FLOW AND THERMAL PERFORMANCE OF A GAS TURBINE NOZZLE GUIDE VANE WITH A LEADING EDGE FILLET

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Abstract
Complex three-dimensional vortex flows develop at the junction of a gas turbine airfoil and its casing (endwall). These flows increase the transfer of heat from the combustion gases to the metal parts and contribute to reduced aerodynamic efficiency. Past studies have shown that the use of a large fillet at the airfoil-endwall junction can reduce or eliminate the endwall vortex flow pattern. To determine the effects of a fillet on turbine surface conditions, wall shear stress and heat transfer coefficients were measured on the endwall of a low-speed linear turbine vane cascade. A shear stress measurement technique was developed and implemented for this study. High-resolution measurements of magnitude and direction provided quantitative information about the endwall flow features with and without a large fillet at the airfoil-endwall junction. The fillet was shown to increase shear stress magnitude, but reduce turning of the flow at the endwall. Heat transfer coefficients were also measured with and without the fillet. Results indicated that the leading edge fillet changed the distribution of heat transfer on the vane endwall.

Introduction
A major factor in reduced aerodynamic efficiency and increased part temperatures in a gas turbine engine is a complex three-dimensional flow, known as secondary flow. This flow occurs where an axial turbine airfoil meets the inner or outer casing, known as the endwall (see Figure 1). In an axial gas turbine, secondary flows result in aerodynamic losses in a vane or blade stage, which can reduce the engine’s overall efficiency by up to 3%. Secondary flows also tend to convect hot mainstream gases onto the endwall, which result in higher local heat transfer coefficients and increased metal temperatures. A 25°C (50°F) increase in metal temperature can result in a reduction in part life by a factor of two.

Controlling or eliminating secondary flow could increase the aerodynamic efficiency of the engine and reduce the required cooling, or the same amount of cooling could increase part life expectancy. While several options to eliminate secondary flow have been successfully investigated, a fundamental understanding of the flow remains elusive. More information about the flow and its interaction with the turbine surfaces is necessary to advance current engine designs. This paper will discuss the influence of a large fillet at the junction of the endwall and the airfoil, on the surface heat transfer and wall shear stress of a modern nozzle guide vane.

Literature Review
Secondary flow is not unique to gas turbines and has been investigated for symmetric airfoils and cylinders in crossflow. However, a couple of features of a turbine cascade are unique. First, the airfoils are not symmetric and turn the flow through large angles. This results in cross-passage pressure gradients that are not present for symmetric airfoils or cylinders. Second, the flow is accelerated in the turbine passage by a streamwise favorable pressure gradient. These factors, as well as the three-dimensionality of secondary flow, make it difficult to predict secondary flow development and progression.

Langston et al. presented one of the first descriptions of endwall secondary flow in a gas turbine. Other researchers have presented secondary flow models with slight differences, but they all agree on the major components. The incoming boundary layer on the endwall has a constant static pressure profile in the spanwise (normal to wall, along span of airfoil) direction. However, the boundary layer has a non-uniform velocity profile because of the difference in velocity between fluid entrained near the wall and fluid in the mainstream, which corresponds to a non-uniform total pressure profile. As the boundary layer stagnates on the airfoil, the total pressure profile becomes a spanwise pressure gradient that drives the flow to the endwall. This turning creates a vortex that splits at the stagnation point and wraps into two legs around the Endwall.

Figure 1. Secondary flow model presented by Langston.
It is apparent from past studies that a modification to the airfoil-endwall junction has some positive effect in optimizing the aerodynamic and heat transfer environment of a gas turbine vane. Fillets have shown the most promise; thus, this study focused on understanding how that geometry influences the flow. To date, none of the studies have presented measurements of heat transfer on the fillet surface itself. This surface would be of interest to turbine designers, since it would require additional cooling. Furthermore, wall shear stress has not been experimentally measured for vanes with airfoil-endwall modifications. Wall shear stress may be a contributor to total pressure loss in a cascade. Measurements may provide further insight into the secondary flow reduction mechanisms of a fillet.

