THE EFFECT OF COMBUSTOR-TURBINE INTERFACE GAP LEAKAGE ON THE ENDWALL HEAT TRANSFER FOR A NOZZLE GUIDE VANE

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ABSTRACT
To enable turbine components to withstand high combustion temperatures, they are cooled by air routed from the compressor, which can leak through gaps between components. These gaps vary in size from thermal expansions that take place. The leakage flow between the interface of the combustor and the turbine, in particular, interacts with the flowfield along the endwall.

This study presents measurements of adiabatic cooling effectiveness and heat transfer coefficients on the endwall of a first vane, with the presence of leakage flow through a flush slot upstream of the vane. The effect of axial contraction of the slot width due to thermal expansion of the engine was tested for two blowing rates. Contracting the slot width, while maintaining the slot mass flow, resulted in a larger coolant coverage area and higher effectiveness values, as well as slightly lower heat transfer coefficients. Matching the momentum flux ratio of the leakage flow from the nominal and contracted slot widths lowered both cooling effectiveness and heat transfer coefficients for the contracted slot flow. Comparison of the coolant coverage pattern to the measured endwall shear stress topology indicated that the trajectory of the slot coolant was dictated by the complex endwall flow.

INTRODUCTION
To achieve high efficiency and maximum power output, the temperature of the combustion gases entering the turbine section of a gas turbine engine must be as high as possible, while acknowledging the limitations of material strength and durability. This subjects the turbine section components to high heat loads which must be managed by cooling, since turbine inlet temperatures are generally well above the melting point of the metal. Relatively cool air is bled from the compressor and routed to the turbine components, where it provides internal and external cooling. Cooling is particularly important on the endwall of a turbine vane or blade, since the complex flow in that region results in high heat transfer rates, and also tends to sweep coolant away from the endwall surface.

Assembly of individual turbine components inherently results in gaps between parts. The large operational range of a gas turbine results in significant thermal expansion, making these gaps difficult to seal. Furthermore, the temperature profile exiting the combustor may not be uniform, leading to additional thermal expansion issues. Since leakage of the hot combustion gases into the gaps is detrimental to engine durability, high-pressure compressor bleed air is purged through the gaps. Purge flow, however, results in a loss in turbine efficiency since it does no useful work, and it is desirable to minimize this flow.

One gap that must be considered is between the combustor and the first stage of the turbine section. In addition to preventing hot gas ingestion, purge flow through the combustor-turbine interface gap can provide some cooling to the endwall of the turbine vane. However, the trajectory of this coolant is influenced by the complex endwall flowfield. Thermal expansion of the combustor-turbine interface gap during engine operational cycling also influences purge flow and changes the heat transfer experienced by the endwall.

This paper discusses the effects of leakage flow on turbine vane endwall heat transfer, for a combustor-turbine interface gap with axial thermal expansion. Also considered is the effect of moving the interface far upstream of the turbine vanes.

NOMENCLATURE

- C: true chord of stator vane
- Cx: axial chord of stator vane
- Cf: friction coefficient, see Eq. (1)
- Cp: heat capacity at constant pressure
- g: gravitational acceleration (9.81 m/s²)
- h: heat transfer coefficient
- hoil: height of oil film
- I: average momentum flux ratio, \( I = \frac{\rho_c U_c^2}{\rho_x U_{\infty, \text{in}}} \)
The axial location of an upstream slot has also been shown to be important in the interaction of slot flow with secondary vortices. Kost and Nicklas [13] and Nicklas [14] presented aerodynamic and heat transfer measurements for upstream slot and passage discrete hole film cooling in a transonic cascade. For slot coolant ejection at 1.3% of the core flow rate and no passage film cooling, they found that the slot flow intensified the horseshoe vortex and increased heat transfer coefficients by nearly three times that of no slot flow. They attributed this dramatic increase to the fact that their slot, at 0.2cax upstream of the vane, was intensifying the horseshoe vortex by injecting at the separation location on the endwall. Kost and Mullaert [15] studied the same airfoil geometry, but moved the flush slot

where the pressure side leg develops into a passage vortex. These vortical structures and their interaction (generally termed “secondary flows”) are sources of aerodynamic loss in the cascade; furthermore, they sweep coolant from the endwall and increase endwall heat transfer coefficients.

Graziani et al. [4], Kang et al. [5], and Ames et al. [6] presented results of endwall heat transfer influenced by secondary flows. Measurements of heat transfer coefficients indicated regions of high heat transfer at the blade leading edge, and adjacent to the suction side of the airfoil downstream of the passage throat. All of the investigators attributed the high heat transfer coefficients to the effects of the horseshoe and passage vortices. Goldstein and Spores [3] used naphthalene mass transfer to infer heat transfer coefficients and deduced the existence of additional corner vortices along the suction side of the vane.

Coolant leakage was investigated by Blair [7] who presented endwall heat transfer from flow through a two-dimensional slot upstream of a linear cascade. Measured endwall heat transfer coefficients were similar with and without the additional slot flow. However, measurements of adiabatic effectiveness over a range of blowing ratios revealed that coolant accumulated along the suction side of the vane in the passage. Granier and Schuemberg [8] and Knost and Thole [9] reported similar trends of coolant accumulation.

