Simulations of Multiphase Particle Deposition on a Nonaxisymmetric Contoured Endwall With Film-Cooling

Designing turbine components for maximum aerodynamic performance with adequate cooling is a critical challenge for gas turbine engineers, particularly at the endwall of a turbine, due to complex secondary flows. To complicate matters, impurities from the fuel and intake air can deposit on film-cooled components downstream of the combustor. Deposition-induced roughness can reduce cooling effectiveness and aerodynamic performance dramatically. One method commonly used for reducing the effects of secondary flows on aerodynamic performance is endwall contouring. The current study evaluates deposition effects on endwall contouring given the change to the secondary flow pattern. For the current study, deposition was dynamically simulated in a turbine cascade to determine its effects on film-cooling with and without endwall contouring. Computationally predicted impactions were in qualitative agreement with experimental deposition simulations, showing that contouring reduced deposition around strategically placed film-cooling holes. Deposition reduced cooling effectiveness by 50% on a flat endwall and 40% on an identically cooled contoured endwall. Although 40% is still a dramatic reduction in effectiveness, the method of using the endwall contouring to alter deposition effects shows promise. [DOI: 10.1115/1.4007598]

Introduction

Our way of life makes humans dependent on gas turbines for both aircraft propulsion and land-based power generation. Because of this, engineers are continually seeking methods to improve gas turbine performance by increasing turbine inlet temperatures. The main constraint associated with increasing turbine inlet temperatures is that turbine components rely on sophisticated cooling technologies, such as internal passages and film-cooling, to prevent failure due to thermal fatigue. A thorough understanding of the turbine thermal environment is necessary to advance the state of the art in turbine designs.

One area that is important to understand is the effect of particle deposition on components. Impurities that exist in the fuel and air can heat up and become soft as they pass through the combustor. Particles that remain soft as they approach the downstream turbine can deposit, which leads to increased surface roughness and blocked film-cooling holes. Because engine tests are costly, a method to simulate the three-phase flow of solid and molten particles immersed in the hot gas path was developed to determine the effects of deposition on film-cooling effectiveness.

Near the endwall of a turbine, complex secondary flows are present, which contribute to high heat transfer rates and high aerodynamic loss. The vortical flows can also sweep particles toward the endwall and result in significant deposition. Previous studies, detailed in the following section, have shown that non-axisymmetric contouring of the endwall can be successfully implemented to reduce the effects of secondary flows on heat transfer and loss. The objective of this study was to determine the effects of deposition on film-cooling with and without a non-axisymmetric contoured endwall.

In this study, deposition was simulated experimentally in a linear blade cascade. Experiments were first conducted with a film-cooled flat endwall and then simulated again with a contoured endwall having a near identical film-cooling configuration. Deposition was simulated dynamically using a wax injection technique developed by Lawson and Thole [1], and the effects of deposition on film cooling were quantified using infrared (IR) thermography. Particle trajectories and endwall accretion rates were predicted computationally and compared to experimentally simulated deposition patterns to determine the ability of simulations to capture deposition physics.

Review of Relevant Literature

Nonaxisymmetric endwall contours are also known as three-dimensional contours, since they are not restricted to symmetry in the radial or circumferential directions in an engine. The intent of a nonaxisymmetric contour is to manipulate the endwall secondary flows to prevent or limit their development in the passage.

Harvey et al. [2] presented a design methodology intended to minimize exit flow overturning and secondary kinetic energy. Hartland et al. [3] conducted experiments on the geometries designed by Harvey et al. [2] and observed a 30% reduction in secondary flow losses by adding the contoured endwall. Ingram et al. [4] conducted an experimental and computational study to compare secondary loss results. Similar to the findings by Hartland et al. [3], Ingram et al. [4] found that computations underpredicted the reduction in secondary losses achieved by adding a nonaxisymmetric endwall contour. The findings by Hartland et al. [3] and Ingram et al. [4] show that further development of computational methods is necessary to obtain the degree of accuracy of experimental research in secondary flows.

Praisner et al. [5], Knezeveci et al. [6], Lynch et al. [7], and Lynch et al. [8] all studied the effects of nonaxisymmetric endwall contouring on secondary flows using a Pack-B airfoil. The Pack-B airfoil design has been extensively used in low-pressure turbine research. Praisner et al. [5] used a computational design technique to develop a nonaxisymmetric contoured endwall for the Pack-B passages. The contour designed by Praisner et al. [5] was effective at reducing losses and turbulent kinetic energy as compared to the flat endwall, with measured and predicted reductions of 10% and...
without a nonaxisymmetric contour. Film-cooled endwall geometries were installed to determine the effects of deposition on cooling effectiveness before and after endwall contouring. They concluded that the contour reduced heat transfer by up to 20% in high heat transfer regions. Lynch et al. [8] placed a row of film-cooling holes in the region of high heat transfer in the Pack-B passage with and without endwall contouring. They compared measured and computationally predicted values of adiabatic effectiveness in the passage at film-cooling blowing ratios of 1.0 and 2.0. Although laterally averaged effectiveness was overpredicted by the shear stress transport (SST) k-ω model, effectiveness simulations replicated the trends observed by experiments. Measurements and predictions showed that the endwall contour reduced coolant spreading across the endwall. They attributed the reduction in spreading to the reduction of cross passage flow with the contoured endwall. Although the contoured endwall effectively reduced the secondary losses, it actually had a negative effect on film-cooling.