**Experimental Design**

A large closed-loop, low-speed wind tunnel was used to perform heat transfer and wall shear stress measurements on a scaled-up nozzle guide vane test section. The flow is driven by a 50-hp axial fan, and passes through a primary heat exchanger to modulate the overall flow temperature. It enters a splitter section, where flow can be diverted for vane cooling experiments (not used in this study). The primary flow passes through several screens and a contraction section, which can be used to simulate a gas turbine annular combustor. The flow enters the corner test section, which contains the vanes for this study, and then passes back into the fan.

The corner test section holds two full nozzle guide vanes and a third partial vane connected to a flexible wall. The sides of the test section contain bleeds to remove the sidewall boundary layers and ensure that the flow around the vanes is periodic. The vane design is a three-dimensional extrusion of the two-dimensional midspan airfoil geometry from the Pratt & Whitney PW6000 turbine engine. The vane is scaled up by a factor of nine for high measurement resolution, and is instrumented with static pressure taps at 40% of the span (measured from the bottom endwall) to determine the static pressure coefficient. A description of the

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<td>Flow exit angle</td>
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**Table 1 PW6000 nozzle guide vane geometry and flow parameters.**
turbine vane parameters is given in Table 1.

The bottom endwall surface of the corner test section was manufactured out of a 2.54 cm (1") thick sheet of Last-A-Foam FR-6706 low-density closed-cell polyurethane foam, which has a very low thermal conductivity (0.0287 W/m-K). This material was used to minimize conduction losses for the heat transfer studies. The endwall was instrumented with type-E thermocouples in various locations throughout the vane passages, for calibration of the infrared images.

The top endwall was made of acrylic, and had thirteen image ports spaced around the test section. The layout of the image ports allowed overlap of infrared images to create a full endwall temperature map. Each image port was sealed with a removable acrylic cover when an image was not being taken.

### Heat Transfer Measurements

Heat exchanger

Axial fan

Corner test section with vanes

Figure 2. Schematic of the low-speed recirculating wind tunnel used in this study.

Endwall heat transfer measurements were taken on a constant heat flux plate attached to the endwall and fillet surfaces. The heaters, manufactured by Electrofilm Manufacturing, consisted of a 37 µm copper layer on top of a 75 µm thick kapton layer, in which 25 µm inconel elements were embedded in a serpentine pattern. The endwall heater had four separate circuits, three of which could be turned off when a fillet geometry was installed over the top of them, so as to not introduce error in the determination of the fillet surface heat transfer. The heaters were attached to the foam surfaces using double-sided tape embedded in a very thin layer of silicone adhesive. E-type thermocouples embedded in the foam endwall and fillet surfaces were placed in contact with the bottom surfaces of the heaters with Omegabond thermal cement.

The heater top surfaces were painted with Krylon flat black paint, which has a nominal emissivity of 0.96, and enables good resolution of surface temperatures with the infrared camera. Small crosses were etched into the paint on the endwall heater, for image positioning and thermocouple location determination. Three E-type ribbon thermocouples were attached to the endwall heater top surface with Omegabond thermal cement. The ribbon thermocouples were used to check the calibration with the bottom-surface thermocouples.

An infrared camera (Flir P20) was used to capture surface temperatures on the heat flux plate. Infrared radiation emitted from the surface is converted to temperatures by the camera. Calibration of the imaged surface temperatures was achieved by matching acquired thermocouple measurements with the known location of the thermocouple in an image. A measurement bias between the bottom-mounted thermocouple and the infrared top-surface measurement was accounted for by a one-dimensional calculation of the thermal resistance of the heater. The bias of 0.5°C was slightly larger than the uncertainty of a type-E thermocouple (±0.2°C).

Infrared images were taken by sequentially removing the inserts in the imaging locations on the top endwall and placing the camera (in a fixture) over the imaging location. Based on an uncertainty analysis, it was determined that five images should be taken at each location and averaged, where each image is also an average of 16 frames taken by the camera. The viewing area of the camera on the heat flux surface was 16.6 cm by 22.3 cm (6.5" x 8.78"), which it digitized onto 240 by 320 pixels. Flir’s ThermaCam Researcher software was used to process the images and calibrate them. An in-house routine was developed to assemble the individual images into a single map of the entire endwall.

