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

Gas turbine designs seek improved performance by modifying the endwalls of nozzle guide vanes in the engine hot section. Within the nozzle guide vanes, these modifications can be in the form of an axisymmetric contour as the area contracts from the combustor to the turbine. This paper investigates the effect of axisymmetric endwall contouring on the cooling performance of a film cooled endwall. Adiabatic effectiveness measurements were performed in a planar passage for comparison to a contoured passage, whereby the exit Reynolds numbers were matched. For the contoured passage, measurements were performed both on the flat endwall and on the contoured endwall. Fully expanded film cooling holes were distributed on the endwall surface preceded by a two-dimensional slot normal to the inlet axis. Results indicated that the coolant coverage from the upstream leakage slot was spread over a larger area of the contoured endwall in comparison to the flat endwall of the planar passage. Film cooling effectiveness on the flat endwall of the contoured passage showed minimal differences relative to the planar passage results. The contracting endwall of the contoured passage, however, showed a significant reduction with average film cooling effectiveness levels approximately 40% lower than the planar passage at low film cooling flow rates. In the case of all endwalls, increasing leakage and film cooling mass flow rates led to an increase in cooling effectiveness and coolant coverage. [DOI: 10.1115/1.4002965]

1 Introduction

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

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

This paper discusses the effect of axisymmetric endwall contouring on the cooling of a low-pressure, turbine vane endwall. The effect of varying the mass flow rate through the upstream leakage slot and film cooling holes is also considered. Note that this paper is part of a larger study that also investigates the effect of axisymmetric endwall contouring on the heat transfer characteristics of the guide vane endwall. A second paper, Thrift et al. [1], presents heat transfer results on the endwalls of the planar and contoured passages over a range of leakage flow rates.

2 Review of Relevant Literature

Many variations of endwall contouring exist. Endwall contouring can be applied to either one or both endwalls. Contouring can be upstream of the vane passage, upstream and within the vane passage, or only within the vane passage. Endwall contouring can be classified as either two- or three-dimensional. Two-dimensional, also referred to as axisymmetric endwall contouring, is a modification to the endwall surface along the axial direction only. This type of contouring typically involves a modification to the entire endwall in the form of a linear or nonlinear surface. A three-dimensional contour, otherwise known as a nonaxisymmetric contour, makes use of localized hills and valleys to reduce the radial pressure gradient at the endwall, leaving the mainstream flow mostly unchanged. While three-dimensional contouring has shown to be beneficial for aerodynamic losses and endwall film cooling (Knezevic et al. [2], Praisner et al. [3], Gustafson et al. [4,5], and Okita and Nakamata [6]), it has generally only been applied to turbine blade endwalls and is not further considered here. The following presents findings related to axisymmetric endwall contouring, where all the contouring is located on one endwall as in the current study.

Dossena et al. [7] performed a detailed investigation of the flow field in a three vane, linear cascade with axisymmetric contouring of one endwall in the form of a nonlinear slope. The passage had a height contraction of 30%, beginning at the passage inlet and ending at the exit. A comparison between the contoured and flat endwalls showed that the secondary vortex structures were strongly affected by the endwall contour. On the flat side, the structure was similar to that of a planar passage. On the contoured endwall, the structure deviation inhibited the formation of the pas-
sage vortex and its growth toward midspan. Consequently, the contoured passage produced substantially lower loss levels than the planar passage.

Barigozzi et al. [8] investigated the effects of endwall film cooling on the aerodynamic performance of a seven vane, linear cascade with axisymmetric endwall contouring. The contoured passage was similar to that used by Dossena et al. [7], with a height contraction of 30%, which started at the passage inlet and ended at the exit. Contoured endwall results, with and without film cooling, were compared with previously obtained results for a planar passage. Flow field measurements downstream of the contoured passage showed a nonsymmetric energy loss distribution, with a reduced wake compared with the planar case. Near the flat side of the contoured passage, the energy loss resembled the planar distributions, clearly showing the loss core of the passage and corner vortex. Near the contoured side, however, the passage vortex appeared to be suppressed at the endwall and joined to the corner vortex. Barigozzi et al. [8] reported that the overall loss associated with the contoured passage was 20% lower than the overall loss of the planar passage. The overall loss reduction was attributed to a reduction in profile loss as there was a reduction in secondary losses on the planar side yet an increase of approximately the same amount on the contoured side. For both the planar and contoured passages, coolant injection modified secondary flows, reducing their intensity. Similar to Dossena et al. [7], however, secondary losses on the contoured side were always larger than those on the planar side, with global losses of the contoured passage being smaller than the planar case.

