Effects of Orientation and Position of the Combustor-Turbine Interface on the Cooling of a Vane Endwall

First stage, nozzle guide vanes and accompanying endwalls are extensively cooled by the use of film cooling through discrete holes and leakage flow from the combustor-turbine interface gap. While there are cooling benefits from the interface gap, it is generally not considered as part of the cooling scheme. This paper reports on the effects of the position and orientation of a two-dimensional slot on the cooling performance of a nozzle guide vane endwall. In addition to surface thermal measurements, time-resolved, digital particle image velocimetry (TRDPIV) measurements were performed at the vane stagnation plane. Two slot orientations, 90 deg and 45 deg, and three streamwise positions were studied. Effectiveness results indicate a significant increase in area averaged effectiveness for the 45 deg slot relative to the 90 deg orientation. Flowfield measurements show dramatic differences in the horseshoe vortex formation. [DOI: 10.1115/1.4004817]

1 Introduction

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

This paper reports the effects of orientation and position of an upstream leakage slot on the cooling of a low-pressure, nozzle guide vane endwall. In addition, time-resolved flowfield measurements within the vane stagnation plane are presented. By simultaneously varying the slot width, the effects of varying the mass flow rate through the upstream leakage slot while maintaining the momentum flux ratio is also considered.

2 Review of Relevant Literature

Several past studies have investigated the performance of leakage flows from upstream interface gaps on endwall effectiveness levels. Many of these studies, however, have included film cooling from discrete holes making it difficult to separate the combined effects of film cooling and leakage flow. The focus of this particular literature search is on the interface gap between the combustor and turbine.

There is a wide breadth of available data on endwall cooling effectiveness for differing leakage gap geometries. Only a few studies, however, use consistent flow conditions and cascade geometries from which effects of varying the slot geometry and location can be properly analyzed. Cardwell et al. [1] investigated the effects of mass flux ratio and momentum flux ratio by varying the width of a 45 deg leakage slot located 0.3C\textsubscript{ax} upstream of the vane passage inlet. The slot width was varied from a nominal width to a half and double width configuration with mass flows ranging from approximately 0.4\% to 1\%. Note that the vane passage used by Cardwell et al. [1] also included leakage flow from a mid-passage gap and film cooling holes. As concluded by previous researchers, Cardwell et al. [1] found that effectiveness levels were dependent on the leakage mass flow. In addition, Cardwell et al. [1] found that the coolant coverage pattern was a function of momentum flux ratio.

Axial location has been shown to be an important parameter in the cooling performance of an upstream interface gap. Kost and Nicklas [2] and Nicklas [3] made aerodynamic and thermodynamic measurements within a linear turbine cascade with a transonic flowfield utilizing leakage flow from a 45 deg slot located 0.2C\textsubscript{ax} upstream of the vane leading edge. Film cooling holes were also distributed within the vane passage. Kost and Nicklas [2] showed that the horseshoe vortex (HSV) was strengthened by the ejection of coolant from the leakage gap at a mass flux ratio of 1.3\%. This was attributed to the location of the slot, being positioned in the region of the saddle point. In a similar experiment, Kost and Mullaert [4] investigated the effect of moving the slot further upstream to 0.3C\textsubscript{ax} upstream of the vane cascade. Results showed that the intensification of the horseshoe vortex could be avoided by moving the gap further upstream. In addition, it was found that the leakage flow stayed closer to the endwall and provided better cooling even with less than half the amount of coolant as in the case where the slot was at 0.2C\textsubscript{ax}.

Lynch and Thole [5] and Kost and Thole [6] performed adiabatic effectiveness experiments using leakage flow from identical 45 deg slots with the same vane cascades and inlet conditions. Experiments by Lynch and Thole [5] and Kost and Thole [6] differed in the upstream location of the interface gap corresponding to 0.96C\textsubscript{ax} and 0.38C\textsubscript{ax} upstream of the vane cascade respectively. It was noted that the coverage area of the leakage coolant was similar between the two locations. The effectiveness levels within the passage, however, were lower for the slot placed further.
upstream as the coolant had more distance to mix with the hot mainstream flow. Lynch and Thole [5] also investigated reducing the slot width while maintaining the leakage mass flux ratio, essentially increasing the momentum flux ratio. Similar to that found by Cardwell et al. [1], Lynch and Thole [5] showed that increased momentum flux ratios resulted in larger coolant coverage areas and increased local effectiveness levels.

