\partial T_\infty} \partial T_\infty\right)^2}
\]  
\(\text{(C.14)}\)

Note that heat flux uncertainties are now functions of the base measured quantities \(T_w, T_\infty, T_{\text{amb}}, \text{V}_{\text{heater}}, \) and \(\text{I}_{\text{heater}}\). The total uncertainty in Nusselt number is calculated by substituting Equations C.7, C.8, and C.12-C.14 into Equation C.6. Using the known values and uncertainties of the measured questions a total uncertainty can be obtained as shown in Table C.3. An uncertainty of approximately \(\partial \text{Nu} = \pm 6(2.5\%)\) and \(\pm 96(8\%)\) at \(\text{Nu} = 250\) and 1200 respectively.
Table C.3. Uncertainty analysis for Nusselt Number

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
<th>Bias Uncertainty</th>
<th>Precision Uncertainty</th>
<th>Total Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_w$ (°C)</td>
<td>38, 54</td>
<td>0.54</td>
<td>0.2</td>
<td>0.58</td>
</tr>
<tr>
<td>$T_\infty$ (°C)</td>
<td>28</td>
<td>0.2</td>
<td>0.2</td>
<td>0.28</td>
</tr>
<tr>
<td>$T_{amb}$ (°C)</td>
<td>25</td>
<td>0.2</td>
<td>0.2</td>
<td>0.28</td>
</tr>
<tr>
<td>$V_{heater}$ (V)</td>
<td>223</td>
<td>0.22 (0.1%)</td>
<td>0</td>
<td>0.22</td>
</tr>
<tr>
<td>$I_{heater}$ (A)</td>
<td>1.4</td>
<td>0.0014 (0.1%)</td>
<td>0</td>
<td>0.0014</td>
</tr>
<tr>
<td>Nu</td>
<td>250, 1200</td>
<td>5.6, 86</td>
<td>2.3, 42</td>
<td>6, 96</td>
</tr>
</tbody>
</table>
Particle Image Velocimetry Uncertainty Analysis

Uncertainty in the PIV velocity measurements was calculated based on the average particle displacement, maximum displacement gradient, average particle image density, and signal-to-noise ratio. Raffel et al. [57] provides several plots to assess the measurement precision in PIV based on numerical simulation. The simulations are termed Monte Carlo simulations as a single parameter is varied at a time to evaluate the resulting effect. Figures C.3-C.7 shows the simulation results for both RMS and bias measurement uncertainty as a function of average particle displacement, average particle image density, signal-to-noise-ratio, and maximum displacement gradient.

![Figure C.3](image.png)

**Figure C.3.** RMS measurement uncertainty in digital cross-correlation PIV evaluation as a function of particle image displacement [57].
Figure C.4. Bias measurement uncertainty in digital cross-correlation PIV evaluation as a function of particle image displacement [57].

Figure C.5. RMS measurement uncertainty in PIV evaluation as a function of particle image displacement for various particle image densities [57].
Figure C.6. RMS measurement uncertainty in PIV evaluation as a function of particle image displacement and various amounts of white background noise [57].

Figure C.7. RMS measurement uncertainty in PIV evaluation as a function of displacement gradient for various particle image densities and interrogation window sizes [57].
Based on the given values from the current PIV experiments the uncertainty in the instantaneous velocity measurements could be evaluated using Figures C.3-C.7 as shown below in Table C.4. The uncertainty in the instantaneous velocity measurement was found to be approximately $\partial U = 0.16\text{m/s (2\% of } U_\infty)$ for a bulk particle displacement of 8 pixels.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
<th>Bias Uncertainty (pixels)</th>
<th>Precision Uncertainty (pixels)</th>
<th>Total Uncertainty (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Particle Image Displacement</td>
<td>8 pixels</td>
<td>0.005</td>
<td>0.01</td>
<td>0.011</td>
</tr>
<tr>
<td>Particle Image Density</td>
<td>32 particles/window</td>
<td>N/A</td>
<td>0.015</td>
<td>0.015</td>
</tr>
<tr>
<td>Background Noise</td>
<td>5%</td>
<td>N/A</td>
<td>0.025</td>
<td>0.025</td>
</tr>
<tr>
<td>Displacement Gradient</td>
<td>0.06 pixel/pixel</td>
<td>N/A</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>Instantaneous Velocity, U</td>
<td>8 m/s ($U_\infty$)</td>
<td>0.005</td>
<td>0.15</td>
<td>0.16</td>
</tr>
</tbody>
</table>

Table C.4. Uncertainty analysis for PIV with a bulk particle displacement of 8 pixels.
Vita

ALAN A. THRIFT

Alan Thrift graduated from the University of North Carolina at Charlotte (UNCC) with his B.S. in Mechanical Engineering in 2005. During his junior year at UNCC he worked as a student undergraduate research fellow at the National Institute of Standards and Technology (NIST) in Gaithersburg, Maryland. At NIST, he designed a high load capacity straining jig for use in the NIST Center for Neutron Research. After graduation, Alan worked as an engineer at Belford Research Inc. in Bluffton, South Carolina over the summer of 2005. While working at Belford Research Inc., he helped perform research related to the mechanical straining of silicone semiconductor chips to improve electrical performance. Alan’s work was published in his first paper as a co-author in the Journal of Applied Physics.

Alan began graduate school in 2005 at Virginia Polytechnic Institute and State University (VT) in the Virginia Tech Experimental and Computational Convection Lab (VTExCCL) under the mentorship of Dr. Karen Thole. For his master’s thesis (2007), *Aerodynamic Force and Pressure Loss Measurements on Low Aspect Ratio Pin Fin Arrays*, Alan developed a unique methodology to perform force measurements on wall bounded obstructions. The project was funded by Pratt & Whitney as an initiative to evaluate the aerodynamic performance of advance cooling technologies utilized in gas turbine blades. Alan presented results from his pressure loss work at the 2007 ASME Turbo Expo conference in Montreal, Canada and was published as a co-author for his contributing work in two separate papers in the *Journal of Turbomachinery* and *International Journal of Heat and Mass Transfer* in 2011. Alan was published as a lead author for his force measurement work in the *Journal of Fluids Engineering* in 2010.

Alan graduated from VT in 2007 with his M.S. in Mechanical Engineering. He joined Dr. Karen Thole at The Pennsylvania State University (PSU) at University Park in 2007 to continue his work with gas turbine engines in the Penn State University Experimental and Computation Convection Laboratory (PSUExCCL) as a doctoral student. Alan’s doctoral work was funded by Mitsubishi Heavy Industries (MHI) and was concerned with the cooling of a first stage, nozzle guide vane endwall for land based gas turbines. The work from his dissertation has been presented at both the 2010 and 2011 ASME Turbo Expo conferences in Glasgow, Scotland and Vancouver, Canada respectively. He is the lead author on three separate publications from his presented work at the conferences in the *Journal of Turbomachinery*. Alan is lead author on two final papers from his dissertation work which have been submitted and are awaiting review for acceptance to the *Journal of Turbomachinery* and *International Journal of Heat and Mass Transfer*. After completing his dissertation, Alan has accepted a position as a mechanical engineer with Siemens Energy Inc. in Charlotte, North Carolina. Alan will continue to work on the cooling of turbine components in land based, power generation gas turbines.