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 present 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. [1] presented adiabatic effectiveness results on the endwalls of the planar and contoured passages over a range of leakage flow rates.

2 Review of Relevant Literature

Although several categories for classification exist, endwall contouring can typically be labeled as either two- or three-dimensional. Two-dimensional endwall contouring is most commonly referred to as axisymmetric contouring, which 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 a nonlinear surface. A three-dimensional contour, otherwise known as a nonaxisymmetric contour, makes use of localized hills and valleys to alter the near wall pressure gradient and reduce secondary flows.

As mentioned previously, several researchers have concluded that nonaxisymmetric contouring is beneficial in reducing aerodynamic losses (Knezevic et al. [2], Praisner et al. [3], Gustafson et al. [4,5], and Mahmood et al. [6]). Not as common is the investigation of endwall heat transfer on a nonaxisymmetric contoured endwall. Saha and Acharya [7] presented the first known results of heat transfer on a nonaxisymmetric contoured endwall. Through computations, they evaluated nine nonaxisymmetric endwall contours and found that the best design reduced the average heat transfer on the contoured endwall by 8% relative to the flat endwall of a planar passage. In addition, Saha and Acharya [7] showed that local reductions in heat transfer could be as large as 300% near the suction side leading edge.

Currently, only two researchers have performed experimental measurements of endwall heat transfer on a nonaxisymmetric contoured endwall. Mahmood and Acharya [8] showed that the endwall Nusselt numbers were smaller on the contoured surface than on the flat surface, especially upstream of the midpassage location. In a more recent study, Lynch et al. [9] measured heat transfer levels on a nonaxisymmetric contoured endwall that were approximately 20% lower than the flat endwall results in regions of high heat transfer. While nonaxisymmetric contouring has shown to be beneficial for aerodynamic losses and endwall heat transfer, it has generally only been applied to turbine blade endwalls.

Similar to the findings for nonaxisymmetric contouring, past researchers have shown that axisymmetric contouring can be suc-

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

Heat transfer is a critical factor in the durability of gas turbine components, particularly in the first vane. An axisymmetric contour is sometimes used to contract the cross-sectional area from the combustor to the first stage vane in the turbine. Such contouring can lead to significant changes in the endwall flows, thereby altering the heat transfer. This paper investigates the effect of axisymmetric contouring on the endwall heat transfer of a nozzle guide vane. Heat transfer measurements are performed on the endwalls of a planar and contoured passage whereby one endwall is modified with a linear slope in the case of the contoured passage. Included in this study is upstream leakage flow issuing from a slot normal to the inlet axis. Each of the endwalls within the contoured passage presents a unique flow field. For the contoured passage, the flat endwall is subject to an increased acceleration through the area contraction, while the contoured endwall includes both increased acceleration and a turning of streamlines due to the slope. Results indicate heat transfer is reduced on both endwalls of the contoured passage relative to the planar passage. In the case of all endwalls, increasing leakage mass flow rate leads to an increase in heat transfer near the suction side of the vane leading edge.

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cessful in reducing aerodynamic losses within vane cascades (Morris and Hoare [10], Kopper et al. [11], Boletis [12], Dossena et al. [13], Lin et al. [14,15], and Barigozzi et al. [16]). Only two studies exist in the available literature that consider the heat transfer on an axisymmetric contoured endwall. Piggush and Simon [17,18] performed an extensive study to quantify the effects of leakage flow and misalignment of an upstream interface gap 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 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. Lin and Shih [19] performed a computational study to investigate the endwall heat transfer in a nozzle guide vane passage with axisymmetric contouring of one endwall. Two configurations of the same contoured endwall were investigated. In one configuration, all upstream contouring was upstream of the vane passage. In the other, contouring started upstream of the vane passage and continued through the passage. Heat transfer calculations were performed on both the flat and contoured endwalls. Comparing the contoured and flat endwalls between the two configurations, 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. Lin and Shih [19] did not investigate the results from a planar passage as a baseline for comparison to the contoured configurations.

The study reported in this paper is unique in that in addition to providing detailed heat transfer measurements on both endwalls of the axisymmetric contour passage, baseline measurements on the flat endwall of a planar passage are also presented. The contoured passage introduces a unique situation where each endwall is subject to increased acceleration relative to the planar case. In addition, the contoured endwall is under the influence of streamline curvature. The planar passage results serve as a baseline for comparison for the endwalls of the contoured passage as the flat endwalls do not experience either of these conditions.

### 3 Experimental and Computational Methods

All experiments were performed in a closed loop, low speed wind tunnel, depicted in Fig. 1 and previously described by Thrift et al. [1]. 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 containing a bank of heat exchangers 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. [1], the measurement method dictated that the endwall of interest must 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 2 presents a schematic illustrating the three endwall configurations. Note that for the contoured passage the area contraction began 1.25C_{ax} upstream of the cascade inlet with an endwall slope of approximately 16 deg.