Though not as common as effectiveness measurements, several studies present flowfield data within the vane stagnation plane illustrating the formation and dynamics of the leading edge horseshoe vortex. Relatively early studies by Kang et al. [7] and Radomsky and Thole [8] studied the effect of inlet Reynolds number and turbulence level respectively on the formation of the HSV. In the case of these two studies, however, no leakage flows were present. The flowfield data that they reported was only time-averaged given their use of a laser Doppler velocimeter (LDV). Some studies have also used particle image velocimetry to not only capture time-averaged velocity flowfield data but time-resolved results as well. Three recent studies by Prairier and Smith [9], Hada et al. [10], and Sabatino and Smith [11] investigated the dynamic behavior of the HSV and the associated endwall heat transfer with no leakage flows. All three studies perform measurements in the stagnation plane of a two-dimensional, symmetric airfoil utilizing a cylindrical nose. Prairier and Smith [9] showed a strong correlation between the unsteadiness of the HSV and the corresponding endwall heat transfer. Sabatino and Smith [11] found that the temporal behavior of the HSV system was driven by the unsteady characteristics of the impinging turbulent boundary layer. Hada et al. [10] showed that the vorticity of the HSV and the corresponding endwall heat transfer increased with a reduction in leading edge diameter. Although each study investigated a different factor influencing the dynamics of the HSV, each study showed the unsteady nature of the structure. The HSV is described as undergoing a bimodal switching where for the majority of the time the vortex structure is present. For less time but occurring at somewhat of a regular frequency, the vortex structure is weakened and translated upstream before starting the quasi-periodic cycle over again.

While the previously mentioned flowfield studies have presented detailed measurements of the HSV, none of the studies included leakage flow from an upstream interface gap. One of the few studies to provide detailed flowfield measurements in the stagnation plane of a nozzle guide vane while incorporating leakage flow was performed by Sundaram and Thole [12]. Sundaram and Thole [12] performed LDV measurements in the same vane cascade used in the studies by Kang et al. [7] and Radomsky and Thole [8]. Unlike these two studies, however, the endwall incorporated leakage flow from an upstream 45 deg slot as well as an upstream row of film cooling holes inclined at 30 deg. Measurements were performed with and without a trench on the leading edge row of film cooling holes. The flowfield measurements indicated a vortex forming both upstream of the ejected film coolant and downstream between the film cooling holes and the vane when no trench was used. In the presence of the trench, however, the upstream vortex was shown to disappear and only a vortex upstream of the ejected film coolant was observed.

To date, most studies have focused on the effects of mass flux ratio on the cooling performance of an upstream leakage gap. Only a few studies have investigated changing gap parameters such as width, orientation, and location to observe the resulting effect on endwall effectiveness levels. In addition, most flowfield measurements at the leading edge do not include leakage flow. Those that do are limited to time-averaged results that are incapable of discussing any instantaneous flowfield structures. The study reported in this paper seeks to understand the effects of orientation and position of an upstream leakage gap on the cooling characteristics of a vane endwall by providing detailed measurements of adiabatic effectiveness. In addition, the resulting adiabatic effectiveness results will be supported by time-resolved flowfield measurements in the stagnation plane of the nozzle guide vane.

3 Experimental Methods and Benchmarking

Experiments were performed in a low speed, closed loop wind tunnel, depicted in Fig. 1 and previously described by Thrift et al. [13]. The flow was driven through the wind tunnel by a 50 hp fan controlled by a variable frequency drive. After passing the fan the flow was turned 90 deg and then passed through a finned-tube heat exchanger. The primary heat exchanger circulated temperature controlled water at approximately 10 °C to remove the initial heat supplied to the flow by the fan.

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

After the split, the primary flow continued down the center path where it encountered an electrical resistance heater bank capable of supplying 55 kW of heat. Using a small percentage of the heater bank capacity, the main flow was heated to approximately 50 °C to increase the temperature difference between the main flow and coolant flow. After the heater bank, the main flow passed through a series of screens used for flow straightening and then into a contracted straight flow section. The contraction reduced the flow area from 1.11m² to 0.62 m² through rounded inlets alter which the flow area was constant. At the exit of this section was the experimental test section where all measurements were performed. The test section incorporated a 90 deg bend to assist in the turning of the flow through the vane cascade. Air exiting the test section was turned by a final 90 deg elbow before encountering the fan and completing the closed-loop.

The vane test section was a two-dimensional, linear vane cascade as illustrated in an over-head view schematic in Fig. 2. The test section contained two full nozzle guide vanes and a third partial vane connected to a flexible wall to maintain the desired pressure distribution along the three vanes. Constructed of low-density closed cell polyurethane foam, the vane design was a span-wise partition of a two-dimensional midspan vane geometry. The vanes were scaled up by a factor of 2.4 from engine size to achieve high measurement resolution. A description of the nozzle guide vane parameters is given in Table 1.