Computational predictions by Knost and Thole [10] indicated that streamline paths near the endwall became increasingly more directed toward the pressure side of the vane as flow rates increased through their upstream slot configuration.

Studies have been completed with upstream leakage in a contoured endwall vane cascade, where the contour serves to accelerate the flow and thin the boundary layer with the aim of reducing secondary flows. Burd et al. [11] found that coolant effectiveness levels and coolant coverage were increased for higher coolant ejection rates from an upstream flush slot on their contoured endwall. They conjectured that the momentum of the slot coolant at higher flow rates enabled it to avoid being entrained into the secondary flow vortices.

The only known study of thermal expansion of the combustor-turbine gap in the axial direction is Cardwell et al. [12]. Cardwell et al. found that adiabatic cooling effectiveness coverage area was a function of slot momentum flux ratio, but effectiveness levels were dependent on the slot mass flow ratio.

The axial location of an upstream slot has also been shown to be important in the interaction of slot flow with secondary vortices. Kost and Nicklas [13] and Nicklas [14] presented aerodynamic and heat transfer measurements for upstream slot and passage discrete hole film cooling in a transonic cascade. For slot coolant ejection at 1.3% of the core flow rate and no passage film cooling, they found that the slot flow intensified the horseshoe vortex and increased heat transfer coefficients by nearly three times that of no slot flow. They attributed this dramatic increase to the fact that their slot, at 0.2cax upstream of the vane, was intensifying the horseshoe vortex by injecting at the separation location on the endwall. Kost and Mullaert [15] studied the same airfoil geometry, but moved the flush slot

Greek
\( \delta \) boundary layer thickness
\( \eta \) adiabatic effectiveness, \( \eta = (T_x - T_{aw})/(T_x - T_c) \)
\( \theta \) momentum thickness
\( \theta_i \) incident light angle
\( \lambda \) light wavelength of sodium vapor lamp
\( \mu_{oil} \) dynamic viscosity of oil
\( \nu \) kinematic viscosity
\( \rho \) density
\( \tau_{wall} \) wall shear stress
\( \phi \) phase difference of light rays, or non-dimensional vane metal temperature, \( \phi = (T_x - T_c)/(T_x - T_w) \)

Subscripts
\( \theta \) baseline conditions (no blowing)
aw adiabatic wall
c coolant conditions
in inlet conditions
s streamline coordinate
w conductive (metal) wall
\( \infty \) local freestream conditions

RELEVANT PAST STUDIES

The endwall flowfield of a first stage turbine vane consists of unique features that contribute to high heat transfer and aerodynamic losses. Although there are slight differences in detail, flowfield studies by Langston et al. [1], Sharma and Butler [2], Goldstein and Spores [3], and others concur on the dominant structures in the endwall region for an approach flow that is uniform with a two-dimensional boundary layer. The incoming boundary layer on the endwall rolls up into a horseshoe vortex at the leading edge of the vane. The horseshoe vortex splits into suction and pressure side legs
to 0.3C\textsubscript{ax} upstream of the vane. They found that for this configuration, slot flow stayed closer to the endwall and provided better cooling than flow from the slot located at 0.2C\textsubscript{ax} upstream of the vane.

The study reported in our paper expands upon the work of Cardwell et al. [12] by including heat transfer coefficient and shear stress measurements. It is also important to note that the slot locations relative to the vane differ between this study and that of Cardwell et al. [12], which will be addressed.

**EXPERIMENTAL FACILITY AND METHODOLOGY**

Endwall friction coefficients, adiabatic effectiveness, and heat transfer coefficients were measured in a closed-loop, low-speed wind tunnel, depicted in Figure 1 and previously described by Cardwell et al. [12]. The flow was driven by a 50-hp axial fan, and passed through a heat exchanger. A porous plate with 25% open area diverted flow into the secondary path from the primary flow path. Note that the tunnel has upper and lower secondary flow paths; only the top secondary flow path was used in this study. For adiabatic effectiveness measurements, the primary path was heated with a 55kW electrical resistance heater bank to increase the flow temperature to 45°C. The secondary path was further cooled with a heat exchanger to 20°C to increase the temperature differential between the mainstream and coolant leakage flow. To measure the convective heat transfer coefficients, the coolant and mainstream temperatures were matched with no primary flow heating or additional secondary flow cooling needed.

Downstream of the heat exchanger shown in Figure 1, the primary flow next passed through several screens and a contraction section. The contraction, located 2.9 vane chords upstream of the vane, reduced the flow area from 1.11 m\textsuperscript{2} to 0.62 m\textsuperscript{2} through symmetric 45° bounding walls. The primary flow area remained constant up to the corner test section.

The corner 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 vane design was a three-dimensional extrusion of a two-dimensional midspan airfoil geometry. The vanes were scaled up by a factor of nine to allow for high measurement resolution. The vanes were manufactured from low-density closed cell polyurethane foam for low conductivity. A description of the turbine vane parameters is given in Table 1.

The boundary layer entering the cascade was measured at a location 0.63C upstream of the vane stagnation. Table 2 lists the turbulent inlet boundary layer parameters, which were maintained throughout this study. The measured inlet turbulence intensity and length scales were 0.7% and 4 cm (0.07C\textsubscript{ax}), respectively. Although the turbulence intensity is lower than that typically found in an engine, the effect of turbulence was considered to be of secondary interest and was not examined in this study.

