Effects of Surface Deposition, Hole Blockage, and Thermal Barrier Coating Spallation on Vane Endwall Film Cooling

With the increase in usage of gas turbines for power generation and given that natural gas resources continue to be depleted, it has become increasingly important to search for alternate fuels. One source of alternate fuels is coal derived synthetic fuels. Coal derived fuels, however, contain traces of ash and other contaminants that can deposit on vane and turbine surfaces affecting their heat transfer through reduced film cooling. The endwall of a first stage vane is one such region that can be susceptible to depositions from these contaminants. This study uses a large-scale turbine vane cascade in which the following effects on film cooling adiabatic effectiveness were investigated in the endwall region: the effect of near-hole deposition, the effect of partial film cooling hole blockage, and the effect of spallation of a thermal barrier coating. The results indicated that deposits near the hole exit can sometimes improve the cooling effectiveness at the leading edge, but with increased deposition heights the cooling deteriorates. Partial hole blockage studies revealed that the cooling effectiveness deteriorates with increases in the number of blocked holes. Spallation studies showed that for a spalled endwall surface downstream of the leading edge cooling row, cooling effectiveness worsened with an increase in blowing ratio. [DOI: 10.1115/1.2720485]

Introduction

In a typical gas turbine, components in the hot gas path operate under conditions of very high temperature, pressure, and velocity. These hostile conditions cause thermal oxidation and surface deterioration leading to reduced component life. Surface deteriorations are caused from deposits formed by various contaminants present in the combustion gases. With depleting natural gas resources, the focus now is on using alternate fuels like coal derived synthetic gas for gas turbine operation. These alternate fuels are not as clean as natural gas, resulting in deposition, erosion, and/or corrosion on turbine airfoils. Hence, it is important to understand how surface distortions can affect the performance of film cooling.

During engine operation, contaminants in the hot gas path are in a molten state resulting in surface deposition that leads to film cooling hole blockage and oxidation of the metal surface as a result of spallation. The dominant delivery mechanism of the contaminants onto the surface depends on the size or state of the contaminant [1]. Since the region around the film cooling holes are at a relatively lower temperature, deposits tend to adhere to surfaces near the holes. In some cases the deposits on the surface extend into the film cooling holes and end up partially or completely blocking the holes. Also, deposits can detach from the surface peeling the thermal barrier coating (TBC) resulting in spallation.

Higher power outputs from the turbine require higher turbine inlet temperatures and increasing gas temperatures in turn increase the rate of deposition. This study is aimed at understanding which of these effects is more detrimental to film cooling adiabatic effectiveness levels. With the knowledge of which of these effects are more detrimental, cooling designs can be made to combat those effects. The work presented in this paper compares three different surface distortions on film cooling effectiveness including: surface deposition, film cooling hole blockage, and TBC spallation.

Relevant Past Studies

A number of studies in the literature have documented the effect of using alternate fuels on turbine surfaces. There are also studies showing the effect of surface roughness on film cooling. Only a few studies exist on the effect of blocked holes on film cooling effectiveness. In addition, there have been few studies simulating the effect of TBC spallation on film cooling effectiveness.

Studies concerning the effect of using coal and other solid fuels have focused on ash and other contaminant levels on deposition, corrosion, or erosion. A study by Wenglarz et al. [2] showed that high levels of ash, up to 40 t/yr, can enter a turbine. DeCorso et al. [3] found that most coal derived fuels after conversion and purification had greater levels of impurity levels than those found in natural gas. They found that in natural gas sulfur levels are about 10 ppm, but a coal derived fuel after liquefaction and separation had about 1000 ppm of sulfur. Studies have also been carried out to see the effect of small amounts of ash flowing through a turbine. Moses and Bernstein [4] found that a fuel burning with 0.5% ash results in about 1.5 t/day of ash flowing through a turbine. A study by Wenglarz [5] yielded that the mass fraction of the ash particles adhering to surface and contributing to deposit buildup was 0.06 with an average particle diameter of 10 μm. Recent studies by Bons et al. [6] were carried out to study the effect of ash deposits present in coal fuels in an accelerated deposition test facility. They carried out their tests with ash particles at a slightly higher mean mass diameter of 13.3 μm than those found in coal fuels. For a test duration of 3 h, they found the net particle loading to be 165 ppmw/h (parts per million by weight per hour) resulting in an average deposit thickness of 1.3 mm.

