Combustor Turbine Interface Studies—Part 1: Endwall Effectiveness Measurements

Improved durability of gas turbine engines is an objective for both military and commercial aerospace engines as well as for power generation engines. One region susceptible to degradation in an engine is the junction between the combustor and first vane given that the main gas path temperatures at this location are the highest. The platform at this junction is quite complex in that secondary flow effects, such as the leading edge vortex, are dominant. Past computational studies have shown that the total pressure profile exiting the combustor dictates the development of the secondary flows that are formed. This study examines the effect of varying the combustor liner film-cooling and junction slot flows on the adiabatic wall temperatures measured on the platform of the first vane. The experiments were performed using large-scale models of a combustor and nozzle guide vane in a wind tunnel facility. The results show that varying the coolant injection from the upstream combustor liner leads to differing total pressure profiles entering the turbine vane passage. Endwall adiabatic effectiveness measurements indicate that the coolant does not exit the upstream combustor slot uniformly, but instead accumulates along the suction side of the vane and endwall. Increasing the liner cooling continued to reduce endwall temperatures, which was not found to be true with increasing the film-cooling from the liner. [DOI: 10.1115/1.1561811]

Introduction

Today’s world heavily depends on gas turbine engines for military and commercial propulsion as well as for industrial applications. This dependence demands engines that are both durable and produce large amounts of power-two requirements that are somewhat conflicting. The reason for this conflict is that high temperatures are needed at the entrance to the rotor, which in turn cause reduced component life. One critical region in an engine is the combustor-turbine junction. Compressor designs, particularly for aerospace engines, typically have film holes or slots that provide cooling along the platform of the combustor. The coolant that is injected out of the liner produces highly nonuniform temperature and pressure radial profiles as it exits the combustor and enters into the turbine. These nonuniformities typically span 10–15% of the inner and outer radii at the platforms of the first vane. In addition, most combustor-turbine interfaces include, depending on the design, a backward-facing slot, a forward-facing slot, or a flush slot through which coolant leaks into the main gas path. The interaction between the liner and slot flows as well as interaction with the highly turbulent mainstream, which is produced from combustor dilution holes, dictates the total pressure profile coming in to the passage of the first vane. In turn, this total pressure profile in the near-platform region dictates the development of the secondary flow through the vane passage. Moreover, these secondary flows dictate the heat transfer to the platform of the first vane and the exit flow angles for the rotor.

To improve the understanding of this complex flow, high-resolution measurements were taken using a combined large-scale combustor simulator and first vane. The combustor simulator includes large dilution holes, film-cooling holes along the approaching liner, and a backward-facing slot at the combustor-turbine interface. Although this simulator does not include the effects of combustion, these studies were directed at understanding the effects of representative combustor flow fields on the secondary flow development in the first vane passage. The combustor simulator produces non-uniformities in both the span (radial) and pitch (circumferential) directions through the use of the liner film-cooling, insuring representative near-platform flows, and dilution jets, insuring the appropriate levels of mainstream turbulence.

This paper is the first of a two-part series that investigates the effects of different combustor liner flows on adiabatic effectiveness levels on the downstream vane platform. Part 1 presents a discussion of the inlet conditions that were simulated as well as measured adiabatic wall temperatures along the vane platform. Part 2 presents flow and thermal field measurements for selected cases of interest. In general, the objectives for the work presented in this paper were the following: i) to evaluate the effect of various liner flows on the exit total pressure profiles and ii) to quantify adiabatic wall temperatures as a function of different liner cooling flows and different slot flows.

Summary of Past Literature

Many past studies have led to a range of secondary flow models in turbine airfoil passages. Secondary flows refer to velocity components not aligned with the inviscid flow through the passage. Driving the secondary flows are two pressure gradients: the inherent pressure gradient from turning the flow and the pressure gradient resulting from nonuniform inlet conditions along the radial span of the airfoil passage. Langston [1] proposed an accurate representation of the secondary flows for a vane with an approaching two-dimensional turbulent boundary layer along the endwall. A horseshoe vortex is formed by a downturning of the flow as it approaches the vane leading edge. The pressure side leg of the horseshoe vortex turns into the passage vortex while the suction side leg of the horseshoe vortex is suppressed by the passage vortex as it conveys through the passage. It is questionable whether or not the secondary flows in today’s engines behave in this manner because the inlet conditions to the first vane are much different than the turbulent boundary layer assumption made by Langston and others.