To simulate the leakage interface between a combustor and a turbine, a two-dimensional slot was placed upstream of the turbine vane bottom endwall, as illustrated in Figure 2. This combustor-turbine interface gap was based on the geometry presented by Cardwell et al. [12], and will be referred to as an upstream slot in this paper. Cardwell et al. [12] based their upstream slot design on discussions with several industrial contacts. The primary difference in the slot geometry between our study and that of Cardwell et al. [12] is the location of the slot relative to the vane stagnation (0.38C\textsubscript{ax} for Cardwell et al. [12] compared to 0.96C\textsubscript{ax} for this study). The much larger upstream placement of the slot for this study was necessary to accommodate other endwall-airfoil geometries.

Two upstream slot widths were chosen to model the combustor-turbine interface. The nominal metering width, at a scale that is nine times that of the engine, was 1.43 cm (0.024C\textsubscript{ax}) with a flow length-to-width ratio of 1.9. The contracted slot metering width was 50% of the nominal slot width, and will be referred to as the half slot. The distance of the slot centerline to the vane stagnation was maintained for both slot widths. Both slots had an injection angle of 45° relative to the endwall.

Coolant was extracted from the top secondary channel of the wind tunnel by a 2-hp blower and fed to a plenum located below the upstream slot. To calculate the average mass flow exiting the slot, the inviscid blowing ratio from the slot was

![Figure 1. Large low-speed wind tunnel with separate flow conditioning paths and corner test section.](image-url)

<table>
<thead>
<tr>
<th>Table 1. Vane Geometry and Flow Conditions</th>
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<tbody>
<tr>
<td>Scaling factor</td>
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<td>Scaled vane chord (C)</td>
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<tr>
<td>Axial chord/chord (C\textsubscript{ax}/C)</td>
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<tr>
<td>Pitch/chord (P/C)</td>
</tr>
<tr>
<td>Span/chord (S/C)</td>
</tr>
<tr>
<td>Inlet Reynolds number (Re\textsubscript{in})</td>
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<tr>
<td>Inlet mainstream velocity (U\textsubscript{\textit{\infty}})</td>
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<tr>
<td>Inlet, exit angle</td>
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<tr>
<td>Inlet, exit Mach number</td>
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<tr>
<th>Table 2. Inlet Boundary Layer Characteristics</th>
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<td>Boundary layer thickness/span (δ/S)</td>
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<tr>
<td>Displacement thickness/span (δ\textsuperscript{\textit{\theta}}/S)</td>
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<tr>
<td>Momentum thickness/span (θ/S)</td>
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<tr>
<td>Shape factor (δ\textsuperscript{\textit{\theta}}/θ)</td>
</tr>
<tr>
<td>Momentum thickness Reynolds number (Re\textsuperscript{\textit{\theta}})</td>
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multiplied by an assumed discharge coefficient of 0.6, which is the commonly-accepted value for flow from a sharp-edged orifice [16]. Note that the coolant-to-mainstream density ratio was fixed at 1.1 for all adiabatic cooling effectiveness measurements. The density ratio in this study is lower than that typically found in an engine; however, past film cooling research has indicated that jet behavior best scales with momentum flux ratio. The leakage mass flow ratios and nominal slot metering width for this study were selected based on input from industrial contacts, so that the blowing and momentum flow ratios are expected to be representative of engine values. For the convective heat transfer coefficient measurements, the coolant was maintained to within 0.1°C of the mainstream temperature, resulting in a density ratio of 1.0. Table 3 lists the coolant settings investigated for this paper. Mass flow from the upstream slot is reported as a percentage of the mass flow entering a single vane passage.

Endwall Friction Coefficient Measurements

Endwall friction coefficient measurements were made using oil film interferometry; see the review by Naughton and Sheplak [17], and implementation of the method in linear cascades by Harrison [18] and Holley and Langston [19]. Oil film interferometry (OFI) is based upon the dynamic behavior of a thin oil film. OFI is similar in concept to oil flow visualization, except that the oil layer is extremely thin (on the order of visible light wavelengths), and the height of the oil film is quantified to enable calculation of wall friction coefficients. The equation used in this study to determine the friction coefficient is:

$$C_f = \frac{L}{2h_{oil}} \int_0^\infty \rho U_{\infty,in}^2 dt$$

(1)

To derive Eq. (1), a mass and momentum balance is performed on a differential control volume aligned with the endwall (limiting) streamline. The mass balance gives the height of the oil and its average convective velocity as functions of time and distance along the streamline. Based on an order-of-magnitude analysis for the oil film, the streamwise momentum can be simplified by recognizing that the Reynolds number for the oil film is much less than one (thus the inertial terms are negligible). Then, the momentum equation can be solved for the oil velocity (retaining the pressure gradient, gravity, and shear terms) by applying no-slip boundary conditions at the wall-oil interface, and the desired shear at the air-oil interface. Combining the momentum and mass balance results yields the thin oil-film equation:

$$\frac{\partial h_{oil}}{\partial t} + \frac{\partial}{\partial s} \left( \frac{\tau_{w,s} h_{oil}^5}{2 \rho_{oil}} - \frac{h_{oil}^3}{3 \rho_{oil}} \left( \frac{\partial p}{\partial s} - \rho_{oil}g_s \right) \right) = 0$$

(2)