Table 1 provides the vane geometry and flow conditions for both the planar and contoured passages. For both the computational fluid dynamics (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 2 documents the location and dimensions of the upstream leakage slot for the three endwall configurations. Note that the drawings are not to scale.

<table>
<thead>
<tr>
<th>Table 1 Vane geometry and flow conditions</th>
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<tr>
<td>Scaling factor</td>
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<td>Scaled vane chord (C)</td>
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<td>Axial chord/chord (C_{ax}/C)</td>
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<td>Pitch/chord (P/C)</td>
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<td>Planar passage</td>
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<td>Inlet span/chord (S_{in}/C)</td>
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<td>Velocity ratio (U_{in}/U_{coul})</td>
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<td>Inlet Reynolds number (Re_{in})</td>
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<td>Exit Reynolds number (Re_{coul})</td>
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<td>Contoured passage</td>
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Fig. 1 Depiction of the low speed, closed loop wind tunnel

Fig. 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 examined in this study.

Computations were performed assuming incompressible, viscous, low speed conditions using the FLUENT [20] commercial software package. FLUENT [20] is a pressure-based incompressible flow solver for unstructured meshes. FLUENT [20] 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 renormalization group method (RNG), k-ε model with nonequilibrium wall functions, was used for the turbulence model. Using the same modeling technique, Hermanson and Thole [21] were able to show good agreement between computational predictions and experimental measurements of secondary flows. Knost and Thole [22] 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 midspan 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 0.1 m, 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 × 10^{6} tetrahedral cells. Conversely, the contoured passage consisted of approximately 4.1 × 10^{7} tetrahedral cells since the entire passage span had to be modeled. Once the domain was created and meshed, the entity was imported into FLUENT [20] 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 ≤ y+ ≤ 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 were no longer changing. To evaluate whether the results were grid independent, 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 converged and grid independent.

3.1 Secondary Flow Field 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 Fig. 3) into the mean flow direction within the stagnation plane (V_x and V_z). The transformation quantifies the deviation of the local velocities from the inviscid flow field. This method of analysis has been previously used by both Kang and Thole [23] and Hermanson and Thole [21] to produce secondary flow plots upon planes of interest within a planar passage.

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_x and V_z).

3.2 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 Fig. 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 were recorded, and a set of five infrared (IR) thermography images was captured at each measurement location. An inframetrics P20 IR camera was used to acquire the spatially resolved temperatures on the endwall at a known heat flux. The heat transfer coefficients were then put into nondimensional form in terms of Nusselt number based on the axial chord.

The overall 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. [1]. The overall uncertainty in Nusselt numbers was ±7.5 (3%) at a Nu value of 250 and ±3.44 (5.6%) at a Nu value of 620.

4 Discussion of Results

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.

4.1 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 Fig. 4 for each endwall.

Figure 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 Fig. 5, which compares the secondary flow vectors and total pressure contours between each endwall at vane stagnation.

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 Fig. 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 cascade, thereby minimizing the horseshoe vortex, as shown in Figs. 5(b) and 5(c). In addition to weakening the horseshoe vortex, Fig. 5(c) 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, values were extracted from the endwall data along the paths of inviscid streamlines obtained from a prediction of the passage flow field using FLUENT [20]. The inviscid streamlines originate at the midspan of the vane at two pitch locations corresponding to y/P = 0.5 and 0.75, as shown in Fig. 4(a). In addition, heat transfer values were obtained across the pitch of the vane passage at an axial location 0.35C_{ax} downstream of the vane leading edge, as shown in Fig. 4(b).

Figure 6 compares the 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 6 shows that the heat transfer levels are high directly downstream of the slot compared with 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 slot 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, Fig. 7 gives the Nusselt number augmentation values along the same inviscid streamlines illustrated in Fig. 4. Figure 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 Fig. 8, which presents the Nusselt number values along the pitch of the passage at 0.35C_{ax} downstream of the vane leading edge for all three endwalls with no leakage flow and 1.0% leakage flow. Figure 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 the 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 9 presents the Nusselt number augmentation values along the same pitchwise line used to produce Fig. 8 with no leakage flow and 1.0% leakage flow. Figure 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 the strength of the horseshoe vortex and the subsequent passage secondary flows as a result of freestream acceleration lead to a reduction in endwall heat transfer for the contoured passage.

4.2 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 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. [1] 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.

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 the 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. and was 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 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 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. [1], 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. [1] 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 Fig. 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 Fig. 13, with a subsequent increase in the leakage flow to 1.0%.

