EXPERIMENTAL AND NUMERICAL STUDY OF IMPINGEMENT ON AN AIRFOIL LEADING-EDGE WITH AND WITHOUT SHOWERHEAD AND GILL FILM HOLES

by

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ABSTRACT

This experimental investigation deals with impingement on the leading-edge of an airfoil with and without showerhead film holes and its effects on heat transfer coefficients on the airfoil nose area as well as the pressure and suction side areas. A comparison between the experimental and numerical results are also made. The tests were run for a range of flow conditions pertinent to common practice and at an elevated range of jet Reynolds numbers (8000-48000). The major conclusions of this study were: a) the presence of showerhead film holes along the leading edge enhances the internal impingement heat transfer coefficients significantly, and b) while the numerical predictions of impingement heat transfer coefficients for the no-showerhead case were in good agreement with the measured values, the case with showerhead flow was underpredicted by as much as 30% indicating a need for a more elaborate turbulence modeling.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A hole</td>
<td>total area of all nine cross-over holes</td>
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<tr>
<td>AHT</td>
<td>heat transfer area</td>
</tr>
<tr>
<td>AR</td>
<td>cooling channel aspect ratio</td>
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<tr>
<td>dgill</td>
<td>gill hole diameter (0.508 cm)</td>
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<tr>
<td>djet</td>
<td>racetrack hole hydraulic diameter (2.228 cm)</td>
</tr>
<tr>
<td>dshower</td>
<td>showerhead hole diameter (0.594 cm)</td>
</tr>
<tr>
<td>Dh</td>
<td>cooling channel hydraulic diameter</td>
</tr>
<tr>
<td>h</td>
<td>average heat transfer coefficient on the leading-edge or side walls, [(vi/AHT) - qload]/(Ts - Tjet)</td>
</tr>
<tr>
<td>i</td>
<td>current through the foil heater on the middle copper piece</td>
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<tr>
<td>k</td>
<td>air thermal conductivity</td>
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| LMTD | logarithmic mean temperature difference, 
\[\frac{(T_s - T_m) - (T_s - T_{jet})}{\log((T_s - T_m)/(T_s - T_{jet}))}\] |
| m | total mass flow rate through all nine crossover holes |
| NuJet | average Nusselt number based on the jet diameter, 
\[h_d/d_{jet}k\] |
| P | channel perimeter without ribs |
| Pamb | ambient (lab) pressure |
| Pfeed | supply channel pressure (upstream of crossover holes) |
| PLE | leading-edge channel pressure |
| qloss | heat losses from the middle brass piece to the ambient by conduction and convection as well as the heat losses by radiation to the unheated walls |
| Rnose | channel radius at the leading edge |
| Rejet | Reynolds number based on the jet diameter, \(\rho U_{jet}d_{jet}/\mu\) |
| Tf | film temperature, \((T_s + T_m)/2\) |
| Tjet | air jet temperature |
| Tm | air mixed mean temperature |
| Ts | surface temperature |
| Ujet | jet mean velocity, \(m/vA_{hole}\) |
| Z | jet place distance to the target surface (Figure 1) |
| v | voltage drop across the foil heater on the middle copper piece |
| beta | showerhead hole angle with the channel axial direction (30°, Figure 1) |
| mu | air dynamic viscosity at jet temperature |
| rho | air density at jet temperature and pressure |
INTRODUCTION

Leading edge cooling cavities in modern gas turbine airfoils play an important role in maintaining the leading edge temperature at levels consistent with airfoil design life. These cavities often have a complex cross-sectional shape to be compatible with the external contour of the blade at the leading edge. Furthermore, to enhance the heat transfer coefficient in these cavities, they are often roughened on three walls with ribs of different geometries. The cooling flow for these geometries usually enters the cavity from one end of the airfoil flows radially to the other side or, in the most recent designs, enters the leading edge cavity from the adjacent cavity through a series of cross-over holes on the partition wall between the two cavities. In the latter case, the cross-over jets impinge on a smooth leading-edge wall and exit through the showerhead film holes, gill film holes on the pressure and suction sides, and, in some cases, forms a cross flow in the leading-edge cavity and moves toward the end of the cavity.