For representative values pertinent to this study (h_{oil}=1 \mu m, \nu_{oil}=100 cSt, \tau_{w,s}=10 Pa, \rho_{oil}=1000 kg/m^3, dP/ds=100 Pa/m, g_s=10 m/s^2), order-of-magnitude analysis on the terms in the parentheses of Eq. (2) shows that the shear stress term is at least two orders of magnitude larger than the other terms, which are then neglected. For spatially constant shear stress over the region of interest (reasonable assumptions in this study because of the large scale and small measurement sizes), the reduced oil film equation can be solved by separation of variables for the height of the oil in terms of the shear stress, distance along the streamline, and time. Note that the shear stress is non-dimensionalized by the inlet dynamic pressure before performing separation of variables, so that the dynamic pressure appears in the integral in Eq. (1). The solution requires measurement of the oil height only at the end of the wind tunnel run, since the conditions leading to the final oil film thickness are integrated over time.

Fizeau interferometry provides a means of measuring the height of the oil. Light strikes the surface of the oil film and is reflected and refracted. The phase difference (\phi) between the initially reflected and refracted rays will attenuate or augment the rays, creating interference bands (fringes). The phase difference is related to the height of the oil by the wavelength of the light rays, the optical properties of air and oil, and the incident light angle:

$$h_{oil} = \frac{\lambda_0}{4\pi} \left( \frac{1}{\sqrt{\nu_{oil}^2 - n_{air}^2 \sin^2 \theta_i}} \right)$$

(3)

The phase difference between successive fringes is an integer multiple of 2\pi. The average spacing between successive fringes (L) is determined by a least-squares cosine fit to the pixel intensity profile of the fringe pattern [19] to determine the period of the intensity profile. The location of

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### Table 3. Slot Coolant Settings

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<tr>
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<th>MFR</th>
<th>M</th>
<th>I</th>
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<tbody>
<tr>
<td>Nominal width</td>
<td>1.0%</td>
<td>0.36</td>
<td>0.13</td>
</tr>
<tr>
<td></td>
<td>0.5%</td>
<td>0.19</td>
<td>0.03</td>
</tr>
<tr>
<td>Half slot</td>
<td>1.0%</td>
<td>0.73</td>
<td>0.50</td>
</tr>
<tr>
<td></td>
<td>0.5%</td>
<td>0.36</td>
<td>0.13</td>
</tr>
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</table>

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![Figure 2. A schematic of the endwall and the combustor-turbine leakage interface (upstream slot) modeled in this study.](image-url)
the friction coefficient measurement in the interferogram is taken as the center of the intensity profile.

To measure endwall friction coefficients in the cascade, small silicone oil droplets, nominally less than 5 mm in diameter, were placed on 0.05 mm thick rectangular sheets of nickel foil that had been adhered to the endwall (see Figure 3). For the entire endwall, three viscosities of silicone oil (100 cSt, 500 cSt, and 1000 cSt) were used to maximize the range of the OFI method throughout the vane passage. The oil viscosity was corrected for temperature variation during a test (generally less than 4°C) by the correlation of Naughton and Sheplak [17]. The time history of the cascade inlet dynamic pressure and the flow temperature were recorded over the entirety of a test, which nominally took twenty minutes. The foil patches were carefully removed after a test and imaged in a fixture with a nearly monochromatic sodium vapor lamp ($\lambda$=589, 589.6 nm) to obtain the interferograms. Several tests were required to obtain over 400 data points on the endwall.

Shear stress vectors were obtained by examining the progression of features in the fringe pattern. For example, a notch in a fringe will propagate downstream in the direction of the limiting streamline. Note, however, that measurements were only obtained on the portion of a fringe that was undisturbed by dust particle wakes or upstream oil droplets.

The OFI method was benchmarked by measuring friction factors in fully developed flow in a square channel. Friction factor measurements agreed with textbook correlations to within 5% over a wide range of channel Reynolds numbers (25,000 < ReDh < 45,000).

The partial derivative method described by Moffat [20] was used to calculate uncertainties for the measurements of friction coefficients. Uncertainty was estimated as $\partial C_f = \pm 5.8\%$ for both low ($C_f=0.004$) and high ($C_f=0.126$) values of friction coefficients, and was dominated by uncertainty in the interferogram spacing.

**Adiabatic Effectiveness Measurements**

Adiabatic wall temperatures for endwall effectiveness of upstream slot coolant were obtained from infrared thermography measurements of the bottom endwall surface. The endwall was manufactured from a 2.54 cm thick sheet low-density closed-cell polyurethane foam, which has a low thermal conductivity (0.0287 W/m-K) to minimize conduction errors. The endwall was instrumented with type-E thermocouples throughout the vane passages for calibration of infrared thermography images. The endwall and thermocouples were painted with flat black paint, which has a nominal emissivity of 0.96 and enabled good resolution of surface temperatures with the infrared camera. Infrared-reflective finishing nails were used as image transformation markers.

An infrared camera was used to capture spatially-resolved surface temperatures on the bottom endwall. Based on an uncertainty analysis, five images were taken at each location and averaged, where each image is also an average of 16 frames taken by the camera. The camera’s spatial integration was approximately 0.17 mm (0.0012C). 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. The thermocouples and the calibrated images generally agreed to within 0.3°C. An in-house MATLAB routine was developed to assemble the individual images into a single endwall map.