While there are no studies that have accounted for multiple rows of inclined film-cooling holes placed in a combustor liner upstream of a backward-facing slot, there have been a few studies...
that have measured endwall heat transfer as a result of injection from a two-dimensional, flush slot just upstream of the vane. Blair [2] measured adiabatic effectiveness levels and heat transfer coefficients for a range of blowing ratios through a slot placed just upstream of the leading edges of his single passage flow channel. One of the key findings was that the endwall adiabatic effectiveness distributions showed extreme variations across the vane pitch. Much of the coolant was swept across the slot toward the suction side corner resulting in reduced coolant near the pressure side. As the blowing ratio was increased, he found that the extent of the coolant also increased. Measured heat transfer coefficients were similar between no slot and slot injection cases. In a later study by Granser and Schulenberg [3], similar adiabatic effectiveness results were reported with higher values occurring near the suction side of the vane.

A series of experiments have been reported for various injection schemes upstream of a nozzle guide vane with a contoured endwall by Burd and Simon [4], Burd et al. [5], Oke et al. [6,7]. In the studies in [4,5,7] coolant was injected from an interrupted, flush slot that was inclined at 45 deg just upstream of their vane. Similar to others, they found that most of the slot coolant was directed toward the suction side at low slot flow conditions. As they increased the percentage of slot flow to 3.2% of the exit flow, however, their measurements indicated better coverage occurred across the pitch of the airfoil endwall. Their flow field measurements indicated that as the bleed flow was increased the secondary flow structure moved from the pressure surface toward the suction surface for the contoured endwall. Above 3.2% their results indicated no increased thermal benefit. In contrast, the study by Oke et al. [6] used a double row of film-cooling holes that were aligned with the flow direction and inclined at 45 deg with respect to the surface while maintaining nearly the same optimum 3% bleed flow of their previously described studies. They found that the jets lifted off the surface producing more mixing thereby resulting in a poorer thermal performance than the single slot.

Similarly, Zhang and Jaiswal [8] made comparisons between the thermal performance of a slot and two rows of film-cooling holes upstream of a vane along the endwall. They also report a migration of the coolant towards the suction side of the vane. Their results indicated uniform coverage was achieved at high slot flows (rather than film-cooling jet flows), while higher effectiveness levels at the trailing edge were achieved at high film-cooling flows.

Roy et al. [9] compared their experimental measurements and computational predictions for a flux cooling slot that extended over only a portion of the pitch directly in front of the vane stagnation. Contrary to the previously discussed studies, their adiabatic effectiveness measurements indicated that the coolant migrated toward the pressure side of the vane. Their measurements indicated reduced values of local heat transfer coefficients at the leading edge when slot cooling was present relative to no slot cooling.

Kost and Nicklas [10] and Nicklas [11] combined an upstream slot with film-cooling holes in the downstream vane passage to examine the effects of each on the secondary flow field and platform heat transfer at transonic conditions. One of the most interesting results from this study was that they found for the slot flow alone, which was 1.3% of the passage mass flow, the horseshoe vortex became more intense. This increase in intensity resulted in the slot coolant being moved off of the endwall surface and heat transfer coefficients that were over three times that measured for no slot flow injection. They attributed the strengthening of the horseshoe vortex to the fact that for the no slot injection the boundary layer was already separated with fluid being turned away from the endwall at the injection location. Given that the slot had a normal component of velocity, injection at this location promoted the separation and enhanced the vortex. Their adiabatic effectiveness measurements indicated higher values near the suction side of the vane due to the slot coolant migration.

All of the studies involving interface slot coolant flows previously discussed used a slot that was flush with the endwall surface. In many modern engine designs, a step exists at the combustor-vane interface that provides a flow path for coolant leakage. In addition, these past studies have not considered the effects of coolant injection from the multiple rows of cooling holes from the upstream combustor liners. The combined effects of upstream film-cooling holes and slot flow from a backward facing step make this study unique as well as necessary for an improved understanding of the combustor-turbine interface.

Experimental Facilities and Instrumentation

The experiments for this study were performed in a low speed, closed-loop wind tunnel. The development of the vane test section was previously described by Radomsky and Thole [12] while the development of the combustor simulator was previously described by Barringer et al. [13]. The geometric scaling factor for the vane and combustor was 9X, which allows for good measurement resolution in the experiments. Measurements that are presented in this paper include adiabatic endwall temperatures, total pressure profiles, and mean and turbulent velocities.