To simulate leakage flow from the combustor turbine interface gap a two-dimensional slot was placed on the bottom endwall, upstream of the vane stagnation as illustrated in Fig. 2.
Note that the upstream slot is at the nominal location in Fig. 2, width, slot orientation, and slot location could be easily changed. The upstream slot was interchangeable so that the flow metering was determined. Unlike the engine, the wind tunnel operates with a constant momentum flux ratio. In the current experiments, three upstream slot widths were chosen to model the interface gap so that the flow becomes fully developed before exiting the slot. Finally, the effects of slot location were considered by moving the slot further upstream and downstream from the nominal location, to \( x = -0.34C_{ax} \) and \(-0.05C_{ax}\) respectively.

The boundary layer entering the cascade was measured at a location 4.25\(C_{ax}\) upstream of the vane stagnation by Thrift et al. [13] using the same inlet conditions. Table 3 lists the turbulent inlet boundary layer parameters, which were maintained throughout this study. The measured turbulence intensity was lower than that typically found in an engine at 1.0%. The effect of turbulence, however, was not considered in this study as past studies by the authors have evaluated this effect [8].

### Table 3 Inlet turbulent boundary layer characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boundary layer thickness/midspan ((\delta'/S))</td>
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</tr>
<tr>
<td>Displacement thickness/midspan ((\delta'/S))</td>
<td>0.038</td>
</tr>
<tr>
<td>Momentum thickness/midspan ((\theta'/S))</td>
<td>0.03</td>
</tr>
<tr>
<td>Shape factor ((\delta'/\theta))</td>
<td>1.3</td>
</tr>
<tr>
<td>Momentum thickness Reynolds number (Re(\theta))</td>
<td>4245</td>
</tr>
</tbody>
</table>

Experiments were also performed with a slot orientation of 90 deg and 45 deg as indicated in Fig. 2. Each slot had a flow length-to-axial chord ratio of 0.17. Subsequently, this gives a changing flow length-to-width ratio of 11.7, 16.8, and 38.6 for the 3.3 mm, 2.3 mm, and 1 mm slot widths respectively. The relatively large flow length-to-width ratios for each slot width, however, suggest that the flow becomes fully developed before exiting the slot. The flow length-to-width ratio of 11.7, 16.8, and 38.6 for the 3.3 mm, 2.3 mm, and 1 mm slot widths respectively. The relatively large flow length-to-width ratio of 11.7, 16.8, and 38.6 for the 3.3 mm, 2.3 mm, and 1 mm slot widths respectively.

#### 3.1 Mainstream and Coolant Flow Settings.

Before performing any experiment, the inlet velocity distribution across the cascade pitch and the vane pressure distributions along the mid-span were verified. The inlet velocity distribution was measured approximately 0.8\(C\) upstream of the cascade inlet at 50% span height. Measurements were taken at eight locations spanning from \(\pm 0.75\)\(P\) about the center vane stagnation. The inlet velocity range across the width of the cascade varied less than 5% from the pitch averaged mean for all experiments. Static pressure measurements around the circumference of the vane at midspan were compared with predictions from a computational study. The computational study was previously performed by Thrift et al. [14] for incompressible, viscous, low-speed conditions using FLUENT [15]. A detailed description of the computational method and domain used for generating the predicted pressure distribution is provided by Thrift et al. [14]. As shown previously by Thrift et al. [13], the measured and predicted pressure distributions agreed indicating that the inviscid flowfield around each vane was matched to the predicted curve.

The mass flux issuing from the upstream leakage slot was measured using a laminar flow element located within the supply pipe to the slot plenum. The leakage coolant issuing from the upstream slot and entering each passage was then defined as a prescribed percentage of a single passages mass flow rate. As shown previously in Table 2, coolant mass flux ratios ranged from 0.3% to 1.0%. With the known mass flux and leakage slot width, an average momentum flux ratio could be calculated according to Eq. (1) below.

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

The vane span height was maintained throughout this study. The slot width, however, had to be reduced simultaneously with the mass flux ratio to maintain a constant momentum flux ratio as indicated in Eq. (1) and shown in Table 2. As mentioned previously the slots were individual, interchangeable components which were machined to the desired widths. The slot widths were verified with the use of calipers and gauge blocks to within approximately \(\pm 0.1\) mm.
3.2 Adiabatic Effectiveness Measurements. Spatially resolved, adiabatic wall temperatures were obtained from infrared (IR) measurements of the endwall. Adiabatic effectiveness experiments were performed at steady state conditions with a temperature difference between the freestream and leakage coolant of approximately 25 °C to reduce measurement uncertainty. To ensure the measured wall temperatures were adiabatic, the endwall was made of a 2.54 cm thick plate of low density closed cell polyurethane foam, which has a very low thermal conductivity (0.03 W/mK). The foam endwall was painted black to maintain a high emissivity on the endwall surface thus providing good resolution of the surface temperatures. Type-E thermocouples were placed throughout each vane passage for calibration of the IR images.