Lin et al. [9,10] performed a computational study to investigate the three-dimensional flow in a nozzle guide vane passage with leakage flow from an upstream slot incorporating a backward-facing step. The vane passage had one flat and one axisymmetric convergent endwall. Both endwalls incorporated endwall leakage flow from an upstream slot. For the contoured endwall, two configurations of the same 45 deg contour were investigated. In one configuration, the contraction was upstream of the vane passage. In the other, the contraction started upstream and continued through the vane passage. With gap leakage, secondary flows were found to be reduced at all endwalls for both contoured configurations. Without gap leakage, however, secondary flows were only reduced along the contoured endwall in which the contraction started upstream of the vane and continued through the passage. Adiabatic effectiveness calculations showed that leakage flow was significantly more effective in providing coolant coverage to both endwalls, flat and contoured, in the configuration where the contraction started upstream of the vane and continued through the passage.

A recent experimental study by Yang et al. [11] investigated the cooling performance of slot flow ejecting tangential to a surface over a range of inclination angles. Yang et al. [11] showed that when the film cooled surface was convergent at an angle of 10 deg or less, cooling effectiveness was reduced compared with when the film cooled surface was horizontal. Yang et al. [11] suggested that the mainstream flow may impinge the film cooled surface and destroy the film jet structure, reducing cooling effectiveness. When the convergent angle was greater than 10 deg, however, an increase in film cooling performance was observed. Relaminarization of the turbulent flow in the mixing zone from increased acceleration was attributed to the increase in cooling effectiveness.

To date, most studies have focused on the aerodynamic effects of axisymmetric endwall contouring. Few studies have investigated the subsequent effects of axisymmetric endwall contouring on film cooling performance. The study reported in this paper seeks to understand the effects of axisymmetric endwall contouring on the cooling characteristics of a vane endwall by providing detailed measurements of adiabatic effectiveness between planar and contoured passages, where the contraction begins upstream of the vane stagnation.

3 Experimental Methods and Benchmarking

All measurements were obtained in a test section containing three, scaled-up nozzle guide vanes. Flow was supplied to the test section by a low speed, closed loop wind tunnel, depicted in Fig. 1. Driving the flow through the wind tunnel was a variable speed, 50 hp fan. Downstream of the fan, the flow was turned by a 90 deg elbow before passing through the primary, finned-tube heat exchanger used to cool the main flow. After being turned by another 90 deg elbow, the air was split into three flow paths. The flow passing through the center passage simulated heated core flow. The flow that passed through the upper and lower passages was used as coolant for both leakage and film cooling flows. Air in each of the two outer flow paths traveled through secondary finned-tube heat exchangers, where additional cooling could take place before passing into respective plenums. Coolant air was drawn from the upper plenum only and into the appropriate local coolant plenums on the attached test section using a 2 hp blower. After the flow split, the core flow passed through a heater bank, a series of screens used for flow straightening, and then into a contracted straight flow section with a rounded inlet. At the exit of this section was the experimental test section, where all measurements were performed. The test section incorporated a 90 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. The test section contained two full nozzle guide vanes and a third partial vane connected to a flexible wall to maintain the desired pressure distribution along the center vane. The vanes were scaled-up by a factor of 2.4 from actual engine size to achieve high measurement resolution. Three different configurations of the test section were used for testing: a planar passage, a contoured passage with the contoured endwall as the ceiling, and a contoured passage with the contoured endwall as the flooring. Figure 2 presents a schematic illustrating the three endwall configurations. In the case of the contoured passage, the relocation of the contoured endwall as either the ceiling or floor was necessary to study both the flat and contoured endwalls as measurements could only be performed on the floor due to optical access. For the contoured passage, the area contraction began 1.25Cax upstream of the cascade inlet with a linear slope of 16 deg. The contraction continued through the passage, giving an area reduction of 83% from inlet to outlet.

Table 1 presents the vane geometry and flow conditions for both the planar and contoured passages. Note that the exit Reynolds number was matched between the planar and contoured passages. Matching exit Reynolds numbers required a lower inlet velocity for the contoured passage due to the additional spanwise area contraction.