Various studies have been conducted recently to determine the negative effects of particle deposition on film-cooled turbine sections. Lawson and Thole [1,9–11] simulated deposition dynamically using wax particulate injection to determine the effects of deposition on flat plate, endwall, and leading-edge film-cooling effectiveness. Lawson and Thole [1,9–11] determined that cooling effectiveness reached an equilibrium state, at which point additional particulate injection had negligible effects on cooling effectiveness. Albert et al. [12] dynamically simulated deposition on a showerhead cooling geometry using a similar wax injection technique. In all previous wax deposition studies [1,9–12] it was concluded that deposition was sensitive to the relationship between the particulate solidification temperature and the mainstream gas temperature. Lawson and Thole [1] developed a thermal scaling parameter (TSP) to scale particle solidification times from engine conditions to laboratory conditions. The TSP was used in the current study to scale the particle phase change mechanism that is so important to deposition [13–15].

To the authors’ knowledge, no other study has been conducted to determine the effects of deposition on a film-cooled endwall with contouring. The objective of the current study was to determine the effects of deposition on a film-cooled endwall with and without a nonaxisymmetric contour. Film-cooled endwall geometries identical to those used by Lynch et al. [8] were used in the current study to quantify cooling effectiveness before and after dynamic deposition simulation.

Experimental Methods

To determine the effects of deposition on endwall film-cooling with and without endwall contouring, experiments were conducted in the same low-speed wind tunnel used by Lawson and Thole [1]. A Pack-B blade cascade test section was located in the closed-loop wind tunnel, as shown in Fig. 1. The Pack-B is a low pressure turbine blade geometry that has been used in various experimental studies in the literature [5–8]. Flow through the wind tunnel, shown in Fig. 1, was powered by a 37 kW axial fan. The primary flow was split into two secondary flow paths and a mainstream flow path upstream of the test section. A chiller system supplied cooling water to a primary heat exchanger and two secondary heat exchangers; however, it is important to note that only the top coolant flow passage was used for this study. The mainstream flow passed through a heater bank, which heated the mainstream to approximately 325 K, while the secondary heat exchanger cooled the secondary flow path to approximately 300 K. Downstream of the heater bank, the mainstream flow passed through a flow conditioning block containing screens and honeycomb for flow straightening. A turbulence grid was located 9.1 chord lengths upstream of the blade cascade and was used to achieve 4% turbulence intensity at the entrance to the blade cascade test section [7]. The auxiliary blower shown in Fig. 1 was used to supply coolant from the top coolant plenum in the wind tunnel to the film-cooling supply plenum located beneath the endwall of the cascade.

The cascade consisted of seven blades with six full passages. Two of the six passages were equipped with the endwall cooling geometry tested for this study. Figure 2 shows the endwall configuration of the Pack-B blade cascade with the passages of interest (passage 2 and passage 3) identified. The endwall was constructed out of low thermal conductivity polyurethane foam (k = 0.033 W/m-K), so that adiabatic effectiveness could be quantified using IR thermography. The film-cooling holes in passage 2 were identical to the film-cooling holes in passage 3. Identical to the study by Lynch et al. [8], hole 1 on the flat endwall was drilled at an inclination angle of 45 deg relative to the surface, while holes 2–5 were drilled at an inclination angle of 40 deg relative to the surface. All five cooling holes in the flat endwall had L/d = 4.1.