Bonding of ash and other contaminants to turbine component surfaces is dependent on their surface properties [3]. Deposition of...
contaminants and other particulate matter present in the mainstream gas takes place through impingement onto the surface. Typically in a gas turbine, the first rows of vanes are subjected to direct impingement resulting in particulate deposition [7]. The mechanism for this could be attributed to the fact that the particles are broken up and redistributed by impact [8]. Also, the leading edge film cooling regions on a vane endwall are more prone to deposition. Because this is the first cold surface the hot mainstream gas encounters before it enters a turbine and as mentioned before the molten contaminants would quench and adhere near the relatively colder regions. Bons et al. [9] showed through their turbine surface measurements that the leading edge region will have the greatest degradation. Also, actual turbine component measurements by Bons et al. [9], Taylor [10], and Tarada and Suzuki [11] revealed that all the regions of a turbine blade and vane are prone to deposition and roughness with varying levels.

A number of studies have been reported showing the effect of surface roughness on film cooling. Goldstein et al. [12] placed cylindrical roughness elements at the upstream and downstream location of film cooling holes on a flat plate. They observed that at low blowing rates, there is a decrease in adiabatic wall effectiveness by 10–20% over a smooth surface. However, at higher blowing rates they observed an enhancement of 40–50% in cooling performance. Schmidt et al. [13] did a similar study using conical roughness elements. They also found a higher degradation of film cooling effectiveness at lower momentum flux ratios than at higher momentum flux ratios. Barlow and Kim [14] studied the effect of a staggered row of roughness elements on adiabatic effectiveness. They found that roughness degraded cooling effectiveness compared to a smooth surface and the smaller elements caused a greater reduction than the larger elements.

There have been very few studies on the effect of roughness on endwall film cooling. In the study by Cardwell et al. [15], they showed that a uniformly rough endwall surface decreased cooling effectiveness at higher blowing ratios but at lower blowing ratios there was no significant change.

Not many studies are available in the literature showing the effect of blockage on film cooling effectiveness. The study done by Bunker [16] showed that for a partially blocked row of cylindrical holes there was significant degradation of local adiabatic effectiveness near the hole exit regions. The partial hole blockage was simulated by applying air plasma sprayed TBC over the holes without any protection. Bogard et al. [17] showed a cross section of a film cooling hole for an actual turbine part which depicted partial blockage of the hole downstream with the deposition extending outside of the hole.

Ekkad and Han [18,19] carried out heat transfer and film cooling measurements on a cylindrical model simulating TBC spallation. Ekkad and Han [18] found that spallation enhances heat transfer up to two times compared to a smooth surface. They also found that local heat transfer distributions were greatly affected by spallation depth and location. Ekkad and Han [19] found that higher freestream turbulence coupled with spallation greatly reduced film effectiveness. They found that spallation reduced film effectiveness inside the spallation cavity but increased it downstream of the spallation cavity.

In summary it is important to understand the effects of surface distortion due to contaminants present in fuel and combustion air. Though there have been many studies addressing the issue of surface deposition and surface distortion as a whole, there have been very few studies showing the effect of these distortions on a film cooled endwall.

**Experimental Methodology**

The experimental section was placed in a closed loop wind tunnel facility, as shown in Fig. 1. The flow encounters an elbow downstream of the fan and passes through a primary heat exchanger used to cool the bulk flow. The flow is then divided into three channels including the center passage and two cooled secondary passages located above and below the test section. Note that only the top secondary passage was used for this study. The primary core flow, located in the center passage, convects through a heater bank where the air temperature is increased to about 60°C. The secondary flow, in the outer passage, was cooled to about 20°C, thereby maintaining a temperature difference of 40°C between the primary and secondary flows. The secondary flow provided the coolant through the film cooling holes and through the leakage paths. Also, for all the tests carried out in this study a density ratio of 1.1 was maintained between the coolant and mainstream flows. Due to the fact that density ratios are not being matched to that of the engine, velocity ratios for the cooling holes will be significantly higher than those found in an engine for the same mass flux or momentum flux ratios. While there have been studies indicating that momentum flux scales jet lift-off for flat plate film cooling, it is unknown as to whether it best scales lift-off for endwall film cooling.

Downstream of the flow/thermal conditioning section is the test section that consists of two full passages with one center vane and two half vanes. Table 1 provides a description of turbine vane geometry and operating conditions. The vane geometry used in the current study is a commercial first-stage vane previously described by Colban et al. [20,21]. The passage under study consisted of an endwall surface with film cooling holes and leakage paths simulating the combustor-vane interface and vane-to-vane interface.