Various methods have been developed over the years to keep the turbine airfoils temperatures below critical levels consistent with the required life for each component. Parallel with advances in airfoil material properties, advances in airfoil cooling schemes have also been remarkable. A main objective in turbine airfoil cooling design is to achieve maximum heat removal from the airfoil metal while minimizing the required coolant flow rate. One such method is to route coolant air through serpentine passages within the airfoil and convectively remove heat from the airfoil. The coolant is then ejected either at the tip of the airfoil, through the cooling slots along the trailing edge or the film holes on the airfoil surface at critical locations. To further enhance the heat transfer, the cooling channel walls are often roughened with ribs. Extensive research has been conducted on various aspects of the rib-roughened channels and it is concluded that geometric parameters such as passage aspect ratio, rib height to passage hydraulic diameter or blockage ratio, rib angle of attack, the manner in which the ribs are positioned relative to one another (in-line, staggered, crisscross, etc.), rib pitch-to-height ratio and rib shape (round versus sharp corners, fillets, rib aspect ratio, and skewness towards the flow direction) have pronounced effects on both local and overall heat transfer coefficients. The interested reader is referred to the work of investigators such as Burggraf [1], Chandra and Han [2], El-Husayni et al. [3], Han [4], Han et al. [5, 6, 7], Metzger et al. [8, 9, 10], Taslim and Spring [11,12], Taslim et al. [13, 14, 15], Webb et al. [16] and Zhang et al. [17].

Airfoil leading-edge surface, being exposed to very high gas temperatures, is often a life-limiting region and requires more complex cooling schemes especially in modern gas turbines with elevated turbine inlet temperatures. A combination of convective and film cooling is used in conventional designs to maintain the leading-edge metal temperature at levels consistent with airfoil design life. This study focuses on the leading-edge jet impingement and effects that showerhead film holes have on the impingement heat transfer coefficient. In this flow arrangement, the coolant enters the leading-edge cooling cavity as jets from the adjacent cavity through a series of crossover holes on the partition wall between the two cavities. The cross-over jets impinge on the leading-edge wall and exit through the leading-edge film holes on the pressure and suction sides, or form a crossflow in the leading-edge cavity and move toward the airfoil tip. A survey of many existing gas turbine airfoil geometries show that, for analytical as well as experimental analyses, such cavities can be simplified by simulating the shape as a four-sided polygon with one curved side that simulates the leading edge curvature, a rectangle with one curved side (often the smaller side) or a trapezoid, the smaller base of which is replaced with a curved wall. The available data in open literature is mostly for the jet impingement on flat surfaces that are smooth or rib-roughened and a few cases of impingement on concave but smooth surfaces. These studies include the work of Chupp et al. [18], Metzger et al. [19], Kercher and Tabakoff [20], Florschetz, et al. [21, 22, 23], Metzger and Bunker [24], Bunker and Metzger [25], Van Treuren et al. [26], Chang et al. [27], Huang et al. [28], and Akella and Han [29]. However, as dictated by the external shape of an airfoil leading edge, the test section in this investigation was a symmetric channel with a circular nose, two tapered side walls and a flat fourth wall on which the crossover jets were positioned. Experimental results for this setup with circular as well as racetrack-shaped crossover holes and for a variety of target surface geometries have already been reported by Taslim et al. [30-33]. The present study, however, deals with the impingement of jets issued from racetrack-shaped crossover holes on a concave target surface with and without the presence of showerhead film holes. Depending on the flow arrangement, the impingement air was ejected entirely through a row of holes on the target surface along the leading edge simulating the airfoil showerhead film holes, or split through the showerhead holes and two rows of holes on the side walls representing the pressure and suction side “gill” film holes, or partially (76%) through the showerhead holes and the balance (24%) through one end of the channel representing an airfoil tip. Data were gathered for a range of jet Reynolds number up to 48000 and were compared with those numerically calculated.