The boundary layer entering the cascade was measured at a location 4.25Cax upstream of the vane stagnation. Table 2 lists the turbulent inlet boundary layer parameters, which were maintained throughout this study. The measured turbulence intensity was lower than that typically found in an engine at 1.0%. The effect of turbulence was not considered in this study as past studies by the authors have evaluated this effect [12].

To simulate the leakage interface between a combustor and a turbine, a two-dimensional 90 deg slot was placed 0.17Cax upstream of the vane cascade in the bottom endwall. Figure 2 docu-
ments the location and dimensions of the upstream leakage slot for the three endwall configurations. Also shown in Fig. 2 is the location of slot thermocouples, used to measure the local coolant temperature of the leakage flow. The thermocouples were placed one slot width below the external surface and within the upstream slot with the thermocouple head being positioned at the center of

![Fig. 2 Schematic of the (a) planar passage, (b) contoured passage with contour on the ceiling, and (c) contoured passage with contour on the floor](image)

### Table 1 Vane geometry and flow conditions

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### Table 2 Inlet boundary layer characteristics

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<td>Boundary layer thickness/span (δ/S)</td>
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the slot width. The thermocouples were arranged along the slot pitch in increments of 0.125P, with a thermocouple located at each vane stagnation corresponding to y/P = 0, -1, and 1.

In addition to coolant flow from an upstream slot, the vane endwall also incorporated film cooling. The film cooling pattern made use of laid back, fan shaped holes inclined at 26 deg to the surface with a laid back and half-diffusion angle of 13 deg. As illustrated in Fig. 3, the film cooling holes were distributed along the suction side leading edge and pressure side of the vane endwall. Note that the film cooling holes are distributed only in one passage and partially in the other to maintain periodicity.

### 3.1 Coolant Flow Settings

For every test condition, the inlet velocity distribution and the pressure distribution along the vane midspan were verified. The inlet velocity distribution was measured approximately 0.8c upstream of the cascade inlet. Inlet velocities measured across the width of the cascade varied less than 5%. Figure 4 compares the measured and predicted pressure distributions around the midspan of the center vane for both the planar and contoured passages. Note that the midspan plane was defined in the nonconverged, inlet passage, as indicated in Fig. 2.

Figure 4 indicates that the inviscid flow field around the center vane was matched to the predicted curves. The predictions were obtained from a computational study performed for incompressible, viscous, low speed conditions using FLUENT [13]. The computational domain was such that the vane was divided at the stagnation point and the trailing edge with a single flow passage being modeled. Periodic boundary conditions were placed along the pitchwise boundaries of the computational domains. The predicted pressure distributions were representative of a vane within a continuous linear cascade. A detailed description of the computational method used during this study was provided by Thrift et al. [1]

For the leakage flow, the issuing coolant entering each passage was a prescribed percentage of a single passages mass flow rate. For the film coolant, however, only the rightmost passage received

![Fig. 3 Schematic of the film cooling hole distribution](image)

![Fig. 4 Midspan pressure distributions along the center vane for both the planar and contoured passages starting at the origin of the center vane](image)
a prescribed coolant percentage. For this study, coolant flow rates of 0.3%, 0.5%, and 0.7% were investigated from both the up-stream leakage slot and film cooling holes for all three endwall configurations illustrated in Fig. 2. In matching coolant mass flux ratio between the three endwall configurations, the relative amount of coolant is the same for each passage. A total of nine experiments were conducted for each endwall.

The mass flux issuing from the upstream leakage slot was measured using a laminar flow element, located within the supply pipe to the slot plenum. Unlike the leakage mass flow, the mass flow rate from the film cooling holes was determined using the local blowing ratios. Equation (1) provides the relationship for calculating the mass flow from a single film cooling hole. Note that the mass flow depends on the location of the film cooling hole as dictated by the local freestream velocity and blowing ratio. In addition, a discharge coefficient is required to obtain the true mass flow through the film cooling hole. Gritsch et al. [14] indicated that the discharge coefficient from a laidback, fan shaped hole was essentially constant at a value of 0.8 for low Mach number flows over a large range of pressure ratios. Given the conditions of the current study, a discharge coefficient of 0.8 was assumed for all film cooling holes.