A one-dimensional correction for endwall conduction effects was applied to all adiabatic effectiveness measurements. The resulting $\eta$ correction was 0.14 at $\eta=0.75$ measured downstream of the slot, and 0.05 for $\eta=0.07$ measured at the exit of the vane passage.

The partial derivative method was used to calculate uncertainties for the measurements of adiabatic cooling effectiveness. For adiabatic surface temperatures, a precision uncertainty of $\pm 0.33^\circ$C was determined by taking the standard deviation of six measurement sets of IR camera images, with each set consisting of five images. Since the IR images were calibrated as closely as possible to the thermocouples in each image, the bias uncertainty for an image was assumed to be 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 ($\pm 0.3^\circ$C). In this way, a bias uncertainty of $\pm 0.39^\circ$C was determined. Using the bias and precision uncertainties, a total uncertainty of $\pm 0.51^\circ$C was estimated for the IR surface temperature measurements. Overall uncertainty in adiabatic effectiveness was calculated to be $\partial \eta = \pm 0.036$ at a $\eta$ value of 0.1, and $\partial \eta = \pm 0.026$ at a $\eta$ value of 0.8.

**Heat Transfer Measurements**

Endwall heat transfer measurements were taken by imaging surface temperatures on a constant heat flux plate attached to a 2.54 cm thick sheet of the closed-cell polyurethane foam. The heat flux plate consisted of a 37 µm copper layer on top of a 75 µm thick kapton layer, in which 25 µm inconel elements were embedded in a serpentine pattern. The heater covered the entire endwall, from immediately
downstream of the slot to 0.3C downstream of the vane trailing edge, as shown in Figure 2. E-type thermocouples, embedded in the foam endwall, were placed in thermal contact with the bottom surface of the heater by thermal cement. A conduction bias between the bottom-mounted thermocouple and the infrared top-surface measurement was accounted for by a one-dimensional calculation of the thermal resistance of the heater. The infrared camera was also used to capture surface temperatures on the heat flux plate.

The input heat flux to the endwall was calculated by measuring the voltage across the heater circuit, as well as the voltage across a precision resistor (1Ω) in series with the circuit, which gave the current. This flux was corrected for conduction and radiation losses, which accounted for a maximum of 0.2% and 21% of the input power, respectively. Note that the conduction and radiation corrections varied locally with the highest correction occurring at the highest surface temperatures. Uncertainty in Stanton numbers was dominated by the uncertainty in surface temperature measurements. For those measurements, a precision uncertainty of ±0.22°C was estimated from the standard deviation of six IR image measurement sets, and a bias uncertainty of ±0.88°C was determined in the same way as for the adiabatic effectiveness measurements. Overall uncertainty in Stanton numbers was \( \delta St = \pm 0.00011 \) (3.3%) at a St value of 0.003, and \( \delta St = \pm 0.0009 \) (7.5%) at a St value of 0.011.

**DISCUSSION OF RESULTS**

Endwall friction coefficient measurements without any upstream slot flow over the endwall will be presented first, followed by adiabatic cooling effectiveness results with upstream slot flow. Heat transfer coefficient measurements with and without upstream slot flow are then discussed. Finally, the net heat flux reduction parameter, which incorporates both heat transfer coefficients and adiabatic cooling effectiveness, is presented.

**Friction Coefficients without an Upstream Slot**

Oil film interferometry (OFI) measurements of endwall friction coefficients were linearly interpolated to a uniform grid, and then downsampled to create the vector plot in Figure 4. Several unique features due to secondary flow are visible in the vector field. Shear decreases as flow approaches the vane leading edge and begins to stagnate. A saddle point is also visible upstream of the vane, where flow diverges around the leading edge. The saddle point is the intersection of flow attachment and separation lines on the endwall. At the separation line, the incoming boundary layer separates from the endwall and rolls up into the horseshoe vortex. The vortex roll-up causes flow to wash down the airfoil toward the endwall and then move upstream, away from the airfoil-endwall junction.

Further in the passage, cross-passage flow from the passage vortex sweeps from the pressure side of the lower airfoil to the suction side on the adjacent airfoil. Because of this, the exit turning angle of the airfoil near the endwall is much larger than the exit turning angle of inviscid flow away from the wall. The measured friction coefficient magnitudes near the passage throat have increased relative to the inlet values, resulting from the flow acceleration in the vane passage.

Figure 5 shows a prediction of inviscid streamlines near the mid-span of the vane using FLUENT 6.2 [21], which are
superimposed upon the endwall streamlines computed from the friction coefficient measurements. The endwall streamlines are representative of the flow streamlines in the limit as the wall is approached, and thus are also known as limiting streamlines. Figure 5 shows the endwall separation line also computed from the endwall friction coefficient vector measurements. The separation line was obtained by computing multiple streamline paths starting very near to the saddle point, and taking the mean of the paths. Note that since the saddle point is a location of zero shear stress, its exact location could not be found since the OFI method requires some detectable amount of oil flow.

**Adiabatic Effectiveness Levels from an Upstream Slot**

Contours of adiabatic cooling effectiveness for upstream slot flow at various flow rates and metering widths are presented in Figure 6. Note that for all cases in this study, there is a hot region around the vane-endwall junction where slot coolant is not present. This pattern is caused by the roll-up of the horseshoe vortex at the vane leading edge, which brings hot mainstream gases down to the endwall, and sweeps coolant off of the endwall. Further downstream, the combination of the horseshoe and passage vortices entrains coolant and sweeps it to the suction side of the vane.