TEST SECTIONS

Figures 1 and 2 show schematically the rig layout and its cross-sectional area, the target surface, and the cross-over hole geometry. A conventional technique of heated walls in conjunction with thermocouples was used to measure the heat transfer coefficient. The test wall, where all measurements were taken, consisted of nine removable machined copper pieces which were heated by foil heaters attached on the back of the pieces. By proper adjustment of the ohmic power to the foil heater immediately underneath the copper pieces, the desirable surface temperature was obtained. The test rig was 85.5 cm long. The circular wall simulating the leading-edge nose with an inner radius of 1.1 cm and an arc angle of 137° was made of acrylic plastic with a 9.9 cm long recess in the middle to house the nine copper...
For the showerhead flow cases, sixty one 0.595-cm-diameter holes at a center-to-center distance of 0.87 cm were drilled along the leading edge nose at a 90° angle with the channel longitudinal axis. Seven of these holes passed through each copper piece on the leading edge (a total of twenty one holes on the copper pieces). The remaining forty one holes were drilled symmetrically on both sides of the copper pieces on the Lexan® nose piece. This single row of holes, with properly scaled flow area, simulated an airfoil showerhead hole design that is typically configured as two rows. This test rig, however, was limited to one row of holes because the brass pieces were covered with etched-foil heaters through which could not be drilled. A flange on each side of the leading-edge piece facilitated the connection of the side walls to this piece. A circular recess along the inner radius with a depth of 3.2 mm and a length of 18.5 cm allowed the copper pieces to be fitted into the Lexan® shell flush with the channel surface. The two identical side channels with cross-sectional areas of 271 cm² (15.24 cm by 17.78 cm) and the same length as the leading-edge piece were also made of clear acrylic plastic. The side channels’ main function was to maintain the dump pressure to consequently control the amount of flow through the “gill” holes on the airfoil suction and pressure sides. Eight angled cylindrical holes with a diameter of 5.08 mm and a center-to-center
distance of 3.658 cm were drilled on each side channel wall at an angle of 30° with the side wall to simulate gill holes on the suction and pressure sides of an airfoil. These holes were staggered along the length of the test section with respect to the crossover jet holes on the jet plate. A valve at the exit of each side channel controls the amount of gill hole flow on each side. A removable 2.54 cm thick jet plate corresponding to a Z/d_{jet} value of 2.8 was made of acrylic plastic to produce the impinging jets. Nine racetrack-shaped holes with a cross section shown in Figure 2 (made of two half circles of 1.524 cm diameter and a rectangle of 1.524 cm by 2.06 cm) were drilled at a distance of 6.17 cm from each other (center-to-center) on the jet plate. The jet plate was attached and sealed to the side channel walls to simulate the partition wall between the leading-edge and its adjacent cooling cavity in an airfoil. The crossover holes were centered with respect to both the length and width of the jet plate. For the nominal position of the jet plate, a jet impinged at the center of each copper piece. The removable copper pieces, installed in the acrylic nose piece, provided the ability to change the impingement surface geometries in the test rig. Two sets of nose copper piece geometries were manufactured and tested: (1) a set with no showerhead holes and (2) a set of three with showerhead holes. On each piece seven 0.596 cm diameter holes were drilled. Custom-made thin etched-foil heaters with a thickness of about 0.2 mm were glued around the outer surface of each copper piece to provide the necessary heat flux. For each geometry, three identical brass pieces, separated by a 1 mm thick rubber insulator, were mounted next to each other. Heat transfer coefficients were measured on the middle piece while the other two pieces acted as guard heaters to minimize the heat losses to the adjacent walls. The test section wall temperature was adjusted to a desirable level by varying the ohmic power to these heaters. Six thermocouples were embedded in each of the three middle copper pieces with their beads close to the exposed surface. Three thermocouples were embedded in each guard copper piece. The average of the six thermocouple readings in the middle copper pieces, which, if different, only differed by a fraction of a degree, was used as the surface temperature in the data reduction software for the average heat transfer coefficient. A nominal surface temperature of 45°C was selected so that with a jet temperature of about 20°C, a reasonable 25°C temperature difference existed between the wall surface and air. AC power was supplied to individual heaters through an existing power panel with individual Variacs for each heater. Heat flux for each heater was calculated using the measured voltage and amperage, and the surface area of each heater. Typical amperage and voltage levels for each heater varied from 0.23 - 0.4 Amps and 20-45 Volts, respectively. Air properties were evaluated at jet temperature.