\[
m_{FC} = C_d \rho_a A_o \frac{U_{in}}{\rho_c} M_l
\]

To calculate the local blowing ratio, \(M_l\) was recast in terms of a pressure difference between the inlet stagnation pressure and the film cooling plenum, as shown in Eq. (2) [15].

\[
M_l = \sqrt{\frac{\rho_c}{\rho_a}} \left( \frac{P_{plenum} - P_{o,in}}{0.5 \rho_c U_{in}^2 (1 - C_p) + 1} \right)
\]

Calculation of the local blowing ratio required the local freestream velocity and local pressure coefficient. The local inviscid velocity was obtained from the computational fluid dynamics (CFD) results at the midspan plane at a given film cooling hole location in the x-y plane. The local pressure coefficient was also obtained from the CFD predictions along the endwall at a given film cooling location.

Blowing ratios were large near the vane stagnation due to low freestream velocities. This is illustrated in Fig. 5, which plots contours of local blowing ratio for the flat endwall of the planar passage over a range of FC flow rates. In addition, Fig. 5 plots the local blowing ratios for the flat and contoured endwalls of the contoured passage at an intermediate film cooling flow rate. Near the pressure side leading edge and progressing toward the passage throat, blowing ratios gradually decreased as the freestream velocity increased. Figure 5 shows that as the film cooling flow rate was increased, the local blowing ratio for each film cooling hole increased as well. The endwalls of the contoured passage were subject to slightly smaller blowing ratios, however, the distribution of the blowing ratios with increasing film cooling was similar to that within the planar passage. Mass flow rates were small in the regions of stagnation, where blowing ratios were relatively high compared with the film cooling holes within the vane passage.

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

An inframetrics P20 IR camera was used to acquire the spatially resolved adiabatic temperatures on the bottom endwall. To capture the entire endwall surface, measurements were taken at 14 different viewing locations, which is a method similar to that used by previous researchers [16–19]. Five images were taken at each location and averaged to produce the final image. From a camera distance of 55 cm, each picture covered an area that was 24 × 18 cm² with the viewing area being divided into 320 × 240 pixel locations resulting in a spatial integration of 0.75 mm (0.39 FC hole diameters).

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

A one-dimensional conduction correction, as described by Ethridge et al. [20], was applied to all adiabatic effectiveness measurements. The correction involved measuring the endwall surface effectiveness with no coolant flow. A correction value of \(\eta_c = 0.1\) was measured within the vane passage. At the passage throat, the correction value was found to be \(\eta_c = 0.05\). Upstream of the leakage slot, a correction value of \(\eta_c = 0.15\) was measured. The endwall upstream of the leakage slot was made of medium density fiberboard with a thermal conductivity (0.13 W/m K) higher than that of the foam making up the cooled surface, resulting in larger conduction losses.

3.3 Uncertainty Analysis. An uncertainty analysis was performed on the measurements of adiabatic effectiveness using the partial derivative method described by Moffat [21]. A precision uncertainty for the adiabatic surface temperatures was determined.
by taking the standard deviation of six measurement sets of IR camera images. Each image set consisted of five images, as mentioned previously. Based on a 95% confidence interval, the precision uncertainty of the measurements was ±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). In this way, a bias uncertainty of ±0.54°C was determined. Combining the bias and precision uncertainties, a total uncertainty of ±0.58°C was estimated for the IR surface temperature measurements. The uncertainty in adiabatic effectiveness was then found based on the partial derivative of $\eta$ with respect to each temperature in the definition and the total uncertainty in the measurements. An uncertainty of $\sigma_{\eta} = 0.02$ was calculated over the range $\eta=0.03–0.7$.

4 Discussion of Results

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

4.1 Effects of Endwall Contouring on Leakage Flow

Adiabatic effectiveness experiments were performed over a range of leakage and film cooling flow rates. To isolate the effect of endwall contouring on the upstream leakage flow, only those experiments with the lowest film cooling flow rate are considered. In this way, the influence of the film coolant on the upstream leakage flow is minimized. Figure 6 compares contours of adiabatic effectiveness among the three endwalls at various leakage flow rates with a film coolant flow rate of FC=0.3%.