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 indicate an intensification of the passage sec-

Fig. 7 Comparison of Nusselt number augmentations with no leakage flow and 1.0% leakage flow, sampled along inviscid streamlines released from (a) 50% pitch and (b) 75% pitch

Fig. 8 Comparison of Nusselt numbers between the three endwalls along the pitch of the vane passage at 0.35Cax with no leakage flow and 1.0% leakage flow

Fig. 9 Comparison of Nusselt number augmentations along the pitch of the vane passage at 0.35Cax with no leakage flow and 1.0% leakage flow
ondary flows relative to the low leakage flow cases. The ejection of coolant at the vane stagnation promotes the formation of the horseshoe vortex. Figure 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 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 Fig. 5. The consequence of intensifying the horseshoe vortex is the dramatic increase in the Nusselt numbers at the vane stagnation 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 [26] and Nicklas [27], 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 [26] 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 [27] 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 [28] investigated the effect of moving the slot further upstream to

![Fig. 10 Comparison of Nusselt number contours between the three endwalls with 0.25% leakage flow](image1)

![Fig. 11 Comparison of Nusselt number contours between the three endwalls with 0.5% leakage flow](image2)

![Fig. 12 Comparison of Nusselt number contours between the three endwalls with 0.75% leakage flow](image3)

![Fig. 13 Comparison of Nusselt number contours between the three endwalls with 1.0% leakage flow](image4)

![Fig. 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](image5)
0.3Cax 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 Figs. 6 and 8. Figure 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, average increases in heat transfer of 24%, 31%, and 39% are 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 the 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 Fig. 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, Fig. 15 presents the Nusselt numbers along the pitch of the passage at 0.35Cax downstream of the vane leading edge for all three endwalls over the entire range of leakage flow rates. Figure 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 Fig. 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.

4.3 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 Figs. 10–13. Figures 10–13 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 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 with the planar passage. With 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 reductions in heat transfer levels along the inviscid streamlines are 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 Fig. 9. Although the contoured passage heat transfer results are still generally lower than those in the planar passage, the average reductions in heat transfer along the pitchwise line are 8% and 9% for the flat and contoured endwalls, respectively.

Figure 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 large 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 Figs. 15(a) and 15(b), 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 Fig. 15(c). Recall that the reduced injection angle for the slot in the contoured passage 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 with the no leakage flow case.

Figure 16 presents the heat transfer augmentation values with and without leakage flow along the same pitchwise line used in Fig. 15 for the flat and contoured endwalls of the contoured passage, respectively. Figure 16(a) shows that the augmentation values are generally below unity on the flat endwall of the contoured passage except near the midpitch location. Contoured passages show a reduction in heat transfer levels for the flat endwall near midpitch. Approaching the suction side of the vane passage, Fig. 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.

5 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. A velocity transform 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. [1] 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.

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Nomenclature

\[ C = \text{true vane chord} \]
\[ C_L = \text{lift coefficient}, \frac{F_L}{0.5 \rho U_{\infty}^2} \]
\[ \text{Nu} = \text{Nusselt number}, \frac{q \text{conv}}{\Delta T_{\text{wall}}} \]
\[ P = \text{vane pitch or pressure} \]
\[ P_o = \text{total gauge pressure} \]
\[ q_{air} = \text{heat flux} \]
\[ Re = \text{Reynolds number}, U \frac{H_{11005}}{\nu_{air}} \]
\[ S = \text{vane span or distance along the vane circumference} \]
\[ T = \text{static temperature} \]
\[ U, V, W = \text{velocity components} \]
\[ V_{s} = \text{streamwise velocity in the x-y plane}, \]
\[ U \cos \Psi_{inv} + V \sin \Psi_{inv} \]
\[ V_{s} = \text{streamwise velocity,} \]
\[ V_{s} \cos \Phi_{inv} - W \sin \Phi_{inv} \]
\[ V_{s} = \text{normal velocity,} \]
\[ V_{s} \sin \Phi_{inv} + W \cos \Phi_{inv} \]
\[ X, Y, Z = \text{coordinate system} \]

Greek

\[ \rho = \text{air density} \]
\[ \nu = \text{kinematic viscosity} \]
\[ \eta = \text{area averaged adiabatic effectiveness} \]
\[ \Psi_{inv} = \text{inviscid turning angle in the xy plane}, \tan^{-1} \left( \frac{V_{s, inv}}{U_{inv}} \right) \]
\[ \Phi_{inv} = \text{inviscid turning angle in the xz plane}, \tan^{-1} \left( \frac{W_{inv}}{V_{s, inv}} \right) \]

Subscripts

\[ \text{ax} = \text{axial chord} \]
\[ \text{conv} = \text{heat transfer through convection} \]
\[ \text{exit} = \text{exit of the vane passage at the throat} \]
\[ \text{in} = \text{measured at the inlet of the cascade} \]
\[ \text{inv} = \text{inviscid conditions} \]
\[ \text{wall} = \text{wall measurement} \]
\[ \infty = \text{freestream} \]

References