The trapezoidal supply channel was formed by the exterior walls of the side channels, the jet plate and a 1.27 cm thick aluminum back plate as shown in Figure 1. The end caps were fixed such that it was possible to control the flow and pressure in each channel, thus simulating many variations.
Percentage of air flow rate through each cross
over hole for some flow arrangements. The impingement heat transfer coefficient was calculated. When showerhead holes were active, the heat loss in the entrance regions of the showerhead holes, drilled through the copper, were taken into consideration as well. The fourth and sixth

Figure 6 Nusselt number variation with Reynolds
number for the nominal case of no showerhead flow.

COMPUTATIONAL MODELS

The computational models were constructed for a representative repeated domain with two symmetric planes in each case. Figure 4 shows this representative domain for the target geometry with showerhead holes and details of the mesh distribution on the surface of the domain. The computational domain size for the no-showerhead geometry was the same. The CFD analysis was performed using Fluent/UNS solver by Fluent, Inc., a pressure-correction based, multi-block, multigrid, unstructured/adaptive solver. Standard high Reynolds number k-ε turbulence model in conjunction with the generalized wall function was used for turbulence closure. The average y⁺ for the first layer of cells was calculated to vary between 4.5 to 10 for all cases. Other available turbulence models in this commercial code, short of two-layer model which required a change in mesh arrangement for each geometry and was beyond the scope of this investigation, were also tested and did not produce results significantly different from those of k-ε model. Mesh independence was achieved at about 700,000 cells for a typical model. Cells in all models were entirely hexagonal, a preferred choice for CFD analyses, and were varied in size bi-geometrically from the boundaries to the center of the computational domain in order to have finer mesh close to the boundaries. Figure 4 shows the mesh distribution around the periphery of a typical model.

RESULTS AND DISCUSSION

A total of 112 tests were run in this investigation. All tests had several common features. There were always nine impinging jets issuing from the jet plate. The middle jet (fifth) always impinged on the copper leading-edge test piece in the middle of the test section and the reported heat transfer results are always for the copper test pieces in the middle (one leading-edge piece and two side pieces). Heat losses from the middle copper pieces to the ambient by conduction and convection as well as the heat losses by radiation to the unheated walls were taken into consideration when the

impingement heat transfer coefficient was calculated. When showerhead holes were active, the heat loss in the entrance regions of the showerhead holes, drilled through the copper, were taken into consideration as well. The fourth and sixth

Figure 6 Nusselt number variation with Reynolds
number for the nominal case of no showerhead flow.
jets impinged on the side copper pieces that acted as guard heaters. The remaining six jets impinged on the acrylic leading-edge wall to simulate the flow field in a typical leading-edge cooling channel. The jet Reynolds number is based on the total measured mass flow rate and the total area of the nine impingement holes. This choice of Reynolds number was based on the airfoil thermal circuit designers’ common practice. However, Figure 5 shows that the fifth jet for which the results are presented here, has mass flow rates of 11.234%, 11.275%, 11.236%, 11.036% and 11.047% of the total flow for the tested geometries and flow arrangements. Given that the average flow for each hole is 11.11%, if one wishes to find the jet Reynolds number based on the mass flow rate through the fifth hole, one can use the multipliers 1.012, 1.015, 1.011, 0.996 and 0.994 for different cases, respectively, which corresponds to a maximum increase of 1.5% and a maximum decrease of 0.6%.

Four outflow arrangements for the exiting cooling air, shown in Figure 3b and Table 1, were tested. The “nominal flow” case was the case in which the air, after impinging on the leading-edge wall, was ejected equally through the side channel holes which simulate the gill holes. Middle and at each end of the supply and leading-edge channels, measured no significant difference between different locations along each channel (about 1 cm of water column for a supply pressure ranging from 110 to 172 KPa). Experimental uncertainty in heat transfer coefficient, following the method of Kline and McClintock [34] was determined to be 6%. For clarity, we will discuss the results of no-showerhead flow and showerhead flow separately and then compare them.