Following the flat endwall experiments, film-cooled contoured endwall passages identical to those studied by Lynch et al. [8] were installed to determine the effects of deposition on cooling with a nonaxisymmetric contoured endwall. Figure 3 shows an isometric view of the nonaxisymmetric contoured endwall, which was designed using a computational optimization technique by Praisner et al. [5]. The contoured endwall sections were cast individually in a mold using polyurethane two-part expanding foam (k = 0.033 W/m-K). The five cooling hole locations were held constant between the flat and contoured endwalls; however, because of the nonuniform surface, inclination angles of cooling holes in the contoured endwall varied from hole to hole. Holes 1 and 2 had inclination angles of 80 deg and 60 deg, while holes 3–5 had inclination angles ranging from 45 deg to 30 deg. The L/d ratios for the cooling holes in the contoured endwall varied from 7 to 8.5. All cooling holes in the flat and contoured endwalls were aligned corresponding to oil flow visualization streaklines determined by Lynch et al. [7].
Film-cooling operating conditions were characterized by the average ideal blowing ratio, $M_{\text{ideal}}$, of all five holes for a given experiment. The ideal blowing ratio was determined using measurements of plenum pressure and local endwall static pressure. The local endwall static pressures were measured at each hole exit location using pressure taps mounted in an adjacent passage without film-cooling. The operating conditions and geometric specifications used for this study are listed in Table 1 and were identical to those used by Lynch et al. [8].

The local pressure distribution was measured around the center blade in the Pack-B cascade to ensure accurate comparison to the inviscid flow pressure distribution around the mid span of the airfoil [7]. Pressure measurements were acquired from strategic locations around the mid span of every blade to ensure periodicity across the cascade.

**Adiabatic Effectiveness Measurements.** Surface temperatures were measured using a FLIR SC620 IR camera with $640 \times 480$ pixel resolution. Upon reaching steady state conditions for each experiment, five images were acquired through each of six port locations in the ceiling of the test section. Each image was calibrated independently by adjusting the effective background temperature and emissivity independently to match the IR data with the calibration thermocouple data for each corresponding location in each image. The data was then stitched together to obtain a spatially resolved matrix of endwall temperatures. The measured adiabatic effectiveness could then be calculated using the equations in the Nomenclature.

To correct for the small amount of conduction error through the low thermal conductivity endwall, a one-dimensional correction developed by Ethridge et al. [16] was applied. By blocking the cooling holes in the area of interest and acquiring temperature data using the method described above, the conduction correction could be made using the equations in the Nomenclature. Measured conduction correction values were approximately 0.06 across the entire endwall area of interest.

**Experimental Deposition Simulations.** Deposition was simulated in the blade cascade using the same two-nozzle injection system used by Lawson and Thole [1], as illustrated in Fig. 4. The Stokes number ($\text{Stk}$) was used to scale the particle inertial characteristics from engine to laboratory conditions, while the thermal scaling parameter ($\text{TSP}$) was used to scale the particle phase change characteristics.

The thermal scaling parameter was developed by Lawson and Thole [1] and was used to characterize particle phase in Lawson and Thole [1,10,11]. The TSP is the solidification time of a particle normalized by the time it takes the particle to travel from the injection location to the surface of interest. In the engine, the travel time would be the time it takes the particle to travel from the combustor to the nozzle guide vane. If the solidification time is greater than the travel time, then the particle would be in molten form upon reaching the turbine. If the solidification time is shorter than the travel time, then the particle would be in solid form upon reaching the turbine. Solidification time was calculated using a lumped mass approximation for a particle with zero slip velocity.
immersed in a gas with a lower temperature than its solidification temperature. The solidification takes place in a two-step process. First, $t_1$ is the time it takes the particle to exponentially decrease in temperature to reach the solidification temperature. Then, $t_2$ is the time it takes the particle to lose the equivalent to its latent heat of fusion while remaining at the solidification temperature. The equations for evaluating the solidification time and TSP are shown in the Nomenclature. It is important to note approximations, such as lumped mass and no-slip are not perfect, but good enough for the purpose of developing a TSP as an order of magnitude analysis to characterize particle phase between experimental cases.

A Stokes analysis was performed to determine that a median wax particle size of 34 $\mu$m was necessary to match the Stokes number for fly ash particles having a median size of 5 $\mu$m in the engine. Because TSP is highly dependent on particle size, the TSP of the median particle size was used to characterize particle phase for each experiment conducted. All blade cascade deposition simulations were conducted using wax with $T_{p,s} = 333$ K injected at $T_1 = 320$ K to achieve a TSP = 0.3 or $T_1 = 330$ K to achieve a TSP = 1.1. A liquid wax pressure of 103 kPa (15 psi) was set to achieve a wax flow rate of 1.9 g/s from each nozzle, which equated to a particle-to-mainstream volume fraction of approximately $7 \times 10^{-5}$ (510 ppmw). For that liquid wax flow rate, an atomizing air pressure of 138 kPa (20 psi) was set to achieve a median particle size of 34 $\mu$m as measured by a Malvern particle analyzer. The particle properties and scaling parameters for the blade cascade experiments are shown in Table 2 [17–21].

Two experiments were conducted for each operating condition evaluated. First, wax was injected at steady state conditions to simulate deposition for up to four minutes (900 g of wax injection). Following completion of the deposition simulation, the surface was photographed then painted black to achieve uniform emissivity across the deposit-laden surface. The calibration thermocouples were cleaned off before an adiabatic effectiveness experiment was conducted, as described in the previous section, to quantify the effect of the simulated deposition on film-cooling effectiveness.