A particular goal of this study was to analyze the heat transfer on the fillet surface. However, imaging the fillet is complicated by its three-dimensional nature. Perspective distortion is introduced, in which regions of the fillet closer to the camera appear larger than regions farther away. To correct this, crosses were etched into

Figure 3. Infrared images of the heat flux surface provide surface temperatures. The images are calibrated and assembled into an endwall map.
black paint on a stainless steel foil grid. The fillet was wrapped with the foil grid and imaged with the infrared camera. The grid provided control points for spatial transformation of the fillet surface in the image. The camera location was fixed so that subsequent images would be at the same location and orientation. Error in subsequent camera positioning was less than 4 pixels (2.8 mm at the nominal focal distance of the endwall).

An in-house MATLAB code was developed to perform the spatial transformations. The transformations involved some distortion of the image outside of the control points, so only the portion of the fillet surface within the grid was retained from the transformed images. Three-dimensional spatial transformation enabled a realistic look at the full surface heat transfer. The capability also existed to perform two-dimensional transformations by projecting the fillet surface to the endwall.

The input heat to the endwall and fillet surfaces was calculated by measuring the voltage across a heater circuit, as well as the voltage across a precision resistor (1 \( \Omega \)) in series with the circuit, which gave the current. The total power was divided by the area of the circuit to obtain the input heat flux. This flux was corrected for conduction and radiation losses, which accounted for a maximum of 0.2% and 21% of the input power, respectively. No correction due to conduction to the vane itself was performed, since the vane was also constructed of low-density closed-cell foam. The remaining convective flux was used with the measured surface temperatures to calculate heat transfer coefficients, which were normalized in the form of a Stanton number based on inlet mainstream velocity

\[
St = \frac{h}{\rho C_p U_\infty}
\]

where \( h \) is the heat transfer coefficient (relating convective heat flux to temperature gradients between the wall and the air), \( \rho \) is the density of air, \( C_p \) is the specific heat of air, and \( U_\infty \) is the freestream inlet velocity.

**Theory of Oil Film Interferometry**

A measurement technique for wall shear stress had not yet been developed in our laboratory. A method known as oil film interferometry was selected based on its robustness and simplicity. Holley and Langston presented oil film interferometry results for their low-speed scaled-up turbine blade cascade\(^{10}\). They were able to determine the unique features of endwall secondary flow (saddle point, separation line). Naughton and Sheplak\(^{11}\) present a very good description of the fundamentals and implementation considerations of the oil film interferometry technique.

The basis of oil film interferometry (OFI) is the behavior of a thin oil film under the influence of shear. Movement of the oil film can be dictated by shear on its surface, pressure gradients, gravity, and surface tension. In many cases, such as for this study, shear is the dominant force, by several orders of magnitude. The use of light ray interferometry to measure the height of the oil film as it is thinned by airflow can be used with a thin-oil film reduction of the Navier-Stokes equations for the oil, to obtain the shear acting on the oil.

Derivation of the thin oil film equation consists of a mass and momentum balance on a differential control volume through which oil conveets. A mass balance on the control volume gives the height of the oil and the average convective velocity as functions of time and space. The x- and z-momentum fluid equations can be simplified by performing an order of magnitude analysis to determine that the Reynolds number for the oil film is much less than one. The inertial terms and streamwise viscous terms can then be neglected, which simplifies the x- and z-momentum equations so that shear forces are balanced by pressure gradients and gravity. The momentum equations can be solved for the \( u \) and \( w \) oil velocities in terms of the pressure gradient, gravity, and airflow shear by applying no-slip boundary conditions at the wall-oil interface, and the

\[
U = \frac{\partial U}{\partial x}
\]

\[
U = \frac{1}{h} \int_0^h u dy
\]

\[
\frac{\partial U}{\partial x} = h + \frac{\partial h}{\partial x}
\]

Figure 4. An infrared image of the foil grid used to determine spatial transformation control points.