Experimental Facilities. Figure 1 illustrates the wind tunnel containing the combustor simulator and vane test section. Downstream of a primary heat exchanger is a transition section that divides the flow into three channels that include a heated primary channel, representing the main gas path, and two symmetric secondary channels, representing the coolant flow paths. Within the transition section of the primary channel, the flow immediately passes through a thermal and flow conditioning section containing a bank of heaters followed by a series of screens and flow straighteners. The heater section comprises three individually controlled banks of electrically powered, finned bars supplying a maximum total heat addition of 55 kW. Downstream of the flow straighteners, the heated primary flow enters the combustor simulator. In the combustor simulator, secondary coolant flow is injected into the primary flow passage through liner panels and dilution holes. In addition, the flow is accelerated prior to entering the turbine section. The contraction was designed to match the flow acceleration through that of an aeroengine combustor.

The flow in the secondary passages, also shown in Fig. 1, is first directed through heat exchangers. In addition to heat being rejected from the primary heat exchanger, the heat exchangers in the secondary passages provide additional heat rejection for the coolant flow. The experiments generally required 3–4 h for steady state conditions to be achieved. Throughout the experiments, which took nominally 1 h, all of the temperatures were constantly monitored. Room conditions were maintained throughout the experiment through the use of an air conditioning system, which insured thermal equilibrium with the environment. The flow in the secondary passages is then directed into a large plenum that supplies combustor liner coolant, dilution hole flow, and exit slot.
coolant. The cooling hole pattern in the liners, the dilution holes, and the slot exit are illustrated in Figs. 2(a) and (b). To insure the correct coolant flow splits among the liner panels and dilution rows, separate supply chambers with adjustable shutters were constructed for each liner panel and row of dilution jets. The coolant flow for the exit slot at the end of the combustor panels was maintained by providing the required pressure in the large supply plenum. The mass flow exiting the film-cooling holes was set by applying the appropriate pressure ratio between the supply plenum and the exit static pressure. Using measured discharge coefficients [13], the mass flows through the panels were determined. The mass flows exiting the dilution holes were set by measuring the velocity across the dilution holes. The geometry for the combustor-turbine interface slot, shown in Fig. 2(b) includes feeder holes, two rows of staggered pin fins, followed by the backward facing slot. As will be discussed in the next section of this paper, a range of slot flow conditions were studied. The slot flows were controlled by changing the diameter and spacing of the feeder holes. Figure 2(c) shows the film-cooling hole configuration for the liner panels, while the callout below gives the film-cooling hole geometry for each of the four liner panels. The film-cooling hole diameter for the liners was 0.76 cm, while the liner thickness was 0.9 cm. The diameters of the dilution holes were 8.5 cm and 12.1 cm for the first and second rows, respectively.

The vane test section was designed to model a sector of the turbine. The correct inlet Reynolds number based on chord was set at $2.3 \times 10^5$ for the vane by insuring that the correct mass flow exited the combustor simulator. Exit tailboards were used to insure periodicity for the two passages, which was verified through flow field measurements and static pressure measurements at the vane mid-span. Setting these tailboards required an iterative process during which measurements of the static pressure distribution were compared with computational, inviscid flow predictions using pitchwise periodic boundary conditions. The upstream liner, endwall platform, and vane itself were constructed from urethane foam that had a low thermal conductivity (0.037 W/mK) to allow for measurements of adiabatic surface temperatures. Note that calculations were done to determine whether any heat would be transferred by the warmer fluid above the step to the coolant flow below the step. These calculations indicated a negligible temperature rise ($\sim 10^{-2}$ K) for the coolant fluid.

Instrumentation and Measurement Uncertainty. An infrared camera was used to measure adiabatic wall temperatures on the endwall surface. Measurements were taken at thirteen different viewing locations to insure that the entire endwall surface was mapped. Each picture covered an area that was 19.4 cm by 15.7 cm with the area being divided into 255 by 206 pixel locations. The spatial integration for the camera was 0.37 cm (0.0062 chords). At each viewing location five images were averaged with each image being averaged over 16 frames, which provided a total of 80 data points that were averaged at each pixel location. Overlapping images were also averaged as the complete picture was assembled. Small shiny thumbtacks were embedded in the endwall as location markers to determine the orientation of the picture in a global coordinate system. Each picture was transformed into the global coordinate system and meshed together to obtain the complete temperature representation of the endwall.