Surface photographs were taken through the port locations used to acquire IR data. The surface photographs were obtained using the same method described by Lawson and Thole [1] using a Nikon D40X 10.2 megapixel digital single-lens reflex camera. These photographs were then stitched together to obtain a composite image of the endwall passage of interest. The composite images were then qualitatively compared with adiabatic effectiveness contour plots for each corresponding case.

### Computational Simulations
Computational simulations were performed using the discrete phase model (DPM) in the commercial computational fluid dynamics software FLUENT to predict particle tracks and accretion rates on the flat and contoured endwalls. The computational domain used for the study was developed and benchmarked by Lynch et al. [8] and is shown in Fig. 5. The grid contained $3.5 \times 10^6$ cells in a multiblock structured format, with O-grids around the airfoil and each of the film-cooling holes. Grid spacing was refined near walls such that $y^+$ values were kept below 1. A plenum was modeled under the endwall with a mass flow inlet condition set such that the ideal blowing ratio of the film cooling was matched to the experiments. The SST k-ω turbulence model [22] was used for closure of the steady Reynolds-averaged Navier–Stokes equations.

### Table 2  Particle properties and scaling parameters

<table>
<thead>
<tr>
<th></th>
<th>Engine (fly ash)</th>
<th>Laboratory (wax)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Median particle diam., $d_p$ ($\mu$m)</td>
<td>5</td>
<td>35</td>
</tr>
<tr>
<td>Particle density, $\rho_p$ $(kg/m^3)$</td>
<td>1980 [17]</td>
<td>800</td>
</tr>
<tr>
<td>Sp. latent heat of fusion, $\Delta h_{ fus}$ (J/kg)</td>
<td>650,000 [18]</td>
<td>225,600</td>
</tr>
<tr>
<td>Sp. heat, $C_p$ (J/kg-K)</td>
<td>730 [19]</td>
<td>2090</td>
</tr>
<tr>
<td>Particle solidification temp., $T_{p,s}$ (K)</td>
<td>1533 [20]</td>
<td>333</td>
</tr>
<tr>
<td>Mainstream gas temp., $T_1$ (K)</td>
<td>1500 [21]</td>
<td>321/330</td>
</tr>
<tr>
<td>Particle initial temp., $T_{p,i}$ (K)</td>
<td>1593 [21]</td>
<td>357</td>
</tr>
<tr>
<td>Gas viscosity, $\mu$ (kg/m-s)</td>
<td>$5.55 \times 10^{-5}$</td>
<td>$1.82 \times 10^{-5}$</td>
</tr>
<tr>
<td>Particle travel distance, $L_{trav}$ (m)</td>
<td>0.25</td>
<td>2.18</td>
</tr>
<tr>
<td>Particle velocity, $U_{p,i}$ (m/s)</td>
<td>63 [7]</td>
<td>10.4</td>
</tr>
<tr>
<td>Film-cooling hole diam., $d$ (mm)</td>
<td>0.51</td>
<td>4.4</td>
</tr>
<tr>
<td>Median thermal scaling parameter, TSP</td>
<td>0.76</td>
<td>0.3/1.1</td>
</tr>
<tr>
<td>Median particle Stokes number, $St_k$</td>
<td>6.54</td>
<td>6.54</td>
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</table>
The DPM model tracks the dispersed particle phase in a Lagrangian approach by integrating a force balance on individual particles. In this study, because of the sparse particulate concentration (volume fraction $\frac{7}{100}$), only one-way coupling was employed (particles did not influence the fluid phase) and only hydrodynamic drag and gravity forces were modeled. Particles were injected uniformly over the inlet of the domain using a surface injection. A Rosin–Rammler diameter distribution was used to model the particle size distribution generated for the experiments, as measured by the Malvern particle analyzer. Figure 6 shows the Rosin–Rammler particle diameter distribution compared to the experimentally measured distribution.