Figure 5. Oil film control volume with x-direction components shown.
desired shear at the air-oil interface. These velocities are related to the average convective velocities by integrating over the height of the oil. Combining the velocity distribution from momentum with the mass balance for the oil yields the thin oil-film equation

$$\frac{\partial h}{\partial t} + \frac{\partial}{\partial x} \left( \frac{\tau_w}{2\mu} + \frac{h^3}{3\mu} \left( \frac{\partial P}{\partial x} - \rho g_x \right) \right) + \frac{\partial}{\partial z} \left( \frac{\tau_w}{2\mu} + \frac{h^3}{3\mu} \left( \frac{\partial P}{\partial z} - \rho g_z \right) \right) = 0 \quad (2)$$

For representative values pertinent to this study ($h = 1$ µm, oil viscosity $\nu = 100$ cSt, shear stress $\tau_w = 10$ Pa, oil density $\rho = 1000$ kg/m$^3$, $dP/dx = 100$ Pa/m, $g_x = 10$ m/s$^2$), order-of-magnitude analysis on the terms in the parentheses in Eq. (2) shows that the shear stress term is at least two orders of magnitude larger than the other terms, which are then neglected. Knowledge of the height of the oil as a function of space and time leads to the wall shear that acted on the oil. For spatially constant viscosity and shear stress (reasonable assumptions in this study because of the large scale and small measurement sizes), the reduced oil film equation can be solved by separation of variables for the height of the oil. Such a solution requires measurement of the oil height only at the end of the wind tunnel run, since the conditions leading to the final oil film thickness are integrated over time.

Fizeau interferometry provides a means of measuring the height of the oil. Light strikes the surface of the oil film and is reflected and refracted. The refracted rays are then reflected by the solid boundary at the lower surface of the oil and pass back out of the oil. The phase difference between the initially reflected and refracted rays will attenuate or augment the rays, creating interference bands. The phase difference is related to the height of the oil by

$$h = \frac{\lambda \phi}{4\pi} \left( \frac{1}{\sqrt{n_{oil}^2 - n_{air}^2 \sin^2 \theta_i}} \right) \quad (3)$$

where $\lambda$ is the light wavelength, $\phi$ is the phase difference, $\theta_i$ is the incidence angle of the light, and $n_{oil}$ and $n_{air}$ are the indices of refraction of the oil and air, respectively. Examination of the interferograms related the phase difference to the physical spacing of the bands.

The wall friction coefficient reported in this study has been normalized by the upstream dynamic pressure and is generally defined as

$$C_f = \frac{\tau_w}{\frac{1}{2} \rho U^2_{\infty}} \quad (4)$$

Implementation of Oil Film Interferometry

The reader is again referred to Naughton and Sheplak$^{11}$ for an excellent description of implementation considerations for the oil film interferometry method. The major components required for oil film interferometry are a light source, a camera to capture images, a properly reflective surface, and, most importantly, oil.

It is desirable to have a single wavelength of light, to eliminate noise in an interferogram and allow for precise determination of the spacing of the patterns. A laser is an ideal source; however, other less-costly sources can work just as well. A low-pressure sodium vapor lamp, recognizable as a faded yellow street lamp, emits light at two very closely spaced wavelengths (589.0 nm, 589.6 nm). For this study, a low-pressure sodium vapor lamp from ABS Lighting was used to illuminate the oil films.

The interferograms are typically on the order of 1 mm (0.040") between bands, so high-resolution imaging is necessary. A digital camera (Pentax *iSt DS digital SLR) with a 75-300 mm zoom/macro lens was
attached to a fixture. The fixture allowed for measurement of the incident angle of light relative to the oil film, and consistency in the positioning of the camera.

A highly reflective surface provides good resolution of interference patterns without lowering the contrast. Glass, Mylar, and polished steel have all been found to be good substrates, but polished aluminum is not. Nickel and other foils can also work, especially since they can be moved to various regions of a large model without having to plate the entire model. For this study, 0.05 mm (0.002”) thick nickel foil with 14% tungsten (by weight) was cut into 7.6 cm by 3.81 cm (3” x 1.5”) rectangular patches, and adhered to the painted surface of the heater with a sprayed-on layer of rubber cement. To ensure that the leading edge of the foil did not interfere with the boundary layer, several tests were performed in which the oil film was in the same geometrical location, but the distance of the film from the leading edge of the foil was varied. In all cases, the variation of measured skin friction was less than 9%, which is approximately the uncertainty of the oil film method.