A detailed description of the endwall construction has been previously described by Knost and Thole [22] and Cardwell et al. [15], who used exactly the same film cooling and slot geometries as we used for this study. The endwall of the vane was constructed of foam because of its low thermal conductivity (0.033 W/m K). The endwall foam was 1.9 cm thick and was mounted on a 1.2 cm-thick Lexan plate. The cooling hole pattern on the endwall was cut with a five-axis water jet to ensure precision and integrity. The upstream slot was constructed with poplar wood as it had a low conductivity and was stiffer. The endwall surface was also covered with a 36 grit sandpaper to simulate a uniform surface roughness. Adiabatic endwall temperatures were measured for different flow rates through film cooling holes, through the

**Table 1 Geometric and flow conditions**

<table>
<thead>
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<th>Scaling factor</th>
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<td>Pitch/chord (P/C)</td>
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<tr>
<td>Span/chord</td>
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<tr>
<td>Hole L/D</td>
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<td>Re_o</td>
<td>2.1 x 10^4</td>
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<tr>
<td>Inlet and exit angles (deg)</td>
<td>0 and 72</td>
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<tr>
<td>Inlet Mach number</td>
<td>0.017, 0.085</td>
</tr>
<tr>
<td>Inlet mainstream velocity</td>
<td>6.3 m/s</td>
</tr>
<tr>
<td>Upstream slot width</td>
<td>0.024C</td>
</tr>
<tr>
<td>Midpassage gap width</td>
<td>0.01C</td>
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coupled with a constant flow rate through the vane-to-vane interface gap (referred to as midpassage gap). The inlet turbulence intensity and length scales were measured to be 1.3% and 4 cm, respectively. For every test condition the dimensionless pressure coefficient distribution was verified to ensure periodic flow through the passages. Three separate plenums were used to control the flow rate through the film cooling holes, through the upstream slot, and through the midpassage gap. A global blowing ratio was calculated using an inviscid blowing ratio along with a global discharge coefficient, \( C_D \), that was obtained from computational fluid dynamics (CFD) studies (as reported by Knost and Thole [23]).

The upstream slot flow was assumed to have a discharge coefficient of 0.6 which is the assumed value for a flow through a sharp-edged orifice and the flow rate was calculated accordingly. Flow through the midpassage gap was measured directly using a laminar flow element (LFE). In this study coolant flow rates are reported in terms of percent coolant mass flow rate per total passage mass flow rate for one vane pitch. Typical times to achieve steady-state conditions were 3 h.

**Instrumentation and Temperature Measurements.** A FLIR P20 infrared camera was used to spatially resolve adiabatic wall temperatures on the endwall. Measurements were taken at six different viewing locations to ensure that the entire endwall surface was mapped. The camera was placed perpendicular to the endwall surface at a distance 55 cm from the endwall. Each picture covered an area 24 cm by 18 cm with the area being divided into 320 by 240 pixel locations. The spatial integration of the camera was 0.715 mm (0.16 hole diameters). Thermocouples were also placed on the endwall surface at different locations to directly measure the temperature to post-calibrate the infrared images. For the post-calibration the emissivity was set at a constant value of 0.92 and the background temperature (\( \sim 45^\circ \text{C} \)) was adjusted until the temperatures from the infrared camera images were within 0.05°C of the corresponding thermocouple data. Six images were taken at each of the viewing locations to obtain an averaged picture using an in-house Matlab program. The same program was also used to assemble the averaged pictures at all locations to give a complete temperature distribution along the passage endwall.

F freestream temperatures were measured at multiple locations along the pitch and the average was determined by using a thermocouple rake consisting of three thermocouples along the span. It was found that the variations along the pitch were less than 0.2°C and that along the span were less than 1.5°C. Voltage outputs from the thermocouples were acquired by a 32 channel data acquisition module that was used with a 12-bit digitizing card. The temperature data were compiled after the system reached steady state.

A one-dimensional (1D) conduction correction as described by Ethridge et al. [24] was applied to all adiabatic effectiveness measurements. This correction involved measuring the endwall surface effectiveness with no blowing through the film cooling holes. This was done by blocking the film cooling holes on the endwall passage under study while maintaining similar flow rates through the adjacent passage to insure the correct boundary condition under the endwall. The resulting \( \eta \) correction was found to be 0.16 at the entrance for a \( \eta \) value of 0.9 and 0.02 at the exit region at a measured \( \eta \) value of 0.5.