Calibration of the infrared camera was made using thermocouples embedded in the endwall along the passage. Thermocouples were also used to monitor freestream, coolant, and endwall temperatures. The coolant temperature and the temperature downstream of the heater were monitored with thermocouples to transform endwall adiabatic temperatures into adiabatic effectiveness levels according to the following equation:

$$\eta = \frac{(T_{aw} - T_{ave})}{(T_{cool} - T_{ave})}$$

where $T_{ave}$ is the mass-averaged temperature exiting the combustor simulator. This mass-averaged temperature was calculated using the coolant temperature weighted with the flow rate in the secondary channels and the heated air temperature weighted by the primary flow rate. The heated air temperature was measured with several thermocouples upstream of the combustor simulator.

Total pressure measurements were taken with a small Kiel probe having a head diameter of 1.6 mm. A small probe diameter was used to insure good measurement resolution for the exit slot, which had a height of 16 mm. The Kiel probe reduced the effect of probe orientation relative to the velocity vectors for a yaw angle range between $\pm 52$ deg and pitch angle range between $\pm 45$ deg. A 0.64 mm H$_2$O pressure transducer converted the pressure difference to a voltage which was digitized with an a/d board housed in a personal computer.

Velocities were measured using a two-component laser Doppler velocimeter (LDV). The LDV was a fiber optic system whereby the probe volume was set by the focusing lens (350 mm) to be 1.3 mm in length and 90 microns in diameter. The flow was seeded with 1 micron diameter olive oil particles. Measured velocities were corrected for bias errors using the residence time weighting correction scheme.

The partial derivative and sequential perturbation methods, described by Moffat [14], were used to estimate the uncertainties of the measured values. Precision uncertainties were calculated.
based on a 95% confidence interval. The velocity measurements that were made to quantify the inlet flow conditions were made using a laser Doppler velocimeter. These measurements included a sample size of 15,000 data points. The precision uncertainty for the streamwise rms velocities was ±2.6%, while the bias uncertainty for the mean velocity measurements was ±1%. Each total pressure measurement used 60,000 data points to compute the mean values. The estimate of bias and precision uncertainties for the mean pressures, which were presented in non-dimensional form was ±1.6%. The bias and precision uncertainties on the adiabatic effectiveness values, using the five-averaged pictures, was ±0.04 giving an uncertainty of ±4.5% at η=0.9 and ±12.6% at η=0.3.

Influences of Panel and Dilution Flows on Combustor Exit Conditions

Prior to making endwall adiabatic effectiveness measurements on the vane platform, a number of measurements were made to determine the sensitivity of the liner and slot panel flows on the exiting total pressure profiles of the combustor simulator. After analyzing these results, a test matrix was designed to determine the sensitivity of the endwall adiabatic effectiveness levels to the panel and slot flows.

The total pressure profiles along the span were measured above the top of the step just upstream of the vane at a pitchwise location in line with the vane stagnation. These total pressure profiles were measured for a range of flow rates exiting the first panel of the combustor liner and are shown in Fig. 3(a). For these tests, the per cent flows exiting the three downstream panels remained the same as the design conditions, which were 3.5%, 2.2%, and 1.5%, respectively. The measured total pressures are presented relative to that occurring at the centerline and are normalized by the local dynamic pressure. Figure 3(a) shows that the pressure is peaked near the top of the slot, becomes a minimum at 20% of the span, and then increases near midspan. The peak near the liner is a result of the film-cooling from the liner panels while the increase near the midspan is a result of the added dilution flow. The measured total pressure results indicate that there is no effect of the increased liner flow on the exit total pressure profile of the combustor simulator. Conversely, Fig. 3(b) shows that the exit total pressure profile is highly dependent on the flow from the fourth panel. As the mass flow from the fourth panel is increased from 1.5 to 3.5% the peak nondimensional total pressure increases from 0.25 to 1.25. Although these results are not shown here, the total pressure was also found to be quite sensitive to the flow rate exiting the third panel.

Fig. 3 Total pressure profiles measured on top of the step at the exit of the combustor indicating the relative effect of the panel flows (a and b) and the dilution holes (c).