The choice of oil depends on the test conditions and the degree of uncertainty tolerable in the results. Typically, silicone oils (polydimethylsiloxane, commercially known as Dow Corning 200 Fluid) are used because of the relative insensitivity of their viscosity with temperature, their transparency, and the ease of removal from the model. Silicone oils can be tailored to a wide range of viscosities by lengthening the polymer chain. Although the viscosity is less sensitive to temperature than petroleum oils, it can still vary by 2% per °C, which makes determination of the oil viscosity one of the largest terms in the uncertainty of the shear stress measurement. It is recommended that the viscosity of each batch of oil be measured to account for variations from the nominal viscosity. Naughton and Sheplak[1] provide a correlation that relates a reference measurement of oil viscosity over a large range of temperatures. For this study, silicone oil in nominal viscosities of 100 cSt, 500 cSt, and 1000 cSt were used for shear stress measurements on the vane cascade endwall. Brian Holley performed reference measurements for each nominal viscosity, for the temperature correction.

The test procedure to obtain shear stress measurements consisted of several steps. First, a nickel foil patch was cleaned and sprayed with the thin layer of rubber cement. It was affixed to the endwall in the desired measurement location, which was recorded so that the measurements could be converted to vane coordinates. Next, the nickel surface was cleaned and lightly polished to remove any cleaning fluid film. A silicone dropper was dipped into the oil and small droplets (<3 mm diameter) were dabbed on to the surface of the nickel. The data acquisition for wind tunnel dynamic pressure and air temperature was started, and the wind tunnel was allowed to ramp up to the engine-representative inlet Reynolds number of 220,000. The test was continued for approximately twenty minutes, or until interference bands were detectable in the oil patches. The tunnel was shut down, and the patch was immediately removed. It was carefully placed in the fixture connected to the camera and images were taken of the interferograms illuminated by the low-pressure sodium vapor lamp.

Freeware imaging software (ImageJ, from National Institutes of Health) was used to obtain interference pattern spacing and direction. An in-house MATLAB code was developed to perform the numerical integration of the wind tunnel time-temperature history and to determine the wall friction coefficient.

Benchmarking of Oil Film Interferometry

The oil film interferometry method has been shown by several independent researchers to be a robust technique with minimal error (on the order of 5-10% uncertainty). It has been validated in several types of flows and has been found to agree well with theory or other measurement techniques. However, it was instructive for us to benchmark our own implementation, to provide an extra degree of certainty. For the benchmarking, wall shear was measured in a square channel, in the fully developed flow region. The channel was operated over a range of Reynolds numbers (based on hydraulic diameter), and the measured friction factor was compared to theoretical predictions. Figure 8 clearly demonstrates that the oil film method was valid over a range of flow speeds.

Fillet Design

An extended component of this study will involve experimental tests of various fillet design parameters to...
determine which have the largest effect on reducing endwall heat transfer and aerodynamic loss. To that end, a test matrix of possible fillet designs was developed. Based on the success of the large linear-profile fillet of Lethander et al., this fillet was chosen as a baseline for comparison with elliptical fillet designs. Elliptical designs might be simpler to integrally cast into a vane or blade shape without creating stress concentration points where the fillet meets the endwall or airfoil. The fillet of Lethander et al. is suited for parameter studies, since the focus of that work was an optimization of the fillet design for maximum thermal benefit.

Due to the limits of time and the complexities of measuring heat transfer and shear stress on the non-conformable surfaces of an elliptical fillet, results have only been obtained for the linear fillet. Future work will include testing for other fillet configurations to determine the dominant design parameters of a fillet that reduce endwall heat transfer and shear stress.

Two complete fillets were machined out of the same closed-cell foam as the endwall and instrumented with type-E thermocouples. One of the fillets was split to go around the two side vanes. Kapton heaters, with the same thickness and composition as the endwall heater, were attached to the fillet surfaces with double-sided tape. The fillets were secured firmly to the endwall and airfoil, and sealed around their perimeter with silicone.