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

Images were post-calibrated by determining the emissivity and background temperature of the image through matching of the IR captured surface temperatures with the acquired thermocouple measurements. Although the thermal conductivity of the foam endwall was low, it was necessary that a small conduction correction still be used. A one-dimensional conduction correction as described by Ethridge et al. [16] was applied to all adiabatic effectiveness measurements. The correction involved measuring the endwall surface effectiveness with no coolant flow. A correction value of \( \eta_o = 0.1 \) was measured within the vane passages. Upstream of the leakage slot, however, a correction value of \( \eta_o = 0.15 \) was measured. The endwall upstream of the leakage slot was made of medium density fiberboard with a thermal conductivity (0.13 W/mK) higher than that of the foam making up the cooled surface, resulting in slightly larger conduction losses.

Using the partial derivative method described by Moffat [17] an uncertainty analysis was performed on the measurements of adiabatic effectiveness based on the uncertainties associated with the measured temperatures. Based on a 95% confidence interval the precision uncertainty of the IR temperature measurements was ±0.2 °C. The bias uncertainty for an image was taken as the root-sum-square of the thermocouple bias uncertainty (±0.2 °C) and the average deviation of the calibrated images from the thermocouples (±0.5 °C) resulting in a bias uncertainty of ±0.54 °C. Combining the bias and precision uncertainties, a total uncertainty in adiabatic effectiveness was then found to be \( \delta \eta = ±0.03 \) over the range \( \eta = 0.03 \) to 0.7.

3.3 Particle Image Velocimetry Measurements. The stagnation plane flowfield upstream of the vane was recorded using a high-image-density, time resolved, digital particle image velocimetry (TRDPIV) system. Figure 2 illustrates the experimental setup used when performing TRDPIV. TRDPIV is a noninvasive, laser-optical measurements technique based on the illumination and tracking of particles which follow the flowfield [18–22]. The TRDPIV system consisted of a high frequency double pulsed laser, high speed CMOS camera, and controller that synchronized the timing between the laser and camera. The dual cavity 15W Nd:YAG laser was capable of firing at 10 kHz per laser cavity. The time delay between laser pulses and the resulting image pairs was adjusted for each run to obtain a bulk particle displacement of approximately seven pixels. Di-Ethyl-Hexyl-Sebacat (DEHS) seeder particles with an approximate mean diameter of 1 μm were introduced directly upstream of the wind tunnel blower using a Laskin nozzle.

The raw images were corrected for the small off normal viewing angle and then vector fields were produced with cross correlation using a standard cyclic FFT-based algorithm [23]. A decreasing, multipass processing technique was employed with a single pass at an interrogation window size of 64 × 64 pixels and then a double pass at an interrogation window size of 32 × 32 pixels with 50% overlap resulting in a final vector spacing of 16 × 16 pixels. In some cases it was necessary to use a final interrogation window size of 16 × 16 pixels with 50% overlap resulting in a vector spacing of 8 × 8 pixels to capture smaller flow features. Multipass and final pass post processing was performed using a 4-pass regional median filter to remove spurious vectors.

To qualify the capabilities of the TRDPIV system, flowfield measurements were performed in the stagnation plane with no leakage slot present to observe the dynamics of the unsteady HSV. The particular dynamics of the HSV is a well studied phenomena by researchers as early as Devenport and Simpson [24] using LDV and as recent as Praisner and Smith [9] and Sabatino and Smith [11] using PIV. Although these studies investigated the HSV upstream of a streamlined cylinder, their studies still serve as valid baselines for comparison. Figure 3 presents instantaneous flowfield of the HSV transitioning from the dominate flow mode to a secondary flow mode. The first contour plot shows the clock-wise rotating HSV at an axial position of approximately \( x/C_{ax} = -0.07 \). Directly upstream of the HSV is a secondary vortex with the opposite sense of rotation followed by a small tertiary vortex with the same rotation direction as the HSV. This mode dominates the flowfield for the majority of the time only to be interrupted by the quasi-periodic switching to a second flow mode. As shown in Fig. 3(b) the dominate mode is interrupted when the secondary vortex is moved away from the endwall and an inrush of fluid is observed behind the HSV, displacing it upstream. A short time later, Fig. 3(c) shows that the size of the HSV is reduced while being displaced further upstream and closer to the endwall. This quasi-periodic switching is identical to that observed by Praisner and Smith [9] and Sabatino and Smith [11].