**Experimental Uncertainty.** An uncertainty analysis was performed on the measurements of adiabatic effectiveness using the partial derivative method described at length by Moffat [25]. The precision uncertainty was determined by taking the standard deviation of six measurement sets of infrared (IR) camera images with each set consisting of six images. The precision uncertainty of the measurements was \( \pm 0.014^\circ \text{C} \). The bias uncertainty was \( \pm 1.0^\circ \text{C} \) based on the uncertainty in the IR camera measurements specified by the manufacturer. The bias uncertainty of the thermocouples was \( \pm 0.5^\circ \text{C} \). The total uncertainty was then calculated as \( \pm 1.02^\circ \text{C} \) for the images and \( \pm 0.51^\circ \text{C} \) for the thermocouples. Uncertainty in effectiveness, \( \eta \), was found based on the partial derivative of \( \eta \) with respect to each temperature in the definition and the total uncertainty in the measurements. Uncertainties in adiabatic effectiveness were calculated to be \( \pm 0.0303 \) at a \( \eta \) value of 0.2 and \( \pm 0.0307 \) at a \( \eta \) value of 0.9.

**Simulations of Surface Distortions**

Direct surface measurements on turbine engine hardware reported by Bons et al. [9], Taylor [10], and Tarada and Suzuki [11] all indicated an increase in surface roughness with an increase in turbine operational hours. Measurements by Taylor [10] were made on a blade with rms roughness heights varying from 30 \( \mu \text{m} \) to 50 \( \mu \text{m} \). His measurements also revealed that the maximum peak-to-valley height of 79 \( \mu \text{m} \) at the leading edge was higher than at other regions of the blade. Tarada and Suzuki [11] did measurements on industrial and aero-engine blades. They also found that the specimens measured a higher distortion at the leading edge areas with a roughness center line average as high as 85 \( \mu \text{m} \).

Most surface roughness measurements were done on engine blades with very little data available in the literature on measurements done on a vane and no data available for hardware used with synfuels. Bons et al. [9] performed measurements on vanes and blades of an actual land based gas turbine engine. Their findings also reported that on an average the roughness on the leading edge is greater than at the trailing edge. Their maximum roughness height \( (R_k) \) at different locations varied from 20 \( \mu \text{m} \) to as high as 440 \( \mu \text{m} \).

For this simulation, the measurements completed by Bons et al. [9] were referenced and scaled up to match realistic vane endwall distortions. All the measurements done by Bons et al. [9] were represented in terms of center-line average roughness \( (R_k) \), rms roughness \( (R_m) \), and maximum peak to valley distance \( (R_d) \). Studies by Bogard et al. [17] showed that it is appropriate to represent uniform surface roughness in terms of equivalent sand grain roughness \( (k_r) \) instead of \( R_k \). As this study primarily focuses on the effects of localized distortions on endwall film cooling effectiveness the authors thought that it be more appropriate to represent the deposit heights in terms of \( R_k \). Bons et al. [9] measured an \( R_k \) value of 258 \( \mu \text{m} \) on the endwall surface and the simulated deposit heights on our endwall corresponded to 253 \( \mu \text{m} \), 406 \( \mu \text{m} \), and 609 \( \mu \text{m} \) at engine scale.

Even though deposits on an actual vane endwall do not have a characteristic shape and size, a methodical approach was taken to simulate cooling hole distortions by molding specific deposits with definable shapes and sizes. The endwall deposits (shown in Fig. 2(a)) were all semi-elliptical with varying minor radii (in the wall-normal direction) of 0.5D, 0.8D and 1.2D. As mentioned before, the heights at engine scale corresponded to 253 \( \mu \text{m} \), 406 \( \mu \text{m} \), and 609 \( \mu \text{m} \), respectively. The streamwise length of all deposits was 2D.

The deposits were manufactured by injecting epoxy gel into a mold made of aluminum. The epoxy was allowed to dry in the mold for about 10 min before it was peeled out of the mold and left to dry until it was rigid enough to be placed onto the endwall. The dried epoxy was sprayed with adhesive to provide a sticky surface whereby sand was sprinkled on top of the deposit to make the surface rough. As mentioned previously the endwall surface was covered with a 36 grit sand paper [15] with an equivalent sand grain roughness \( (k_r) \) value of 0.42 mm at engine scale. Finally, the deposit was painted black and secured onto the sandpaper using a double sided tape.