Experiments were conducted to determine the sensitivity to the dilution flows by measuring the total pressure profiles at the exit of the combustor above the top of the step on the bottom half of the combustor simulator. Figure 3(c) shows total pressure measurements as a function of only the top dilution row as compared with injection from both the top and bottom dilution holes. Note that the mass flows exiting both the third and fourth panels were increased for these experiments as compared with that shown in Fig. 3(b). The peak in the near wall region exists for both dilution flow cases, even though the dilution injection was changed. Further away from the endwall, the data in Fig. 3(c) indicates that there are higher total pressure values for the top dilution only case as compared with the top and bottom dilution case. These higher total pressures may be attributed to the penetration of the higher total pressure dilution jet closer to the bottom endwall without the resistance of the opposing jet.

Given that there are nearly an infinite number of test cases that could be identified using the combustor simulator, it was imperative to identify a select number that would provide the most information on the effects of the liner cooling and slot cooling. Five different test cases were chosen for the endwall adiabatic effectiveness measurements as indicated in Table 1. Note that the percentage flows are given for the bottom set of liner panels only. The top set of liner panels were set at the same conditions. Also note that the momentum flux ratios for each of the panels are given. Given that the density ratios are nominally 0.97, the mass flux ratios can be calculated by taking the square root of the momentum flux ratios. These five cases were identified based on the total pressure profiles previously described. For the first three cases, the panel flows alone were varied to include a relatively uniform total pressure field, a nominally peaked profile, and a very peaked profile. The total pressure fields above the slot are shown in Figs. 4(a, b, and c) for these three cases. For the last two cases, the film-cooling flow through the panels remained the same.
Figures 4(a, b, and c) show the effect of increased liner flow on the total pressure exiting the combustor. There is a large effect up to \( Z/S = 0.15 \). There is also a noticeable difference in the mainstream that can be related to the influence of the dilution flow. In order to keep the same mass flow rate exiting the combustor while increasing the panel flow, the dilution jets are slightly decreased in order to keep the same mass flow rate exiting the combustor while stream that can be related to the influence of the dilution flow. In to the total pressure exiting the combustor. There is a large effect up non-uniform due to the interaction of the flow from the feed holes and the pin fins. To determine the total pressure in the slot downstream of the pin fins, profiles were measured across the height and pitch directions of the slot and spatially averaged with respect to the pitch. Pitchwise-averaged profiles measured at the combustor exit below and above the slot are shown in Fig. 5 for all five cases. Figure 5 shows the flat (case 1), nominally peaked (case 2), and highly peaked (case 3) profiles above the slot while the total pressure inside the slot remained the same for each of the these three cases. For the variation in slot flows, the data for the low slot flow (case 4) indicates lower total pressures in the slot than for the case for the high slot flow (case 5).

**Turbine Inlet Flow Field Conditions**

To insure that periodic flow fields occurred for both turbine vane passages, two-component laser Doppler velocimeter measurements were made across the two vane passages (inside and outside passages in Fig. 2(a)) at an axial location of the vane stagnation and at a spanwise location at the midspan. Both mean and turbulent velocity fluctuations were analyzed. Figure 6(a) shows the mean velocity measurements made for the inside and outside passages with and without the simulator [12] as compared with a two-dimensional, inviscid computation assuming periodicity. Note that \( Y/P = 0 \) refers to the stagnation location and as \( Y/P \) decreases to \(-1\) one is moving toward the pressure surface of the vane. The computation was made using FLUENT (1998) [12] using periodic boundary conditions. The measurements indicate that equal flow rates exist between the inside and outside passages for the case with the combustor simulator.

Figure 6(b) compares the turbulence levels exiting the combustor simulator as compared with those measured using an active grid turbulence generator [12]. The turbulence values were nominally the same for the inside and outside passages with the combustor simulator. In comparing these measurements to those for the active grid case, similar profiles were obtained with only
slightly higher values near the pressure side. These results indicate the effect of the dilution jets in generating high levels of turbulence at the entrance to the vane. The pitchwise average turbulence level at this location was 25% based on the inlet, incident velocity of 6.3 m/s.

Pressure distributions at the vane midspan were also recorded for all of the five cases and are shown in Fig. 7 as compared with the two-dimensional, inviscid computation. As seen by the figure, the pressure distribution remained nearly the same for all of the five cases that were tested.