Although Devenport and Simpson [24] were unable to capture the entire time-resolved flowfield using LDV, they were able to comment on the bi-modal switching of the near wall velocities. For comparison, Fig. 4 presents probability density functions of the offset streamwise velocity at two different spanwise locations taken at the average axial position of the HSV core, \( z/S = 0.054 \). The first location, \( z/S = 0.0034 \) is below the HSV core and shows a bimodal velocity distribution indicative of the switching between flow modes. In Fig. 4(b), above the vortex core at \( z/S = 0.054 \), only a single velocity peak is observed. The velocity histograms presented in Fig. 4 are very similar to those presented by Devenport and Simpson [24]. Devenport and Simpson [24] commented that the double-peaked histogram implies that the near wall velocity has two preferred states which were deemed the backflow and zeroflow modes. In addition, Devenport and Simpson [24] showed that the turbulent fluctuations at a high near the vortex region as a result of the structures unsteady motion. Radomsky and Thole [8] also identified increased streamwise fluctuations in the region of the HSV. The two flow modes represent the quasi-periodic switching observed by Praisner and Smith [9] and Sabatino and Smith [11] and shown in Fig. 3. The observed dynamics of the HSV compare well with the results of past researchers.
4 Discussion of Results

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

4.1 Effects of Slot Orientation. To study the effects of slot orientation, two separate slot angles of 90 deg and 45 deg were considered with both being located at $x/C_{ax} = 0.17$, upstream of the vane stagnation. Experiments were performed at a constant momentum flux ratio for several mass flux ratios. As shown previously in Table 2, three slot widths were considered for this study. Slot widths of 3.3 mm, 2.3 mm, and 1 mm were studied for MFR’s of 1.0%, 0.7%, and 0.3%, respectively. By reducing the slot width and mass flux ratio simultaneously the momentum flux ratio could be maintained at approximately 2.8 as according to Eq. (1). Maintaining the momentum flux ratio instead of the mass flux ratio represents a realistic constraint on turbine designers. Within an engine, the pressure difference between the coolant and the exit pressure at the endwall remains constant. Maintaining the same pressure ratio results in a constant momentum flux ratio given that pressure difference scales with velocity squared.

Figure 5 compares adiabatic effectiveness levels between the 90 deg and 45 deg slot orientations at the nominal location for several MFR’s of 1.0%, 0.7%, and 0.3%. As shown, the adiabatic effectiveness values for the 90 deg slot orientation are generally higher than those for the 45 deg slot orientation. The difference in effectiveness levels is most pronounced at the highest mass flux ratio of 3.3 mm. At lower mass flux ratios, the difference in effectiveness levels is less significant. The results suggest that a 90 deg slot orientation is more effective than a 45 deg slot orientation for the given range of mass flux ratios.

Fig. 3 Instantaneous flowfield vectors and contours of vorticity in the stagnation plane with no leakage flow showing the transition between two flow modes

Fig. 4 Histogram of velocity probability density for no leakage flow at $x/C_{ax} = 0.08$ and (a) $z/S = 0.0034$ and (b) $z/S = 0.054$

Fig. 5 Comparison of adiabatic effectiveness contours at several different mass flux ratios with $l = 2.8$ between the (a) 90 deg and (b) 45 deg slot orientations at $x/C_{ax} = 0.17$. 

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midspan near the vane stagnation, an intermittent vortex structure forms on the vane surface. As a result of the strong turning of the flow toward the endwall, the endwall flow begins to turn up toward midspan on approach to the vane-endwall junction where the spanwise direction of the velocity is reversed, setting up a scenario where a counterclockwise rotating vortex can form. The stagnation plane flowfield in Fig. 7 depicts a small vortex at $z/S = 0.12$. The second flowfield shows that a short time later the single vortex near the vane surface is gone and appears to be replaced by a strong downwash of fluid from $z/S = 0.15$ to 0.05. Contrarily, an increase in the upwash of fluid near the vane-endwall junction is observable. Subsequently, the final flowfield shows that the interaction of the two opposing flows near the vane surface results in the formation of a counterclockwise rotating vortex at approximately the same position observed in the first time instance. In addition, a vortex is also formed near the vane-endwall junction where the turning of fluid toward the midspan is the greatest. Although Fig. 7 indicates small vortical structures forming at both $z/S = 0.03$ and 0.12, these structures do not appear to convect up the surface of the vane. Instead, the structures were created along the inflection points within $-0.05 < x/C_{ax} < 0$ and $0.03 < z/S < 0.15$ and were quickly dissipated or moved out of the stagnation plane.

To observe the effect of slot orientation on the stagnation flowfield, a lower MFR, Fig. 8 presents time-averaged flowfield results for both the 90 deg slot orientation at $x/C_{ax} = -0.17$ and the 45 deg slot orientations with MFR = 0.1% and $I = 2.8$. Different mass flux ratios. Inspection of Fig. 5 shows that at each mass flux ratio the adiabatic effectiveness levels are substantially higher for the 45 deg slot orientation. The improvement in cooling performance for the 45 deg slot is likely related to the ejection angle being closer aligned to the plane of the endwall. To understand how the ejected coolant alters the stagnation plane flowfield, Fig. 6 presents time-averaged flowfield results for both the 90 deg and 45 deg slot orientations at $x/C_{ax} = -0.17$.