The cooling hole blockages were manufactured similar to the deposits conforming to the required dimensions. Figure 2(b) shows the cross-sectional view of a blocked film cooling hole. The holes were blocked in such a manner as to reduce the exit hole...
area by 25%. The deposit extended one hole diameter into the hole and extended two hole diameters downstream of the cooling hole exit.

The sandpaper on the endwall simulating the uniform roughness corresponded to a thickness of 0.43D or 220 μm at engine scale. In most land-based gas turbine vanes and blades the TBC thickness is on the order of about 300 μm. As such, to simulate a spalled endwall the sand paper was removed from the surface to the specified width as shown in Fig. 2.

Because the mainstream gas flows from the pressure side to the suction side, heavy particles in the flow deposit along the pressure side surface and in the stagnation region. Considering these findings, the film cooling row at the leading edge region of the vane endwall was chosen to carry out deposition, blockage, and spallation studies, as shown in Fig. 3. In addition, deposition was also studied along the pressure side cooling holes and spallation was studied along the length of the midpassage gap.

Discussion of Results

Two baseline cases were measured whereby these included one with upstream slot coolant and one without upstream slot coolant. For the deposition and spallation studies, the upstream slot leakage flow was present. Blockage studies were done for the case without any upstream slot coolant flow. Figures 3(a)–3(e) show the surface distortions, the complete endwall contour, and the two baseline cases for this study. Figures 3(c)–3(e) compare lateral average effectiveness and contours of the two baseline cases only at the leading edge region. Adiabatic effectiveness levels were laterally averaged across the pitch (boxed region) from upstream of the cooling holes at a location X/C=−0.05, to a downstream location at X/C=0.05. It is important to note that for the baseline cases the endwall surface did not have localized surface distortions but did have a uniform surface roughness covering the entire endwall surface [15]. The coolant mass flow rate for the baseline cases was set at 0.5% of the core flow through the film cooling holes and 0.2% through the midpassage gap. For the baseline case with the upstream slot flow, 0.75% of the core flow passed through the upstream slot.

There are some clear differences between the cases with and without upstream slot flow in terms of jet penetration towards the leading edge–endwall juncture as shown in Figs. 3(d) and 3(e). In the case where the slot flow is present, the cooling jets convect closer to the stagnation region. This may be due to the fact that just upstream of the holes the endwall boundary layer is most likely ingested into the upstream slot giving only a thin boundary layer thickness approaching the holes. The thinner boundary layer minimizes any horseshoe vortex formation which causes the coolant to be swept away from the endwall. It can be seen in Fig. 3(e)
that without the upstream slot flow, the horseshoe vortex effect is more dominant, resulting in the formation of a hot band around the stagnation region.

**Deposition Studies.** Figures 4(a)–4(c) show the local adiabatic effectiveness for different deposit heights at the leading edge region. For these cases, the coolant mass flow rates were 0.75% ($M_{in}=0.3$) through the upstream slot, 0.5% ($M_{in}=1.5$) through the film cooling holes, and 0.2% ($M_{in}=0.1$) through the midpassage gap.

Figure 3(d) shows that for the baseline case without any deposition downstream, the coolant from the leading edge holes was swept around the endwall junction from the stagnation region to the suction side. By placing a deposit that is 0.5D in height (refer to Fig. 4(a)) downstream of the cooling row, the coolant levels were much improved. The reason for this improved cooling is attributed to the fact that the coolant impacts the raised deposit as the jet is slightly separated from the wall for the baseline case. Moreover, it is speculated that a Coanda effect further contributes to the improvement, causing the coolant to move toward the endwall surface. With the increase in deposit height to 0.8D (refer to Fig. 4(b)), the coolant stagnates and is then deflected from the surface, thereby reducing the cooling effectiveness relative to that of the 0.5D case. In comparing Figs. 4(b) and 4(c) it can be seen that an increase in deposit height to 1.2D brought about the same effect for both deposit heights of 0.8D and 1.2D.