Endwall Adiabatic Effectiveness Measurements

The endwall adiabatic effectiveness levels for the five cases that were previously discussed are shown in Figs. 8–12. The straight vertical lines on these plots indicate the downstream edge of the backward-facing step. Upstream of this line, the adiabatic wall temperatures were measured on top of the step for the combustor liner. Figure 8 shows the endwall effectiveness measurements for case 1, which was the design flow for the slot and film-cooling panels. Figure 8 indicates that a large portion of the slot flow is exiting near the suction side of the vane, thereby providing most of the cooling in this region. As the flow progresses through the passage, the coolant continues to be swept towards the suction side of the vane by the cross passage flow. It is clear that the combined liner and slot coolant is not effective at the leading edge region. A warm ring (low effectiveness) is present around the vane particularly at the vane stagnation. The endwall region near the pressure side of the vane sees very little effect from the slot flow. Note that the effectiveness levels are between 0.2 and 0.3 in the downstream passage as a result of the film-cooling from the combustor liner.

In determining the effect of increased film-cooling from the liner, comparisons of Fig. 8 can be made with Figs. 9 and 10. As would be expected, there is a slight increase in adiabatic effectiveness on top of the step as the combustor liner film cooling increases. This effect is important because the top of the step is a critical area where there are often durability issues. It is important to keep in mind that not only is the amount of available coolant increasing from 10 to 14% for cases 1 to 3, the total pressure profile has been altered. While the overall effectiveness pattern remains quite similar between these cases, there are several apparent differences. As the peak total pressure increases in the near-wall region, the driving pressure between the flow above the slot...
and below the slot also increases (Fig. 5). As a result of this increase in driving pressure, a larger portion of the liner cooling flow on top of the slot is pulled toward the endwall at the vane stagnation region. In comparing Fig. 8 with Fig. 10, it can be seen that the region just upstream of the vane stagnation for case 3 with the higher panel flows has slightly cooler adiabatic wall temperatures. This can be attributed to two effects. First, there is a larger driving potential between the flow above the step and flow in the slot. Second, there is cooler fluid on top of the step with the higher panel flow (case 3) as compared with the design panel flow (case 1).

The effect of the larger driving potential is not only evident in the stagnation region of the vane, but is also evident for the slot flow in general. Even though the same slot flow is exiting for cases 1 and 3, the coverage of the coolant flow from the slot has been reduced for case 3. This reduction occurs due to the fact that there is ingestion of flow from above the step into the slot that mixes with the slot coolant prior to the coolant exiting the slot. Although the ingested flow is relatively cool, it is still warmer than the slot flow itself. The ingestion of the flow from above the step into the slot itself was predicted by a CFD simulation (FLUENT) done by Stitzel [15]. The results of this CFD simulation are
shown in Fig. 13 for case 1 flow conditions. The strong ingestion into the slot will be illustrated in Part 2 of this paper through flow and thermal field measurements.

The larger amount of film-cooling flow present for case 3 gives only slightly increased effectiveness levels on the downstream endwall surface as compared with case 2. These results suggest that there is an optimal amount of combustor liner film cooling that will benefit the downstream vane platform. One plausible reason for this limit in improvement is that by increasing the liner flow, the inlet total pressure profile is also changed. With increased liner coolant, there is an increase in the peak total pressure resulting in stronger secondary flows. A peak in total pressure will cause a flow split along the vane span above which flow will be transported up the vane and below which flow will be transported toward the endwall similar to that shown in Fig. 13. Even though more cooling is available, much of it will be transported up the vane span.

The effect of increasing the slot flow may be determined by comparing Figs. 11, 9, and 12 in order of increasing slot flow. Note that the effectiveness levels on top of the step were nominally the same for each case. Adiabatic effectiveness levels for the design and half slot flow cases Figs. 11 and 9 are similar near the vane leading edge and along the suction side. The two cases show similar wedge-shaped slot-cooled regions. However, near the pressure side and in the passage the effectiveness levels for the half slot flow are noticeably lower than the design slot flow case. As the amount of slot flow is doubled Fig. 12, the effectiveness levels near the leading edge are as much as 0.15 higher than the design slot flow case. The wedge shaped region that is cooled by the slot is clearly larger for case 5. The double slot flow case is also more effective near the stagnation region as well. At a distance of $x/C = 0.25$ down the passage, the effectiveness levels are nearly the same for both the double and design slot flow cases.