The stagnation plane flowfields show vastly different flow patterns between the 90 deg and 45 deg slot orientations. In the case of the 90 deg slot, all of the coolant ejects into the spanwise direction. As indicated in Fig. 6(a), the ejection of coolant perpendicular to the vane-endwall results in the separation of the incoming boundary layer. The static pressure gradient, formed along the span of the vane as a result of the stagnating boundary layer, forces the separated leakage flow to turn toward the endwall to form the leading edge vortex. The time-resolved results indicate that the ejected coolant sustains the large leading edge vortex as the structure never dissipates. In addition to the large leading edge vortex, a small tertiary vortex with the same direction of rotation is also present directly upstream of the ejected coolant. Unlike the large vortex structure, the tertiary vortex exhibits some unsteady dynamics, dissipating and then reforming. For the majority of the time, however, the tertiary structure is present as it shows up in the time-averaged results presented in Fig. 6(a). Note that the time-averaged tertiary vortex in Fig. 6(a) is similar to that seen in the instantaneous flowfields with no leakage flow presented in Fig. 3.

For the 45 deg slot orientation, no permanent vortex structure is present. As shown in Fig. 6(b), the coolant is ejected along the endwall. Ejecting coolant along the endwall plane fills in the approaching boundary layer in the low velocity region near the wall. Thickening the boundary layer in the near wall region produces a static pressure gradient near the vane-endwall junction that is away from the endwall, opposite of that seen for the 90 deg slot orientation. Proof for the presence of this pressure gradient can be seen in Fig. 6(b) as a region of positive vorticity where the endwall flow begins to turn up toward midspan on approach to the vane surface. As a result of the strong turning of the flow toward midspan near the vane stagnation, an intermittent vortex structure is occasionally formed. The formation of this structure occurs where strong inflection points exist in the time-averaged streamlines presented in Fig. 6(b). These strong inflection points indicate where the spanwise direction of the velocity is reversed, setting up a scenario where a counterclockwise rotating vortex can form. Figure 7 presents three sequential, instantaneous flowfields to illustrate the formation of these vortical structures. The first flowfield in Fig. 7 presents a small vortex at $z/S = 0.12$. The second flowfield shows that a short time later the single vortex near the vane surface is gone and appears to be replaced by a strong downwash of fluid from $z/S = 0.15$ to 0.05. Contrarily, an increase in the upwash of fluid near the vane-endwall junction is observable. Subsequently, the final flowfield shows that the interaction of the two opposing flows near the vane surface results in the formation of a counterclockwise rotating vortex at approximately the same position observed in the first time instance. In addition, a vortex is also formed near the vane-endwall junction where the turning of fluid toward the midspan is the greatest. Although Fig. 7 indicates small vortical structures forming at both $z/S = 0.03$ and 0.12, these structures do not appear to convect up the surface of the vane. Instead, the structures were created along the inflection points within $-0.05 < x/C_{ax} < 0$ and $0.03 < z/S < 0.15$ and were quickly dissipated or moved out of the stagnation plane.

To observe the effect of slot orientation on the stagnation flowfield, a lower MFR, Fig. 8 presents time-averaged flowfield results for both the 90 deg and 45 deg slot orientations with MFR = 0.7% and $I = 2.8$. Comparison of the flowfield results in

![Image](http://turbomachinery.asmedigitalcollection.asme.org/)

**Fig. 6** Average flowfield vectors, streamlines, and contours of vorticity in the stagnation plane with MFR = 1.0% and $I = 2.8$ for the (a) 90 deg and (b) 45 deg slot orientations at $x/C_{ax} = -0.17$

![Image](http://turbomachinery.asmedigitalcollection.asme.org/)

**Fig. 7** Instantaneous flowfield vectors, streamlines, and contours of vorticity in the stagnation plane with MFR = 1.0% and $I = 2.8$ for the 45 deg slot orientation at $x/C_{ax} = -0.17$ for three time instances
Figs. 6 and 8 show that the overall flow structure is very similar between the two MFR cases. For the 90 deg orientation, the ejecting coolant lifts off the approaching endwall flow and is then rolled up into a permanent vortical structure. Likewise, flow ejecting from the 45 deg slot travels along the endwall resulting in a slight upwash of fluid near the vane-endwall junction but no permanent vortex.

Comparison of Figs. 6(a) and 8(a) show that the size of the vortex for the 90 deg orientation is much smaller for the MFR = 0.7% case. Subsequently, the maximum vorticity magnitude is larger. The axial position of the vortex center is approximately the same between the two MFR cases at $x/C_{ax}$ = -0.11, but the spanwise position decreases from $z/S = 0.035$ to 0.03 with a reduction in MFR from 1.0% to 0.7%. For the 45 deg slot, a reduction in MFR results in a weaker upwash of fluid near the vane-endwall junction as seen in a comparison of Figs. 6(b) and 8(b). Subsequently, the severity of streamline curvature near the vane stagnation is smaller resulting in the formation of weaker counter-rotating vortices.