To visualize endwall results in a more quantitative sense, values were extracted from the endwall data along the paths of inviscid streamlines obtained from a CFD prediction of the vane flowfield with FLUENT 6.2 [21]. Streamlines were released at mid-span of the vane from three pitch locations, corresponding to \( Y/P = 0.25, 0.50 \) and 0.75 as shown in Figure 6a. These streamlines are denoted as 0.25P, 0.50P, and 0.75P, respectively. Figure 7 presents the adiabatic effectiveness levels on the endwall resulting from the upstream slot flow, along each of the three streamline paths. The abscissa of the plots is non-dimensionalized distance along the streamline, shifted so that \( s/C=0 \) corresponds to where the streamline would cross an imaginary pitchwise line connecting adjacent vane stagnation points. Note that the plots in Figure 7 have different abscissas, since the streamlines have different lengths through the vane passage. In the shifted streamline coordinate system, the upstream slot is located at \( s/C=-0.35 \). Data is not plotted directly downstream of the slot, but rather starting at \( s/C=-0.3 \), in order to avoid the high measurement uncertainty at the start of the thermal boundary layer in the heat transfer coefficient measurements.

In Figure 7, effectiveness values eventually become zero as each streamline path crosses into the hot region. Note that in the...
most cases, the path of the 0.75P streamline crosses the hot ring region around the suction side of the vane, passes through the narrow band of coolant being swept to the suction side by the passage vortex, and re-enters the pressure side hot region (refer to Figure 6a). These crossings explain why effectiveness along the 0.75P path decays, increases, and then progressively decays.

As described earlier, the slot in this study was positioned at 0.96Cₘₓ upstream of the vane stagnation. A slot located far upstream, away from the influence of the vane, can reduce or eliminate the potential for hot gas path ingestion into the slot, but from a practical point of view may be difficult to implement in an engine. The effect of moving the slot upstream is shown in Figure 8. A similar lack of coolant coverage around the base of the vane is seen between this study’s results and those of Knost and Thole [10], for 1% slot flow from a nominal slot. The horseshoe and passage vortices control the distribution of coolant in the passage, despite the differences in slot location. Note that the upstream slot for the case in Figure 8 had the same geometry as the slot tested by Knost and Thole [10], but Knost and Thole positioned their slot at 0.38Cₘₓ from the vane leading edge. In the contours of Figure 8, and along the inviscid streamline paths in Figure 7, Knost and Thole’s [10] slot blowing at 1.0% MFR exhibits higher local effectiveness than the nominal slot blowing at 1.0% MFR. coolant ejected farther upstream interacts more with the mainstream and thus is less effective when it reaches the vane passage. However, the slot in Knost and Thole’s [10] study also results in more poorly distributed effectiveness levels across the endwall than for a slot ejecting farther upstream (see Figure 8). High effectiveness gradients would be a durability concern since they could lead to large gradients in endwall metal temperature and increased thermal stresses.

The effect of increasing the coolant flow rate can be seen by comparing the contours in Figures 6a and 6c for the nominal slot at two coolant flow rates, as well as in Figures 6b and 6d for the half slot at two coolant flow rates. For a given upstream slot width, effectiveness levels from upstream slot coolant over the endwall increase with increasing slot mass flow rates. Also, coolant coverage is more uniform downstream of the slot and is better able to penetrate into the hot ring around the vane-endwall junction for a slot MFR of 1% as compared to a slot MFR of 0.5%. Figures 7a and 7b show that for a given upstream slot width, the location of zero effectiveness along the 0.25P and 0.50P inviscid streamline paths moves farther downstream (convects farther into the passage) as the slot mass flow rate is increased from 0.5% to 1.0%.

Matching the upstream slot leakage flow while decreasing the slot width increases the coolant momentum, with the effect of increasing overall coolant coverage in the passage. Comparison of Figure 6a with 6b for the same mass flux ratios with differing slot widths (differing momentum flux ratios), and Figure 6c with 6d also for the same mass flux ratios with differing slot widths, shows that the half slot at a given mass flow rate produces more uniform coolant coverage upstream and around the vane-endwall junction than the nominal slot at the same mass flow rate. Along the streamline paths in Figure 7, for a given mass flow rate, coolant from the half slot generally has higher levels of effectiveness than the nominal slot as the coolant enters the vane passage (downstream of s/C=0). Maintaining the mass flow rate from the slot, while decreasing the metering area of the slot, requires a larger plenum-to-freestream pressure differential which forces coolant out of the slot more evenly. This finding is in agreement with the conclusions of Cardwell et al. [12], even though the upstream slot in this study was located farther upstream of the vane leading edge than in Cardwell et al.

The effect of matching the slot momentum flux ratio for the nominal and half slot is perhaps a more realistic situation for a turbine engine. Generally, the coolant will be supplied at a constant pressure relative to the mainstream. The effect of reducing the upstream slot width will be to reduce the coolant mass flow rate, but the average momentum flux ratio will remain the same. The coolant coverage region in Figure 6b for the half slot width at a MFR of 0.5% appears similar to the coverage region from the nominal slot at the same momentum flux ratio (Figure 6c). However, coolant effectiveness levels from the half slot flow are lower relative to the nominal slot at the same momentum flux ratio, since the nominal slot is ejecting more coolant.