Turbulent dispersion of particles was modeled using the discrete random walk (DRW) model. This model simulates a turbulent eddy by calculating an effective instantaneous velocity from the local value of turbulent kinetic energy, where the value of the instantaneous velocity is scaled by a random number between 0 and 1. The effect of the idealized eddy on the particle trajectory is constrained by the smaller of an eddy timescale (also scaled by a random number) or the particle crossing time. The eddy timescale is calculated from the fluid phase turbulent timescale (proportional to $1/\varepsilon$ in the SST k-$\omega$ model), and the particle crossing time is inversely proportional to the local velocity. Since both of those timescales approach zero near the wall ($\varepsilon$ becomes large near walls), the integration along a particle trajectory near a wall with the DRW model requires an infinite number of steps to determine motion of the particle. To counteract this problem, a user-defined function was written for FLUENT to limit the eddy timescale to a lower bound of $1 \times 10^{-5}$ s. This was considered reasonable, since the Kolmogorov timescale of the flow was estimated to be on the order of $1 \times 10^{-5}$ s. Furthermore, the limiter only activated when a particle was located at $y^* < 10$ from a wall, where $y^*$ is a nondimensional wall distance. Reducing the activation value of $y^*$ from 10 to 1 did not have an effect on particle impact locations. Reducing the value of the eddy timescale limiter to $1 \times 10^{-6}$ s increased the number of incomplete particle trajectories, but did not change the overall pattern of the particle accretion on the endwall. With the user-defined function, 10,000 integration steps were sufficient to reduce the number of incomplete particle trajectories (particle does not exit domain or impact surface) to less than 1% of the total number of injected particles.

The DPM boundary conditions were set such that particles that collided with the endwall or blade surfaces were trapped and counted toward the accretion rate for the cell in which they collided with the surface. The accretion rate is defined as the mass flow rate per unit area of trapped particles. Erosion or elastic rebound of particles from a surface was not modeled, since the wax particles were expected to build up on the surfaces. Also, no change in surface roughness due to particle buildup was modeled.

**Discussion of Results**

This section describes the results from experiments and computations that were conducted to determine the effects of deposition on film-cooling with and without endwall contouring. First, the effects of endwall contouring with no deposition are discussed.
followed by the effects of deposition development, effects of contouring with deposition, effects of blowing ratio, and effects of thermal scaling parameter.

Effects of Contouring With No Deposition. Lynch et al. [8] measured adiabatic effectiveness on a film-cooled flat endwall and a film-cooled contoured endwall with identical film-cooling locations as shown in Figs. 2 and 3. For this study, adiabatic effectiveness experiments were conducted on the same geometries tested by Lynch et al. [8] to provide a baseline for experiments with simulated deposition. Figure 7 shows adiabatic effectiveness contours for the flat and contoured endwalls at \( M = 1.0 \) and \( M = 2.0 \) with no deposition. The contoured endwall was effective at reducing secondary flows [8]; however, as shown in Fig. 7, cooling effectiveness through the passage was reduced by the contour, because the region of high elevation in the passage prevented coolant from spreading across the passage.

Centerline and laterally averaged effectiveness line plots for the flat endwall at \( M = 1.0 \) are shown in Fig. 8, and effectiveness line plots for the contoured endwall at \( M = 1.0 \) are shown in Fig. 9. The data obtained in the current study and the data obtained by Lynch et al. [8] are in excellent agreement. It is important to note that the data in Figs. 8 and 9 are representative of the data from hole 3 on the flat and contoured endwalls. Flat endwall centerline data decreases from approximately 0.7 at the hole exit to approximately 0.15 at a distance of 20 d downstream. Contoured endwall centerline data decreases from approximately 0.58 at the hole exit to approximately 0.1 at a distance of 20 d downstream.

In addition to the hole 3 centerline and laterally averaged effectiveness, area-averaged effectiveness from the entire row of holes was calculated. The area represented by the white outline in Fig. 7(a) was used to calculate area-averaged effectiveness for the flat and contoured endwalls. The data from hole 3 implies that the flat endwall outperforms the contoured endwall; however, Fig. 10 shows that, when the data is averaged through the passage, the contour outperforms the flat endwall at \( M = 1.0 \). Due to the tendency of the contour to channel film coolant through the trough near the pressure side, the area-average effectiveness is higher than the single-hole lateral average for the attached film at \( M = 1.0 \).

Cooling effectiveness was lower at \( M = 2.0 \) than at \( M = 1.0 \) for both geometries, implying that coolant jets were separated from the surface. Lower effectiveness on the contoured endwall relative to the flat endwall at \( M = 2.0 \) implies that the flat endwall was less sensitive to separation effects than the contoured endwall. The difference in separation sensitivity was most likely caused by differences in cooling hole inclination angle between the two geometries. Cooling holes on the contoured endwall had high inclination angles and were more likely to separate from the surface to become entrained in the hot mainstream as compared with cooling holes on the flat endwall.

Effects of Deposition Development. Prior to conducting experiments to determine the various effects, experiments were conducted to verify that deposition would reach an equilibrium...
deposition state in a similar manner as it did in the study by Lawson and Thole [1].