For each endwall in this study, the ejection of leakage flow from the upstream slot is concentrated near the suction side leading edge. This can be attributed to the relatively low static pressure on the endwall near the suction side. Lynch and Thole [22], in comparison to a study performed by Knost and Thole [23], showed that moving the leakage slot further upstream away from the influence of the vane, led to more uniform coolant ejection. Another leakage flow pattern that is common among all conditions in Fig. 6 is the sweeping of the leakage flow from the pressure to the suction side. This pattern is caused by the endwall cross-flow and passage vortex, entraining coolant and sweeping it to the suction side of the vane.

Figure 6 shows that increasing the leakage flow from the upstream slot results in increased coolant coverage for each endwall. As the leakage flow rate increases, the pressure difference between the slot plenum and the local static pressure at the exit of the slot increases. This allows the coolant flow to overcome the relatively high static pressure near the vane leading edge.

As mentioned previously, the surface upstream of the leakage slot was made of a material with a thermal conductivity approximately 4.5 times that of the polyurethane foam on the downstream surface. As observed in Fig. 6, this led to effectiveness values around 0.1 directly upstream of the leakage slot, where the driving temperature difference was high.

Heat transfer measurements by Thrift et al. [1] on the passage endwalls highlight the importance of the path taken by the leakage flow. Results indicate that heat transfer is largest near the suction side of the vane leading edge. In addition, the heat transfer in this region is shown to increase with increasing leakage flow rate.

A slight reduction in coolant coverage from the upstream leakage flow is observed for the flat endwall of the contoured passage relative to the flat endwall of the planar passage. Although each slot plenum receives the same percentage of coolant flow, the endwall static pressure in the case of the flat endwall of the contoured passage is such that the slot plenum pressure is significantly lower. A reduction in slot plenum pressure for the flat endwall of the contoured passage results in slightly less coolant coverage at each leakage flow rate relative to the flat endwall of the planar passage, as shown in Fig. 6.

The contoured endwall of the contoured passage shows the largest coolant coverage from the upstream slot at each leakage flow rate in comparison to the flat endwalls in both the planar and contoured passages. This can be attributed to the reduced coolant ejection angle for the upstream slot in the contoured endwall. Figure 2 shows that the upstream leakage slot in the contoured endwall is ejecting the coolant at an angle of 74 deg relative to the contoured endwall. Ejecting coolant at a reduced angle reduces the penetration depth of the leakage flow into the mainstream, resulting in better coverage of the endwall surface. Predictions of the three-dimensional flow field were performed with all three endwalls by Thrift et al. [1] to investigate the horseshoe vortex. To illustrate the difference in penetration depth between the leakage flow issuing from the flat endwall of the planar passage and the contoured endwall of the contoured passage, Fig. 7 provides streamtraces along the vane stagnation plane obtained from the CFD simulations at a leakage flow of 0.75%. As shown in Fig. 7, the coolant penetrates less into the freestream for the contoured endwall, resulting in a smaller horseshoe vortex. This finding is similar to those computational results provided by Lin et al. [10], who also showed increased coolant coverage from a 90 deg upstream slot on the contoured endwall relative to the flat. As in this study, Lin et al. [9] kept the slot orientation at 90 deg relative to the main flow direction, resulting in a reduced ejection angle for their sloped wall case of 45 deg.

As was discussed previously, the relative close location of the upstream slot to the vane stagnation results in leakage flow being...
ejected near the suction side of the vane leading edge. In addition, the high endwall static pressure at the vane stagnation can lead to ingestion of hot mainstream flow into the upstream slot. Hot mainstream flow is detrimental to the upstream leakage slot and the corresponding supply plenum.

To investigate the severity of this ingestion among the three endwalls, temperature measurements were recorded from the thermocouples measuring the slot flow temperature near the exit of the slot. The resulting measurements are presented in Fig. 8 as nondimensional temperatures across the cascade pitch for each endwall configuration at a leakage flow of 0.3% and 0.7%.

Figure 8 shows that the contoured endwall has less flow ingestion than the flat endwalls of the planar and contoured passages. The flat endwall of the planar passage shows a slight improvement in mainstream ingestion relative to the other flat endwall at the lowest leakage flow rate. With an increase in leakage flow to 0.7%, however, the flat endwall of the contoured passage remains the only endwall to experience ingestion into the upstream slot at stagnation. The consequence of flow ingestion can be observed in Fig. 6(c) at the vane stagnation. Nonzero effectiveness levels downstream of the slot at vane stagnation for the flat endwall of the planar passage and the contoured endwall indicate that leakage flow was ejecting from the slot. For the flat endwall of the contoured passage, however, no leakage flow is present at the vane stagnation.