When the endwall separation line for no upstream blowing is overlaid on the adiabatic cooling effectiveness contours in Figure 6a and 6d, it is apparent that for low slot coolant momentum, the trajectory of the coolant is dictated by the secondary flow vortices. For the highest momentum flux ratio tested, however, some coolant penetrates downstream of the separation line, indicating that the coolant has a more significant interaction with endwall secondary flow in this case.

**Heat Transfer Coefficients for an Upstream Slot**

Endwall heat transfer coefficients were measured with and without an upstream slot flow. For the baseline with no slot flow, a qualitative comparison was made to Kang et al. [5], who used the same airfoil geometry. Similar results were achieved, as can be seen in Figures 9a and 9b for both baselines with no slot flow, despite differences in the upstream tunnel geometry and inlet boundary layer thicknesses (δ/S = 0.09 for Kang et al. [5] and δ/S = 0.18 for our work). High heat transfer
coefficients are seen in Figures 9a and 9b near the vane stagnation region resulting from the horseshoe vortex. The effects of the passage vortex are seen further downstream as the contours sweep to the suction surface.

Figures 9b through 9f compare heat transfer coefficients with and without upstream slot flow for the various slot widths and flow rates. The overall effect of injection along the endwall is that the heat transfer coefficients are slightly increased in the cases with injection, relative to the baseline without injection. Figure 10 shows endwall heat transfer augmentation levels relative to the no leakage flow case along the 0.25P, 0.50P, and 0.75P streamline paths. Note that overall, augmentation levels with upstream slot blowing indicate higher heat transfer throughout most of the passage.
The effect of increasing the slot mass flow rate, while maintaining the slot width, is illustrated for the nominal slot in Figures 9c and 9e and for the half slot in Figures 9d and 9f. Stanton numbers increase slightly throughout the passage when the slot mass flow rate is increased from 0.5% to 1.0% for a given slot width. Heat transfer augmentation levels along the inviscid streamline paths (Figure 10) are generally higher as the mass flow rate increases through a given slot width. An exception to this trend, however, is the region around s/C= -0.05 on the 0.25P streamline path (Figure 10a) and the region around s/C= 0.10 on the 0.50P streamline path (Figure 10b). At these locations, there is no difference in heat transfer augmentation for the nominal slot injecting coolant at either mass flow rate. It was noted that the 0.25P streamline path crosses the separation line for no upstream slot blowing at s/C=-0.07, while the 0.50P streamline path crosses the separation line at s/C=0.11 (refer to Figure 9c). These results indicate that the separated flow dictates the heat transfer augmentation near the endwall separation line, rather than the injection itself. However, for the half slot, increasing its mass flow from 0.5% to 1.0% increases the coolant momentum to the point that the injection begins to interfere with the horseshoe vortex.

As mentioned earlier, for a turbine engine, a decrease in upstream slot width would result in a decrease in slot mass flow rate, but nominally the same momentum flux ratio. The effect of decreasing the slot width while maintaining the same momentum flux is seen by comparing the contours of Figures 9d and 9e. Stanton numbers are slightly lower for the half slot at a MFR of 0.5%, compared to the nominal slot at a MFR of 1.0%, even though both have the same momentum flux.

The endwall separation line, deduced from the measured friction coefficients with no upstream slot blowing, is overlaid on the heat transfer coefficient results for no upstream slot blowing (Figure 9b), and for blowing at the lowest and highest momentum flux ratios in Figures 9c and 9f, respectively. There is a region of low heat transfer between the vanes near the passage entrance that persists farther downstream for the case with the highest momentum flux ratio (Figure 9f) relative to the lowest momentum flux ratio (Figure 9c). These behaviors suggest that the upstream slot injection at a high momentum flux ratio reduces horseshoe vortex strength.

**Net Heat Flux Reduction for an Upstream Slot**

The determination of convective heat transfer to a turbine endwall with film-cooling requires knowledge of the film heat transfer coefficient, the metal wall temperature, and the adiabatic wall (recovery) temperature. The addition of coolant to the endwall reduces the adiabatic wall temperature and thus the driving potential for heat transfer. The mixing and flow disturbance induced by the coolant injection, however, generally increases the convective heat transfer coefficients. The net heat flux reduction parameter (NHFR) incorporates the effects of convective heat transfer coefficients and coolant effectiveness on the overall augmentation to the heat load to the component. To deduce the net heat flux reduction parameter, the component wall metal temperature must also be known, since it is the other driving temperature for heat transfer into the part. The non-dimensional form (denoted as $\varphi$) relates the wall temperature to the freestream and coolant temperatures. Since the wall temperature is dependent on the vane metal conductivity and internal cooling scheme, it had to be assumed for this study, which used adiabatic materials for the endwall. A value of $\varphi = 1.6$ was assumed based on typical film-cooled engine conditions (Sen et al. [22]). Note that values of NHFR $< 0$ indicate that the film cooling scheme is causing an increase in the overall heat flux experienced by the surface.