To further explore the effects of reduced MFR, Fig. 9 presents the time-averaged flowfield results for the 90 deg slot with MFR = 0.3% and 1 = 2.8. Comparison of Fig. 6(a), 8(a), and 9 show that reducing MFR results in a consistent reduction in the size of the leading edge vortex. Although the axial position remains at approximately $x/C_{ax}$ = -0.11 for all three MFR cases, the spanwise position of the vortex center is moved closer to the endwall with decreasing MFR. At MFR = 0.3% the vortex is centered at approximately $z/S = 0.025$, as compared to $z/S = 0.035$ and 0.03 for MFR = 1.0% and 0.7%, respectively.

The improvement in endwall cooling performance associated with the 45 deg slot orientation is an obvious result as there is less mixing between the coolant and hot mainstream flow as indicated in the flowfield results. What is not apparent is the qualitative improvement in endwall cooling effectiveness of the 45 deg slot with respect to the 90 deg orientation. To quantify the improvements associated with the 45 deg slot, area averages of the endwall effectiveness values are presented in Fig. 10 for the contours presented in Fig. 5. The area over which the averages were performed extended from the downstream edge of the leakage slot to the passage throat and across the passage pitch from one vane stagnation to the other. The averaging area is depicted in Fig. 5(a) on the 1.0% MFR contour. Ejecting coolant closer to the plane of the endwall results in a large increase in area averaged effectiveness. Figure 10 indicates a percent difference increase in area averaged effectiveness of approximately 177% and 129% for MFR’s of 0.7% and 1.0%, respectively between the 90 deg and 45 deg slot orientations.

In addition to the cooling effectiveness improvements associated with the 45 deg slot, Fig. 5 shows that there is little change in the overall range of effectiveness levels with a reduction in mass flow for both slot orientations. The coolant, however, is shown to penetrate progressively less into the vane passage with a reduction in mass flow. For the 90 deg slot, local effectiveness levels downstream of the slot near the suction side leading edge are shown to increase slightly with reduced mass flow. As indicated in Figs. 6(a), 8(a), and 9, the coolant penetrates less into the freestream with each subsequent reduction in MFR, improving the near slot cooling effectiveness. Local effectiveness levels near the downstream edge of the slot for the 45 deg orientation, however, show little change.

Figure 10 indicates that the area averaged effectiveness increases with mass flux ratio for both slot orientations. Qualitatively, the 45 deg slot realizes a percent difference increase in area averaged effectiveness of 7% with an increase in MFR from 0.7% to 1.0%. Although providing poorer cooling performance, the 90 deg slot orientation sees a greater increase in area averaged effectiveness with increasing MFR. The 90 deg slot realizes a
percent difference increase in area averaged effectiveness of 41% and 29% with an increase in MFR from 0.3% to 0.7% and 0.7% to 1.0%, respectively.

4.2 Effects of Slot Position. The effects of the slot position on the cooling performance and stagnation plane flowfield were studied by moving the 90 deg slot further upstream and downstream from the nominal location to \(x/C_{ax} = -0.34\) and \(-0.05\), respectively. Figure 11 compares adiabatic effectiveness contours for the three upstream locations for the 90 deg slot with MFR = 1.0% and \(l = 2.8\).

Figure 11 shows that there is little difference in the overall range in effectiveness level between the slot at the nominal location and the upstream location. For the downstream location, however, an improvement in the local cooling performance of the leakage slot is observed. The relatively close proximity of the leakage slot to the vane stagnation makes it likely that some coolant is washed down the surface of the vane and onto the endwall improving the effectiveness near the suction side leading edge. To appreciate why these differences in cooling performance occur, Fig. 12 presents time-averaged flowfield results for the 90 deg slot at the further upstream and downstream locations with MFR = 1.0% and \(l = 2.8\).

At a location \(x/C_{ax} = -0.34\) upstream of the vane stagnation, Fig. 12(a) indicates that the high momentum leakage flow lifts off the incoming boundary layer and forms a permanent vortex structure. In addition, the strong back flow underneath the vortex results in a small counter clockwise recirculation zone between the ejecting coolant and main vortex at \(x/C_{ax} = -0.029\). Comparisons of the leading edge vortex at a slot location of \(x/C_{ax} = -0.17\) and \(-0.34\) indicate a similar yet stretched vortex structure as shown in Figs. 6(a) and 12(a). The further upstream slot location is subject to a weaker spanwise pressure gradient allowing leakage flow to travel further downstream before being turned toward the endwall and into the leading edge vortex. The extended penetration of leakage flow downstream results in a lengthened vortex structure and subsequently a weaker core vorticity magnitude. Figures 6(a) and 12(a) show that the vortex center is located approximately \(0.12C_{ax}\) and \(0.06C_{ax}\) from the downstream edge of the leakage slot at the further upstream and nominal location, respectively. The larger distance between the vortex structure and the ejecting coolant allows room for the small recirculation zone mentioned previously to form.