Results

The results for measurements of heat transfer on the endwall, and on the fillet when installed, are presented first. Shear stress measurements with and without the fillet are shown for the endwall only.

Heat Transfer Results

To provide a baseline for comparison to the fillet results, measurements of endwall heat transfer for the unfilleted nozzle guide vane were taken. Kang et al. presented measurements of endwall heat transfer for the PW6000 vane geometry, which were compared to the results from this experiment to determine that the baseline case was displaying the correct trends. Figure 10 shows contours of Stanton number for Kang et al. compared to this study. High contour values denote high heat transfer coefficients; heat can more easily move into the endwall. Although there are some slight differences, particularly in the region directly upstream of the blades and along the flexible wall connected to the lower vane (x/C = 0.5, y/C = 0.25), these are most likely due to different flow conditions for this study. Specifically, the endwall heater plate had a shorter upstream length, and the inlet boundary layer had a different shape due to previous combustor simulator modifications. It can be seen from the baseline results that the horseshoe vortex near the stagnation region of the vane causes increased heat transfer upstream of the vane. Also, the passage vortex, sweeping from the pressure to suction sides of adjacent airfoils, causes higher heat transfer along the suction side of the vane.

Figure 11 shows a three-dimensional view of the measured heat transfer on an endwall with a linear fillet, compared to measurements on the unfilleted endwall. It is not immediately obvious that the fillet has reduced endwall heat transfer, especially on the suction side of the fillet. However, the shapes of the contours have changed, which indicates that the fillet is influencing the secondary flow and its interaction with the surface. Regions of low Stanton number are shifted more to the pressure side of the endwall with the linear fillet. Also, the large gradient in heat transfer at the stagnation region of the vane has been somewhat reduced. Overall, the surface areas of the endwall and the near-wall airfoil region have decreased with the addition of a fillet. Thus, even though the fillet does not dramatically reduce endwall heat transfer, it may...
reduce overall cooling requirements for the nozzle guide vane.

Wall Shear Stress Results

A vector plot of wall shear stress on the endwall of the unfileted vane is presented in Figure 12. The vectors displayed in the figure were uniformly sampled from approximately 400 data points taken over the entire endwall. Several features of the endwall shear stress due to secondary flow are apparent. Shear decreases as flow approaches the vane and stagnates. Flow washes down the airfoil and moves away from the base of the airfoil in the horseshoe vortex. The saddle point, where the flow diverges around the vane, is where the incoming boundary layer separates off of the endwall. Further in the passage, cross-passage flow sweeps from the pressure side of the lower airfoil to the suction side on the adjacent airfoil. The turning at the endwall due to secondary flow is larger than for the bulk flow away from the wall, which smoothly follows the airfoils.

Figure 13 shows endwall shear stress vectors overlaid on the results from the unfileted vane. In the stagnation region, it appears that the fillet is diverting

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flow area of the nozzle guide vane passage, causing flow to accelerate more rapidly. The static pressure coefficient around the vane, shown in Figure 14, supports this argument. The more interesting result from the linear fillet endwall shear is the turning of the endwall flow. The linear fillet begins to turn the passage flow sooner than in the unfilleted case, but the overall turning when the flow exits the vane passage is reduced. This may indicate a weaker passage vortex structure. Flowfield measurements should be taken to confirm this.

**Conclusions**

A wall shear stress method was developed and benchmarked for this study. High-resolution measurements on the scaled-up nozzle guide vane provided quantitative information about the endwall secondary flow features. When a fillet was added to the airfoil-endwall junction, the shear stress magnitude in the passage increased, but the overturning due to the passage vortex decreased, indicating possibly weaker secondary flow.

The addition of a fillet was shown to change the layout of the endwall surface heat transfer coefficients, shifting regions of low heat transfer to the pressure side of the airfoil. The reduced surface area of the endwall-airfoil junction with a fillet may mean reduced part cooling requirements, although this conclusion would require knowledge of the airfoil surface heat transfer with and without the fillet.

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**References**