Deposition simulations were individually conducted with wax injections of 300 g, 600 g, and 900 g to determine the effects of deposition development on endwall film-cooling. These development experiments were conducted using the flat endwall at $M = 1.0$ and TSP = 0.3. Figure 11 illustrates the effectiveness contours and corresponding deposition photographs before deposition, after 300 g, after 600 g, and after 900 g of wax injection. The results in Fig. 11 show the significant negative effect of deposition on cooling and, similar to the results by Lawson and Thole [1], the differences in effectiveness and deposition appears small between 600 g and 900 g of wax injection.

Figure 12 shows area-averaged effectiveness and effectiveness reduction plotted with respect to wax injection mass. Deposition reduced effectiveness by 37% after 300 g of injection and 51% after 600 g of injection. The effectiveness reduction between 600 g and 900 g of injection changed by less than 3%, implying that the deposition had reached an equilibrium state as it had in previous studies [1,9–11]. Deposition reduced the area-averaged cooling effectiveness by approximately 50% on the flat endwall at $M = 1.0$ and TSP = 0.3. Each subsequent simulation used 900 g of wax injection.

Figures 13 and 14 show effectiveness contours and corresponding deposition photographs for the flat and contoured endwalls at $M = 1.0$ and $M = 2.0$. Deposition photographs illustrate a significant difference in deposition patterns between the flat and
contoured endwalls. On the flat endwall, deposition appears relatively uniform in the passage, with a slight hint of a passage vortex deposition streak. Deposition appears to be most dense along the base of the pressure side of the blade between the blade and the film-cooling holes, increasing toward the trailing edge. On the contoured endwall, deposition is most dense in regions of high elevation along the ridges of the contour. Deposition is virtually nonexistent in the valleys of the contour. This deposition pattern exists because particles with high inertia cannot follow fluid streamlines over the ridges, resulting in deposition on surfaces that face the oncoming flow. This effect acts as an advantage in the case with the contoured endwall, where cooling holes are located on a downhill slope facing away from the flow between the contour ridge and the pressure side of the blade. Although there is some deposition around the cooling holes, it is less dense around the contoured endwall cooling holes than it is around the flat endwall cooling holes.

Figure 15 shows accretion rates predicted by the DPM model in FLUENT for the flat and contoured endwalls at $M = 1.0$. The predicted deposition patterns shown in Fig. 15 are very similar to the experimental accretion patterns in Fig. 13(b). For the flat endwall, particles are evenly distributed through most of the passage. Areas of high accretion include the pressure side-endwall junction and the blade wake, whereas accretion levels are low near the film cooling hole exits, where the film jets prevent particle accretion. For the contoured endwall, the prediction indicates nonuniform deposition compared to the flat endwall, with particles more likely to deposit in regions where the endwall is facing toward the oncoming flow and less likely to deposit in regions facing away from the flow. The agreement between computationally predicted accretion rates and experimentally simulated deposition patterns shows promise that computational techniques could be used to design contoured endwall-cooling configurations to mitigate the negative effects of deposition.

Decreased deposition downstream of the cooling holes on the contoured endwall as compared to the flat endwall results in higher cooling effectiveness on the contoured endwall after deposition. Shown in Fig. 16 are the area-averaged effectiveness values before and after deposition on the flat and contoured endwalls at $M = 1.0$ and $M = 2.0$. The results in Fig. 16 quantify what could be observed in Figs. 14 and 15. Deposition has less of an effect on contoured endwall cooling effectiveness than flat endwall cooling effectiveness. Deposition reduced effectiveness at $M = 1.0$ by 49% on the flat endwall and only 40% on the contoured endwall. Although effectiveness reduction is significant in both cases, endwall contouring shows promise in reducing the negative impact of deposition on cooling effectiveness.

**Effects of Blowing Ratio.** The results presented in the previous section showed that, in general, endwall contouring reduced the negative impact of deposition on cooling. This section discusses the effects of blowing ratio on deposition and the resulting cooling effectiveness.

The deposition photographs in Figs. 13 and 14 show that the effects of blowing ratio on deposition patterns are limited to areas within close proximity to the cooling holes. Deposition downstream of cooling holes was more dense at $M = 1.0$ than at $M = 2.0$. The high jet velocities at $M = 2.0$ prevented deposition directly downstream of holes by cooling particles and diverting their trajectories. Even with the local effects of blowing ratio on deposition, the surface roughness generally appears insensitive to blowing ratio. Alternatively, the effect of deposition on cooling effectiveness is highly sensitive to blowing ratio. Unlike the obvious decrease...
in effectiveness caused by deposition at \( M = 1.0 \), the effect of deposition on effectiveness at \( M = 2.0 \) was almost negligible. The area-averaged effectiveness values in Fig. 16 show that the effect of deposition on cooling effectiveness is small at \( M = 2.0 \). Lawson and Thole [9] observed the same trend that the effects of deposition on cooling effectiveness decreased with an increase in blowing ratio. At low coolant rates, when jets are attached to the surface, effectiveness is highly sensitive to deposition, while at high coolant rates, when jets are separated, the effects of deposition on cooling are small. When jets are attached, surface roughness can increase mixing with the mainstream. When jets are separated, they are already mixing with the mainstream and the effects of surface roughness are small.