The above effects can be further quantified by looking at the lateral average plots. Figure 5 shows the change in laterally averaged adiabatic effectiveness between the deposit cases, relative to the baseline case. Note that the values greater than one are enhancements in effectiveness levels and less than one are reductions in effectiveness levels. Effectiveness was laterally averaged across the pitch downstream of the leading edge film cooling row as shown in Figs. 3(b) and 3(c). It can be seen that at a deposit height of 0.5D there is an overall enhancement of cooling effectiveness. The initial peak near the exit of the cooling holes is due to the obstruction and collection of the coolant flow caused by the deposit. Though there is decay in the enhancement of cooling effectiveness as the flow proceeds toward the vane–endwall junction it can be clearly seen that a deposit of height 0.5D enhances the overall cooling effectiveness. Figure 5 also shows that for deposits of height 0.8D and 1.2D, there is an improvement in effectiveness in the near hole region, but due to jet liftoff there is a sudden decrease with reduced cooling relative to the baseline case as the coolant flows toward the vane–endwall junction.

Deposition studies were also carried out along the pressure side cooling rows. For these tests a constant deposit height of 0.8D was used. Figures 6(a)–6(c) compare the adiabatic effectiveness along the pressure side cooling holes with deposits placed at the downstream, upstream, and on both sides of the cooling rows (indicated in the contours with lines). Note that the upstream deposits were similar to downstream deposits (Fig. 2(a)) except that the deposits were placed upstream of the cooling hole row. Both sides refer to placing deposits on both downstream and upstream of the cooling hole rows. Downstream deposits shown in Fig. 6(a) deflected the cooling jets towards the vane–endwall junction.
which is the direction that the jets are directed. As the coolant is directed more toward the juncture, there is an overall reduction in the cooling downstream of each row (compare with Fig. 3(b)). Upstream deposits (Fig. 6(b)) tended to cause the jets to have the same trajectory, but also cause a larger reduction in film cooling effectiveness relative to the downstream deposits. Similar to these results, placing deposits on both sides of the cooling rows (Fig. 6(c)) resulted in the bulk of the coolant getting even more streamlined toward the vane–endwall junction. From Fig. 3(b) it can be seen that for the baseline case, the coolant from the pressure side flowed toward the suction side increasing effectiveness levels downstream of the cooling rows. By placing deposits in this region the coolant was redirected toward the vane–endwall junction causing a reduction in the effectiveness levels downstream of the cooling rows. Though deposits at every location deflected the coolant, upstream deposition was found to cause the most severe reduction in adiabatic effectiveness.

The above effects can be quantified by comparing the laterally averaged adiabatic effectiveness. In this region effectiveness was laterally averaged streamwise as shown in Fig. 7. Shown in Fig. 7 are the laterally averaged local blowing ratios for each row of holes along the pressure side. Note that CFD results were used to quantify the local coolant flows from each cooling hole and the local static pressure was used to calculate the local freestream velocity that was used in the blowing ratio definition. It was seen that deposits upstream of the holes and on both sides of the cooling holes caused similar and higher degradation than deposits placed downstream of the cooling rows. Note that the deposits were placed on the first four cooling hole rows, but the effects are present on all the rows on the pressure side. It can be concluded that upstream deposition resulted in higher degradation than deposits at the downstream of the film cooling holes. This conclusion can be generated because the deposits on both upstream and downstream have nearly the same degradation as the upstream deposits alone.

As the coolant jets were directed toward the pressure–surface endwall junction, it was important to quantify the change in adiabatic effectiveness along flow streamlines passing through the cooling holes and also along the pressure–surface endwall junction. Figures 8(a) and 8(b) show the variation of adiabatic effectiveness along the flow streamlines at different film cooling flow rates with deposits placed on both sides of the cooling rows. Variation in adiabatic effectiveness was measured along two streamlines; S1 and S2 as shown in Fig. 3(b). The flow streamlines were computationally predicted at 2% span from the endwall surface using FLUENT 6.1.2. Figure 8(a) shows that by placing deposits on both upstream and downstream locations the adiabatic effectiveness levels decreased by about 50% along S1 for 0.5% film cooling mass flow rate. A similar effect was seen for 0.75% (M_{in}=2.2) and 0.9% (M_{in}=2.7) flow rate through the film cooling holes though there was a higher reduction at a mass flow rate of 0.9%.

As mentioned earlier, in the presence of deposits on both sides of the cooling rows the coolant was streamlined toward the vane–endwall junction. This was further quantified by measuring the adiabatic effectiveness along streamlines S1 and S2 (refer to Fig. 3(b)) for deposits on both sides of the cooling rows.