As mentioned previously, predictions of the three-dimensional flow field were performed with all three endwalls by Thrift et al. [1]. Included in the simulations were predictions of adiabatic wall temperatures from leakage flow. Figure 9 presents the predicted adiabatic effectiveness levels on each endwall at comparatively low and high leakage flow rates. The predicted adiabatic effectiveness levels display the same trends that were observed experimentally for the leakage flow. The majority of the coolant is ejecting near the suction side leading edge with the coolant sweeping from the pressure to the suction side of the passage. In addition, the predictions show that the coolant coverage and effectiveness levels increase with leakage mass flow rate. Also captured by the simulations is the improvement in coolant performance from the leakage flow for the contoured endwall in comparison to the flat endwalls of the planar and contoured passages.

4.2 Effects of Endwall Contouring on Film Cooling. As in Sec. 4.1, to isolate the effect of endwall contouring on the film cooling, only those experiments with the lowest leakage flow rate of 0.3% are considered. This is to minimize the effect that leakage flow has on the film cooling.

Figure 10 compares contours of adiabatic effectiveness among the three endwalls at various film cooling flow rates. Figure 10 shows that the coolant coverage from the pressure side film cooling holes is shifted from within the passage toward the passage throat with an increase in film cooling flow. As the mainstream flow approaches the passage throat, it accelerates, reducing the local blowing ratios. As the film cooling flow rate is increased, however, the local blowing ratio for each film cooling hole increases, as shown previously in Fig. 5. Larger blowing ratios combined with increased film coolant ejection leads to an increase in coolant coverage near the passage throat.

Figure 10 shows that the film coolant coverage is poorest for the contoured endwall at each film cooling flow rate. To visualize...
these results in a more quantitative manner, values were extracted from the endwall data along the paths of inviscid streamlines obtained from a CFD prediction of the passage flow field using FLUENT. The inviscid streamlines originate at the midspan of the vane at two pitch locations corresponding to \( y/P = 0.25 \) and 0.5, as shown in Fig. 10(a). The streamlines pass directly through the endwall area influenced by the pressure side film cooling holes. Figures 11–13 present the adiabatic effectiveness levels along these inviscid streamlines for those contours presented in Fig. 10.

Figures 11(a) and 11(b) show that the effectiveness values for the contoured endwall are well below those for the flat endwall of the planar passage at the lowest film cooling flow rate along both the 0.25P and 0.5P streamlines, respectively. While not nearly as severe, the effectiveness values for the flat endwall of the contoured passage are also slightly below those for the flat endwall of the planar passage. Figure 11(b) shows that the contoured endwall effectiveness levels are initially higher than those on the flat endwalls of the planar and contoured passages. Recall that the 0.5P streamline passes through the region influenced by the leakage flow. As discussed previously, the contoured endwall shows improved cooling performance from leakage flow in comparison to the flat endwalls.

With an increase in film cooling to 0.5%, the disparity between the effectiveness levels on the contoured endwall and the flat endwall of the planar passage is reduced, as shown in Figs. 12(a) and 12(b). Effectiveness values for the contoured endwall, however, are still below the flat endwall results at 0.5% FC, especially approaching the passage throat. In addition, the effectiveness values between the flat endwalls are very similar. Figure 12(b) highlights the initial improvement in effectiveness level from the leakage flow for the contoured endwall.

With a further increase in the film cooling flow to 0.7%, the difference in effectiveness levels between the contoured endwall and the flat endwalls continues to get smaller, as shown in Figs. 13(a) and 13(b). The contoured endwall continues to exhibit poorer effectiveness values compared with the flat endwalls. Contrary to the results at the lowest film cooling flow rate, the effectiveness values on the flat endwall of the contoured passage are now slightly higher than those on the flat endwall of the planar passage.