Figure 16 shows that the contoured endwall outperforms the flat endwall at \( M = 1.0 \); however, the flat endwall outperforms the contoured endwall at \( M = 2.0 \). Deposition reduced effectiveness by less than 4% on the flat and contoured endwall at \( M = 2.0 \) as compared to the 49% and 40% reductions at \( M = 1.0 \). Before deposition, it is best to operate at \( M = 1.0 \); however, the results after deposition indicate that \( M = 2.0 \) is the better operating condition for maximum cooling effectiveness.

**Effects of Particle State.** Experiments were conducted to determine the effects of thermal scaling parameter on deposition and cooling effectiveness on the flat and contoured endwall. In addition to the experiments at \( TSP = 0.3 \), experiments were conducted at \( TSP = 1.1 \) using \( M = 1.0 \).

Figure 17 shows effectiveness contours and corresponding deposition photographs for the flat and contoured endwalls with \( TSP = 0.3 \) and \( TSP = 1.1 \) at \( M = 1.0 \). The deposition photographs in Fig. 17 show significant differences between the two TSP cases. It is important to note that, at \( TSP = 1.1 \), the median particle size of 35 \( \mu m \) had a TSP value of 1.1, meaning that particles larger than 35 \( \mu m \) had TSP values greater than 1.1. This means that most of the injected wax was in molten form upon reaching the blade cascade. The deposition resulting from the \( TSP = 1.1 \) case appeared to glaze the surface, because of the high fraction of molten particulate. Only the deposits that were in solid form upon impacting the surface showed up as white areas in the photographs. There were dense areas of solid particle deposition near film-cooling holes on both endwalls, because the jets cooled particles, causing them to condense and effectively freeze to the surface. Although more difficult to remove from the surface, the deposition at \( TSP = 1.1 \) resulted in reduced surface roughness as compared to deposition at \( TSP = 0.3 \).

Figure 18 shows effectiveness contours and deposition photographs for the flat and contoured endwalls at \( M = 1.0 \) with (a) \( TSP = 0.3 \) and (b) \( TSP = 1.1 \).

Area-averaged effectiveness before and after deposition for both endwalls and both TSP values are shown in Fig. 18. The effect of deposition on cooling was less severe at \( TSP = 1.1 \) than at \( TSP = 0.3 \), because of reduced surface roughness resulting from molten particle deposition. While effectiveness reductions were 49% and 40% for the flat and contoured endwalls at \( TSP = 0.3 \), they had slightly lower values of 29% and 34% at \( TSP = 1.1 \). Unlike the results from previous studies [1,10,11], the data shown in Fig. 18 suggests that operating at elevated temperatures may have some benefit in terms of reducing the negative impact of deposition on cooling. The difference between the current study and experiments by Lawson and Thole [1,10,11] was that higher...
TSP values were tested in the current study. For comparison, at TSP = 1.1 in the current study, particles larger than 32 µm were in molten form. At TSPmax = 1.2 in the experiments by Lawson and Thole [1], particles larger than 90 µm were in molten form. This means that there was a much higher fraction of molten particles during testing in the current study as compared to the study by Lawson and Thole [1]. This difference in results for different ranges of TSP values suggests that there exists a critical TSP. Below this critical TSP, effectiveness reduction increased with TSP, while above this critical TSP, effectiveness reduction decreased with TSP. By avoiding operation near the critical TSP, the effects of deposition on cooling can be minimized; however, more work is required to fully understand this phenomenon.

Conclusions

Experimental measurements were made to quantify the effects of deposition on film cooling with and without endwall contouring. In addition, computational simulations were conducted to predict particle accretion patterns for comparison with experiments. The effects of deposition development, blowing ratio, and thermal scaling parameter on cooling effectiveness were quantified for flat and contoured endwalls with identical film-cooling hole locations.

Similar to the findings from the flat plate, leading edge, and previous endwall studies, deposition development experiments showed that effectiveness reached an equilibrium state with increased deposition. Upon reaching the equilibrium state, further injection of particulate had a negligible effect on deposition and the resulting cooling effectiveness.