Hole Blockage Studies. Partial blockages of film cooling holes were simulated at the leading edge film cooling row as mentioned previously. Tests were carried out by sequentially blocking each one of the five film cooling holes in the stagnation region though results are presented only for one hole, two hole, and five hole blockages. Similar to the other cases, the coolant mass flow rate for the baseline case for this study was set at 0.5% film cooling
holes, 0.2% midpassage gap, and no coolant injected through the upstream slot. There was no flow through the upstream slot as it was important to isolate any effect of the upstream flow at the leading edge cooling near the stagnation region.

Figures 9(a)–9(c) compare the adiabatic effectiveness contours with different number of holes partially blocked. The blockage extended one cooling hole diameter into the film cooling holes with a net flow area reduction of 25% at the cooling hole exit. Note that the same pressure ratio was used for the baseline and for the cases with blocked holes. Indicated with boxes in Figs. 9(a)–9(c) are the holes that are blocked. It can be seen that there is a dramatic decrease in effectiveness level with the increase in the number of holes being blocked as the coolant from the holes ceases to flow onto the endwall. The main reason for this is that when a hole is partially blocked, the momentum of the jet and the trajectory of the jet is increased, which causes the coolant to exit at a higher exit angle promoting separation. In this particular test the coolant from the partially blocked hole tended to flow directly into the mainstream resulting in lower adiabatic effectiveness on the endwall surfaces.

Figure 10 compares the degradation in laterally averaged effectiveness due to film cooling hole blockage. It can be seen that with the increase in the number of holes being partially blocked from 1 to 5 the laterally averaged effectiveness degrades from about 20% to about 50% of the baseline, respectively. It can be also seen that at upstream of the leading edge cooling holes (at X/C=−0.05) the effectiveness levels are higher due to the change in coolant jet trajectory caused by hole blockages. The study done by Bunker [16] showed a similar reduction of about 40% in the centerline adiabatic effectiveness for a row of blocked film cooling holes on a flat plate.

Spallation Studies. TBC spallation was simulated downstream of the leading edge film cooling row and also along the midpassage gap. The coolant mass flow rates for the baseline case for this study were set at 0.75% upstream slot, 0.5% film cooling holes, and 0.2% midpassage gap. Figures 11(a) and 11(b) compare the contours of adiabatic effectiveness of a spalled leading edge surface at film cooling mass flow rates of 0.5% and 0.9%.

The spalled endwall surface downstream of the cooling holes caused the coolant to flow over a forward facing step. This resulted in flow recirculation and prevented the coolant from sweeping across the vane–endwall junction. It was seen that at a mass flow rate of 0.5% although the near hole region is sufficiently cooled, effectiveness levels along the vane–endwall junction were relatively lower than the baseline case. By increasing the coolant mass flow rate to 0.9% the effectiveness levels were further reduced due to the combined effect of spallation and coolant jet liftoff of the surface.

Figure 12 compares the degradation in laterally averaged effectiveness due to spallation at the leading edge region. It can be seen
that at a mass flow rate of 0.5%, the near hole regions have similar effectiveness levels as the baseline, but effectiveness levels decrease by about 20% near the vane–endwall region. At higher mass flow rate of 0.9%, due to jet liftoff, there is a reduction of about 50% downstream of cooling holes. Effectiveness levels increase near the junction due to jet reattachment, though these levels are still lower than the levels occurring in the baseline case. Note that the cooling holes lie between \(-0.05 < X/D < 0.034\) and the spalled surface extends from \(X/C = -0.034\) to \(X/C = -0.02\).

Figures 13(a) and 13(b) compare the adiabatic effectiveness levels of the baseline and an endwall surface spalled five gap widths (5G) along the midpassage gap. The study by Cardwell et al. [15], showed that due to the presence of the midpassage gap, coolant from the upstream slot flows toward the suction side negating the need for endwall cooling in that region. Even though the surface was spalled along the midpassage gap there seemed to be no effect on the overall endwall effectiveness resulting from the large coolant flow over this region. It can be seen from Figs. 13(a) and 13(b) that the coolant from the upstream slot has the same coverage area and there is no significant reduction in adiabatic effectiveness levels. For the spallation studies along midpassage gap, tests were also carried out for smaller spalled regions. These tests also showed that spallation along midpassage gap had little effect on endwall effectiveness.

### Summary of Cooling Reductions

The two factors that are important to quantify the effects of surface distortions are the film cooling effectiveness levels and heat transfer coefficients. This study focused primarily on film cooling effectiveness alterations due to surface distortion factors. It is important to understand which surface distortion brought about the maximum reduction in cooling effectiveness. This was done by calculating the percent reduction in area-averaged effectiveness. Figure 14 shows the combined effectiveness reduction for each endwall distortion study. This plot allows one to assess which is the most degrading effect.