The measured effectiveness values were pitchwise-averaged at a number of axial locations along the vane. Figure 14 shows the pitchwise averages for each of the five cases. Effectiveness levels are initially high, due to the injection of coolant onto the endwall by the slot. The first location presented is on top of the step and is a result of the liner cooling. The highest value occurs for case 3 given that this condition is the highest liner cooling. Just downstream of the slot, the highest value occurs for the highest slot flow. The values drop off sharply near the leading edge and then decline uniformly throughout the passage for all cases. It is interesting to note that the case with the most combustor panel flow has the lowest pitchwise effectiveness level at the leading edge. This is because of the effect that the increased panel flow has on the slot flow by increasing endwall secondary flow.

Figure 15 shows the area-averaged effectiveness values ($\bar{\eta}$) for each of the flow cases that have been presented. Note that the percentage mass flow only includes the film and slot flows from one side of the combustor panels and does not include the dilution flow. The percentages of the mass flow shown in Fig. 15 are based on the total exit mass flow of the combustor simulator inlet flow to the vane section. The data on this figure shows the trade-offs between increased film-cooling flow relative to increased slot cooling flow. The area-averaged effectiveness slightly increases with increased film-cooling flow before leveling off to a value that is 0.45 for film-cooling flows of more than 12.5% of the total exit flow.

In contrast to increased film-cooling flow, Fig. 15 indicates that there is a continual increase over the small range investigated in endwall effectiveness levels associated with increasing the slot flow. These results suggest that it is more beneficial for endwall heat transfer to increase the slot flow rather than the combustor panel flow. One plausible reason for the larger increase in effectiveness for increased slot flow is because of the reduced ingestion into the slot. The near-slot region dictates the relative differences between the slot flow cases.
The experimental results that were presented in this paper indicate the importance of understanding the inlet total pressure profile upstream of the first vane. The difference in total pressure between the fluid above the backward-facing step and below the backward-facing step caused ingestion into the slot that ultimately reduces the coolant temperature as it exited the slot. As the slot flow was increased, the total pressure of the fluid in the slot also increased thereby decreasing the driving potential between the main gas path and slot fluid. This reduction in driving potential was indicated by the adiabatic effectiveness values measured on the downstream vane endwall. Increasing the slot flow over the range that was investigated increased the area-averaged effectiveness values on the downstream endwall. Alternatively, measured effectiveness values indicated that increased film-cooling flow from the liner did not result in a continual increase in area-averaged effectiveness values.

The nonuniformities that occur at the inner and outer diameter regions of the combustor exit in a gas turbine engine have a large impact on the durability of vane and endwall. Much of the combustor liner coolant is swept toward the vane suction-endwall juncture. Secondary flows that occur in the vane stagnation region as well as those that develop in the vane passage further deteriorate an effective usage of the coolant.

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Nomenclature

\[ A_h = \text{area of film-cooling hole} \]
\[ C = \text{true chord of stator vane} \]
\[ C_d = \text{discharge coefficient}, \quad C_d = m/A_h \sqrt{2P (p_o - p_w)} \]
\[ J = \text{momentum flux ratio}, \quad J = p_v U_p / p_o U_{in}^2 \]
\[ k_{cond} = \text{thermal conductivity} \]
\[ m = \text{mass flow rate} \]
\[ P = \text{pitch of stator vane} \]
\[ P_o, p = \text{total and static pressures} \]
\[ Re = \text{Reynolds defined as } Re = C U_{in} / \nu \]
\[ s = \text{surface distance along vane measured from flow stagnation} \]
\[ S = \text{span of stator vane} \]
\[ T_{ave} = \text{mass-averaged freestream temperature} \]
\[ \nu = \text{kinematic viscosity} \]
\[ \Delta P = \text{ nondimensional pressure}, \quad \Delta P = (P_o - P_{in,ms}) / (0.5 \rho U_{ave}^2) \]
\[ \hat{\eta} = \text{adiabatic effectiveness}, \quad \hat{\eta} = (T_{ave} - T_{wall}) / (T_{ave} - T_{cool}) \]

Subscripts

- dilution rows 1 and 2
- pitchwise average at given axial location
- area average over entire endwall
- adiabatic wall
- coolant conditions
- midspan conditions
- mass-averaged freestream conditions (primary flow)
- inlet = inlet incident velocity

References