Most important to the endwall effectiveness level, however, is the penetration depth of the leakage coolant in to the mainstream. The penetration depth determines the amount of contact between the coolant and mainstream and subsequently how effective the leakage flow will be in cooling the endwall. Figures 6(a) and 12(a) show similar penetration depths of the leakage coolant to a spanwise location of approximately \(z/S = 0.09\). Also note that the spanwise location of the vortex center is similar between the nominal and further upstream slot locations at \(z/S = 0.035\). The similarity between the vortex structures at the further upstream and nominal location correspond to the similarity between endwall effectiveness values.

At the further downstream location, \(x/C_{ax} = -0.05\), the ejecting coolant has a high enough momentum to cause the separation of the approaching boundary layer as shown in Fig. 12(b). Unlike the other two upstream locations, however, a very strong spanwise pressure gradient and a large exit static pressure greatly reduce the penetration of the leakage flow into the mainstream. The ejecting coolant is quickly turned toward the endwall to form a small vortex with a relatively high vorticity magnitude compared to that seen at the other two locations. The leakage coolant interacts less with the hot mainstream flow, improving cooling effectiveness directly downstream of the slot and near the suction side leading edge. As shown in Fig. 12(b), the vortex center is located at approximately \(z/S = 0.01\) as compared to \(z/S = 0.035\) for the further upstream locations. Figure 12(b) also indicates the presence of a small tertiary vortex forming upstream of the ejecting coolant similar to that seen previously when the slot was at the nominal location.

Similar to the area averages performed to assess the performance of slot orientation on the endwall cooling performance, similar area averages were taken over the endwall to quantify the effects of slot location. Figure 13 presents the area averaged effectiveness values for the three slot locations shown in Fig. 11. Note that the averaging area for the further upstream and downstream locations began directly downstream of the leakage slot across one passage pitch, extending to the passage throat as indicated in Fig. 5(a). There is only a small percent difference increase in area averaged effectiveness of 4% between the nominal and further upstream locations. At the closer location, however, a percent difference increase of approximately 30% is realized relative to the nominal location.

![Fig. 11 Comparison of adiabatic effectiveness contours for the 90 deg slot orientation at three different locations for MFR = 1.0% and \(l = 2.8\)](image)

![Fig. 12 Average flowfield vectors, streamlines, and contours of vorticity in the stagnation plane with MFR = 1.0% and \(l = 2.8\) at a slot location of (a) \(x/C_{ax} = -0.34\) and (b) \(x/C_{ax} = -0.05\)](image)
Moving the 90 deg slot also had an overall effect on the cooling. While the most upstream position was shown to have little impact on the resulting endwall cooling performance, the position closest to the vane showed an influence. Moving the 90 deg slot closer to the vane stagnation was shown to improve local and area averaged effectiveness. Unlike the two further upstream locations, the size of the induced vortex was greatly reduced while the vorticity intensity increased.

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Nomenclature
- \( C \) = true vane chord
- \( d_{ef} \) = effective leading edge diameter
- \( f \) = momentum flux ratio
- \( M \) = mach number
- \( MFR \) = leakage mass flux ratio
- \( N \) = total number of measurements
- \( n_i \) = number of measurements in a given bin size
- \( P \) = vane pitch
- \( Re \) = Reynolds number, \( U_{in}C/\nu_{air} \)
- \( S \) = vane midspan height
- \( T \) = static temperature
- \( t \) = time
- \( U \) = streamwise velocity
- \( V \) = spanwise velocity
- \( w \) = slot width
- \( x \) = axial direction
- \( y \) = pitch direction
- \( z \) = span direction

Greek
- \( \alpha \) = flow angle
- \( \delta \) = boundary layer thickness
- \( \delta' \) = displacement thickness
- \( \eta \) = corrected adiabatic effectiveness, \( (\eta_{exp} - \eta_g)/(1 - \eta_g) \)
- \( \eta_{exp} \) = measured adiabatic effectiveness, \( (T_{\infty} - T_{aw})/(T_{\infty} - T_g) \)
- \( \theta \) = momentum thickness
- \( \rho \) = air density
- \( \nu \) = kinematic viscosity

Subscripts
- \( aw \) = adiabatic wall
- \( ax \) = axial chord
- \( c \) = coolant
- \( exit \) = exit of vane passage at throat
- \( in \) = measured at inlet of the vane cascade
- \( \infty \) = freestream velocity at the entrance to the test section
- \( \theta \) = based on momentum thickness

References