Deposition studies had varied results. Deposits improved effectiveness at the leading edge, whereas along the pressure side deposits resulted in reducing the film cooling effectiveness. A very interesting result was observed that deposits directly downstream of the leading edge cooling row enhanced film cooling effectiveness at lower deposit heights (0.5D and 0.8D) and caused little reduction in effectiveness levels at a deposit of height 1.2D. Along the pressure side it was seen that depositions typically degraded effectiveness levels downstream of the cooling rows by redirecting the coolant toward the vane–endwall junction. The reason for the degradation along the pressure side is because the jets are more attached to the surface and are easily deflected by deposits.

### Conclusions

Measurements of adiabatic effectiveness were presented for an endwall surface with simulated surface distortions namely, surface deposition, film cooling hole blockage, and TBC spallation. When the effects of these distortions were compared at the leading edge region, it was found that partial hole blockage and TBC spallation caused a higher reduction in adiabatic effectiveness levels than surface deposition.

Near hole depositions were studied with varying deposit heights and it revealed very interesting results. For a smaller deposit height of 0.5D the overall film cooling effectiveness downstream of the film cooling row was enhanced by about 25%. These deposits tended to deflect the coolant toward the endwall surface. With an increase in deposit height to 1.2D the coolant jet tends to lift off, lowering the film cooling effectiveness. Deposition along the pressure side lowered the effectiveness between successive cooling rows, but enhanced the effectiveness along the vane–endwall junction.

Hole blockage studies showed that partially blocked holes have the greatest detrimental effect on degrading film cooling effectiveness downstream of a film cooling row. At low blowing ratios, partially blocking five film cooling holes reduced film cooling effectiveness by about 30% and blocking a single hole resulted in a reduction in effectiveness of about 10%. Spallation studies conducted at the leading edge film cooling row showed a reduction in effectiveness of about 10%. Past studies have shown that spallation near film cooling holes is a significant form of surface distortion and hence it is very important to consider its effect while designing endwall cooling arrangements. Spallation was also studied along the midpassage gap which showed very little to no effect in endwall cooling effectiveness.

This study has shown the effects that cooling hole blockages can have on reducing endwall effectiveness levels. These effects
should be considered in determining improved film cooling designs so that partial hole blockages can be avoided on turbine components.

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Nomenclature

\[ C = \text{true chord of stator vane} \]
\[ D = \text{diameter of film cooling hole} \]
\[ G = \text{mid passage gap width} \]
\[ k_i = \text{equivalent sand grain roughness} \]
\[ L = \text{length of mid passage gap} \]
\[ LE = \text{leading edge} \]
\[ M = \text{local mass flux/blowing ratio, } M = \rho U_i/\rho_u U_\infty \]
\[ M_{in} = \text{blowing ratio based on inlet mainstream velocity} \]
\[ MP = \text{midpassage} \]
\[ P = \text{vane pitch; hole pitch} \]
\[ PS = \text{pressure side} \]
\[ Re_p = \text{Reynolds number defined as } Re_{in} = CU_{in}/\nu \]
\[ R_c = \text{center-line average roughness} \]
\[ R_s = \text{rms roughness} \]
\[ R_p = \text{maximum peak to valley distance} \]
\[ s = \text{distance measured along a streamline} \]
\[ S = \text{span of stator vane} \]
\[ T = \text{temperature} \]
\[ X,Y,Z = \text{local coordinates} \]
\[ U = \text{velocity} \]

Greek

\[ \eta = \text{adiabatic effectiveness, } \eta = (T_\infty - T_m)/(T_\infty - T_c) \]
\[ \bar{\eta} = \text{laterally averaged effectiveness} \]
\[ \bar{\eta}_a = \text{area-averaged effectiveness} \]
\[ \Delta \bar{\eta}_R = \text{percent reduction in area-averaged effectiveness, } \Delta \bar{\eta}_R = (\bar{\eta}_{\text{baseline}} - \bar{\eta}_{\text{baseline}}) \times 100 \]
\[ \nu = \text{kinematic viscosity} \]

Subscripts

\[ aw = \text{adiabatic wall} \]
\[ c = \text{cooled conditions} \]
\[ in = \text{inlet conditions} \]
\[ j = \text{coolant flow through film cooling holes} \]
\[ \infty = \text{local freestream conditions} \]

\[ R = \text{reduction} \]

References