The flat and contoured endwalls of the contoured passage are subject to different flow conditions. Relative to the planar passage, the flat endwall of the contoured passage is subject to the highest flow acceleration and the lowest static pressure. This is a result of the spanwise area contraction through endwall contouring. The flat endwall of the contoured passage is similar to a flat plate with a favorable pressure gradient along the streamwise direction. Schmidt and Bogard [24] and Teekaram et al. [25] investigated the effect of a favorable pressure gradient on the adiabatic effectiveness downstream of a single row of film cooling holes. Both studies indicated negligible changes in the adiabatic effectiveness between the zero and nonzero pressure gradient cases in the ejection range, where the jets were either fully attached or detached.
the range in which the jets were beginning to detach, however, the
studies indicated a significant increase in adiabatic effectiveness
downstream of the ejection row. In the current study, the use of
laid back, fan shaped film cooling holes makes it unlikely that the
coolant flow is beginning to separate. As such, it is not unexpected
that the flat endwall results of the contoured passage are similar to
the flat endwall of the planar passage.

The same is not true for the contoured endwall, which, in ad-
dition to the streamwise acceleration, is also subject to a second
flow condition, streamline curvature. As the flow approaches the
contoured passage, the streamlines converge, not only in the pitch-
wise direction but along the span in response to the area contrac-
tion as well. Near the contoured endwall, the streamlines must
turn toward midspan as they approach and impinge on the end-
wall. The impingement of the mainstream flow occurs in the up-
stream regions, where the flow is still mostly uniform. As shown
in Fig. 5, the blowing ratios for those holes near the pressure side
leading edge are smaller on the contoured endwall than on the flat
endwall of the planar passage. As a result, the cooling perfor-
mance of the pressure side film cooling holes on the contoured
endwall is poor for those holes further upstream, as shown in Fig.

10. The cooling effectiveness increases for those holes further
downstream, near the throat of the vane passage.

5 Conclusions

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

The influence of the vane and endwall secondary flows on up-
stream slot coolant was evident in the ejection pattern of the leak-
age flow. Leakage flow ejection was not uniform; instead, most of
the coolant was ejected near the suction side leading edge of the
vane. After ejection, the coolant was abruptly swept from the pres-
sure side of the passage toward the suction side. With slot and film
cooling flows present, there was a large region near the suction
side leading edge that was overcooled. The presence of the film
cooling holes in this region is redundant as the leakage flow, even
at the lowest flow rate, adequately covers the surface. It seems

Fig. 12 Comparison of adiabatic effectiveness levels between the three endwalls for 0.5% FC and 0.3% leakage flow, sampled along inviscid streamlines released from (a) 25% pitch and (b) 50% pitch.

Fig. 13 Comparison of adiabatic effectiveness levels between the three endwalls for 0.7% FC and 0.3% leakage flow, sampled along inviscid streamlines released from (a) 25% pitch and (b) 50% pitch.
that it would be beneficial to eliminate those film cooling holes and use the coolant elsewhere such as providing increased coolant mass flow near the vane leading edge.

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

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

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

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Nomenclature

- \( A_k \): area of film cooling hole
- \( C_{\theta} \): true vane chord
- \( C_{D} \): discharge coefficient
- \( C_P \): pressure coefficient, \( (P_{1} - P_{x,m})/0.5pU_{\infty}^2 \)
- \( M \): blowing ratio
- \( m_{FC} \): film cooling mass flow rate
- \( P \): vane pitch or pressure
- \( Re \): Reynolds number, \( U_{\infty}C_{\theta}/\nu_{\text{air}} \)
- \( S \): vane span or distance along vane circumference
- \( s \): distance along streamline
- \( T \): static temperature
- \( U \): velocity
- \( x \): axial direction
- \( y \): pitch direction
- \( z \): span direction

Greek

- \( \delta \): boundary layer thickness
- \( \delta^* \): displacement thickness
- \( \eta \): corrected adiabatic effectiveness, \( (\eta_{\exp} - \eta_{b})/(1 - \eta_{b}) \)

\( \eta_{\exp} \): measured adiabatic effectiveness, \( (T_{aw} - T_{aw(0)})/(T_{aw} - T_{\infty}) \)

- \( \theta \): 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
l: local value
or: refers to no cooling case or total pressure
s: static measurement
slot: measured in the leakage slot or slot plenum
\( \infty \): freestream velocity at the entrance to the test section

\( \theta \): based on momentum thickness

References

13. FLUENT (Version 6.2.1), Fluent Inc., Lebanon, NH.