Figure 11 presents the NHFR calculated along each of the inviscid streamline paths. For the cases that have been studied, it is clear that there is a strong benefit of the leakage coolant at the entrance to the vane passage. Downstream, however, the cooling benefit of the leakage flow produces a detrimental effect where no coolant is present but the heat transfer coefficients have been increased. This increase is caused by alterations to the secondary flows that in turn increase the overall convective heat transfer coefficients. The 0.25P and 0.50P streamline paths, shown in Figures 11a and 11b,
respectively, indicate that the addition of upstream slot flow at any mass flow rate or slot width causes NHFR values that are less than zero near the exit of the vane passage. The lack of coolant penetration into the hot ring around the vane, combined with slight heat transfer coefficient augmentations from slot flow, leads to an increased net heat flux that would be experienced by the part.

It is apparent that the trends of NHFR along the streamline paths in Figure 11 parallel the trends of the effectiveness levels, and many of the same conclusions about the effects of slot flow can be made. Although increasing the slot flow rate tends to increase heat transfer coefficients, the additional coolant from a given slot width at 1.0% MFR relative to 0.5% MFR results in a net heat flux reduction. Also, the better coolant coverage from a half-width slot at a given mass flow rate reduces net heat flux, as compared to the nominal slot at the same mass flow rate. Finally, the case of matched momentum flux ratios for the nominal and half slots shows that, despite reduced Stanton number augmentations from the half slot blowing versus the nominal slot blowing, the lower amount of coolant ejected from the half slot at a MFR of 0.5% is not as effective in reducing heat flux into the endwall as is the coolant from the nominal slot at a MFR of 1.0%.

An overall, area-averaged NHFR is shown in Figure 12 as a function of slot momentum flux ratio. Coolant ejection from the half slot does a better job in decreasing average heat flux into the endwall than coolant from the nominal slot, when both are ejecting coolant at a given mass flow ratio. However, blowing from the half slot at a momentum flux ratio of 0.13 (MFR=0.5%) increases the average net heat flux (decreases NHFR parameter) by 26%, compared to the nominal slot blowing at the same momentum flux ratio (I=0.13, MFR=1.0%). Thus, axial contraction of the upstream slot, in a situation where the slot momentum flux is maintained, would increase endwall heat transfer.

CONCLUSIONS
Measurements of adiabatic cooling effectiveness and heat transfer coefficients on the endwall, from leakage flow through a two-dimensional upstream slot representing a combustor-turbine interface gap, were presented. Two slot flow rates and two slot widths were tested, to determine the effect of the contraction of the slot on the endwall heat transfer.

The dominance of the endwall secondary flow on upstream slot coolant coverage was demonstrated by moving the slot further upstream. The coverage area of the slot coolant, when the slot was located 0.96C∞ upstream of the vane, was similar to the coverage area for a slot located 0.38C∞ upstream. It was noted, however, that effectiveness levels in the passage were lower for the slot placed farther upstream since the coolant had more distance to interact with the mainstream flow.

For a fixed slot width, increasing the slot mass flow resulted in slightly increased heat transfer coefficients, but also higher adiabatic cooling effectiveness levels. Coolant exited the slot more uniformly at the higher slot flow rates, since the higher flow rate required an increase in slot pressure ratio. Overall, the net heat flux to the endwall was reduced, since more coolant was present at the higher flow rates.

Decreasing the slot width while maintaining a constant slot mass flow resulted in larger coolant coverage areas and increased local effectiveness levels. Moreover, heat transfer coefficient augmentations from the half width slot were lower than augmentations from the nominal slot. Area-averaged values of the net heat flux reduction parameter indicated that the coolant from the half slot width reduced the net heat flux to the endwall compared to the nominal slot, when both were ejecting coolant at the same mass flow rate. The higher momentum of slot coolant from the half width slot enabled it to penetrate farther into the passage.

Overlaying the endwall separation line for no blowing on contours of effectiveness and heat transfer coefficients revealed that for low slot momentum, the endwall secondary flow dictated the coolant trajectory in the passage. At the highest momentum flux ratio tested, a significant amount of coolant penetrated downstream of the endwall separation line, and heat transfer coefficients were reduced compared to the low momentum slot flow. The high coolant momentum in this case is suspected to have reduced the horseshoe vortex strength.

These results indicate that leakage flow through the combustor-turbine interface can provide cooling to the endwall, with a more uniform distribution of coolant when the interface is placed farther upstream. More uniform coverage reduces the spatial temperature variation in the turbine endwall metal, which is beneficial for durability. The effects of thermal expansion of the interface, however, need to be carefully considered since higher heat loads can result when the combustor-turbine interface gap contracts. Understanding how a leakage interface expansion impacts the heat transfer and cooling effectiveness of the leakage flow may help a designer to protect the part during unwanted thermal expansion incidents with the least amount of coolant.

Figure 12. Area-averaged net heat flux reduction (NHFR) to the endwall, as a function of upstream slot momentum flux ratio.
ACKNOWLEDGMENTS
The authors would like to thank the National Science Foundation’s GOALI program for funding this research (Grant No. 0412971). We would also like to acknowledge our partners in the GOALI program: Joel Wagner and Peter Tay (Pratt & Whitney), and Dr. Lee Langston and Brian Holley (University of Connecticut).

REFERENCES
[21] FLUENT (version 6.2.1), Fluent Inc., Lebanon, NH.