Comparing surface photographs between the flat and contoured endwalls revealed significant differences in deposition patterns. Deposition on the flat endwall was relatively uniform throughout the passage, while deposition on the contoured endwall was sensitive to endwall topography. Particles deposited densely on surfaces that faced toward the oncoming flow and sparsely on surfaces that faced away from the oncoming flow. Deposition in the vicinity of cooling holes on the contoured endwall was less severe than deposition near cooling holes on the flat endwall, because the cooling holes on the contoured endwall were located on a surface that faced away from the flow. Adiabatic effectiveness results showed that the contour slightly decreased the negative impact of deposition on cooling effectiveness. Computational predictions of particle accretion patterns showed good agreement to the surface photographs, suggesting that predictions might be useful in designing endwalls to minimize the effects of deposition.

Deposition experiments at different blowing ratios showed that cooling was less sensitive to deposition at high blowing ratios than at low blowing ratios. This finding was similar to the findings from the flat plate study by Lawson and Thole [9]. Cooling is less sensitive to deposition at high blowing ratios than low blowing ratios, because jet separation at high blowing ratios increases mixing with the mainstream, regardless of the roughness effect caused by the deposition.

Experiments conducted at two thermal scaling parameter values revealed that cooling effectiveness reductions were less sensitive to deposition when particles were in molten form than when they were in solid form. At low thermal scaling parameter values, deposition resulted from solid particle agglomeration on the surface. In contrast, at high thermal scaling parameter values, particles were molten upon impaction, causing glassy deposition. Glassy deposition appeared to result in decreased surface roughness as compared to solid particle agglomeration. Because surface roughness appeared to decrease with an increase in thermal scaling parameter, cooling effectiveness reduction also decreased with an increase in thermal scaling parameter.

The deposition experiments from the study showed that deposition had a smaller effect on film-cooling with a contoured endwall than with a flat endwall; however, more work is required to determine how to take advantage of this effect in the design of film-cooled endwalls.

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Nomenclature

\[ A = \text{surface area} \]
\[ C = \text{chord length} \]
\[ C_p = \text{particle specific heat} \]
\[ C_l = \text{turbulent viscosity constant} \]
\[ \Delta h_{\text{fus}} = \text{specific latent heat of fusion} \]
\[ d = \text{film-cooling hole diameter,} \ d = 0.44 \text{cm} \]
\[ d_0 = \text{particle diameter} \]
\[ k = \text{turbulent kinetic energy} \]
\[ L = \text{film-cooling hole length} \]
\[ L_c = \text{characteristic length for Stokes number} \]
\[ L_p = \text{particle travel distance} \]
\[ M_{\text{ideal}} = \text{ideal blowing ratio,} \]
\[ M = \sqrt{\rho_c (P_c/\infty)} = \rho (P_c/\infty) \]
\[ Ma = \text{Mach number} \]
\[ \rho = \text{static pressure} \]
\[ P = \text{blade cascade pitch} \]
\[ P_o = \text{total pressure} \]
\[ Re = \text{Reynolds number,} \ Re = \rho U_c C / \mu \]
\[ S = \text{blade span} \]
\[ Stk = \text{Stokes number,} \ Stk = \frac{p d^2 U_p/18}{L_c} \]
\[ T = \text{temperature} \]
\[ t_1 = \text{time for particle to reach} \ T_p \]
\[ t_2 = \text{time for particle to release} \ \Delta h_{\text{fus}} \]
\[ TSP = \text{thermal scaling parameter,} \ TSP = (t_1 + t_2)/(L_p/U_c) \]
\[ T_u = \text{turbulence intensity percent,} \ T_u = u_{\text{rms}}/U_c \]
\[ U = \text{velocity} \]
\[ V = \text{volume} \]
\[ y^* = \text{non-dimensional wall distance} \]
\[ Y_d = \text{Rosin–Rammler mass fraction} \]

Greek Symbols

\[ \eta = \text{adiabatic effectiveness} \]
\[ \eta = (\eta_i - \eta_i)/(1 - \eta_i) \]
\[ \bar{\eta} = \text{lateral averaged effectiveness} \]
\[ \bar{\eta} = \text{area-averaged effectiveness} \]
\[ \eta_{\text{c}} = \text{correction effectiveness (no cooling)} \]
\[ \eta_{\text{m}} = \text{measured effectiveness,} \ \eta_{\text{m}} = (T_{\text{m}} - T_{\text{in}})/(T_{\text{in}} - T_{\text{c}}) \]
\[ \bar{\eta}_s = \text{baseline area-averaged effectiveness (no deposition)} \]
\[ \rho = \text{density} \]
\[ \mu = \text{gas dynamic viscosity} \]

Subscripts

aw = \text{adiabatic wall} \]
\[ c = \text{coolant} \]
\[ e = \text{exit} \]
\[ i = \text{initial} \]
\[ in = \text{inlet} \]
\[ max = \text{pertaining to maximum particle diameter} \]
\[ p = \text{particle} \]
\[ s = \text{solidification} \]
\[ \infty = \text{mainstream} \]
References