**No Showerhead Flow**

Four outflow arrangements for the exiting cooling air, shown in Figure 3b and Table 1, were tested. The “nominal flow” case was the case in which the air, after impinging on the leading-edge wall, was ejected equally through the side channel holes which simulate the gill holes. Middle and at each end of the supply and leading-edge channels, measured no significant difference between different locations along each channel (about 1 cm of water column for a supply pressure ranging from 110 to 172 KPa). Experimental uncertainty in heat transfer coefficient, following the method of Kline and McClintock [34] was determined to be 6%. For clarity, we will discuss the results of no-showerhead flow and showerhead flow separately and then compare them.

**Table 1 Outflow arrangements.**

<table>
<thead>
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<th>Outflow Arrangements (Figure 3)</th>
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<td>NO SHOWERHEAD</td>
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<td>WITH SHOWERHEAD</td>
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was an equally significant drop in heat transfer coefficient because the cooling jets on their way out of the leading-edge channel did not have much interaction with a good portion of the side wall with plugged gill holes. The heat transfer coefficient levels on the leading-edge wall fell in between the two sidewall values. Again, for the reason mentioned above, flow entering from one side of the supply channel produced higher heat transfer coefficients. The maximum difference was measured on the leading-edge wall, at the higher end of the Reynolds number range, to be about 10%.

Figure 8 compares the results of two other outflow conditions of circular and crossflow. The axial flow was caused by the four upstream jets in each case. It is seen that the circular outflow case, in general, produced lower heat transfer coefficients than crossflow case especially on the leading-edge wall. The reason for this behavior is that, based on our flow circuit analysis, the sum of mass flow rates for the four upstream jets in the circular outflow case was less than that for the crossflow case. It is also seen that the leading-edge wall heat transfer coefficients in the crossflow case are higher than those on the side walls indicating that directing the jets towards the side gill holes on their way out of the leading edge channel is a more effective way to produce higher heat transfer coefficients on the side walls.

A comparison is made between the test results of all outflow arrangements in Figure 9. For clarity, this comparison is for the case of inflow from one end of the supply channel. The case of outflow from both ends produces the
Showerhead Flow

Three outflow arrangements for the exiting cooling air, shown in Figure 3c and Table 1, were tested. The “100% showerhead flow” case was the case in which all cooling air, after impinging on the leading-edge wall, was ejected through a row of holes on the target surface. The “symmetric flow” case was the case in which 73.5% of the cooling air, after impinging on the leading-edge wall, was ejected through the leading-edge holes and the remaining 26.5% through the side holes simulating the “gill” holes on the pressure and suction sides of an airfoil. The “one-sided flow” case was the case in which 85% of the cooling air, after impinging on the leading-edge wall, was ejected through the leading-edge holes and the remaining 15% through the side holes on one side of the leading-edge channel. The “crossflow” case was the case in which 85% of the cooling air, after impinging on the leading-edge wall, was ejected through the leading-edge holes while the remaining 15% was ejected from one end of the leading-edge channel simulating the airfoil tip flow. The “circular flow” case was similar to the cross-flow case except that the axial flow was ejected from the same end of the rig it entered the supply channel thus creating a circular flow. In the latter two flow arrangements, portion of the four jets upstream of the middle jet that was not ejected through the leading-edge holes (spent air) formed a crossflow that affected the impingement heat transfer coefficient.

Heat transfer results for the case of 100% showerhead flow are shown in Figure 10. Unlike the no-showerhead cases discussed previously, a big difference is noticed between the heat transfer coefficients on the leading-edge wall and on the side walls. This was expected because the only way out for the impinging jets is through the leading-edge holes. Thus their cooling effects are mostly felt by the leading-edge wall. Only a small portion of the cooling jets, after impingement, will have interactions with the side walls. As a result, a lower heat transfer coefficient was measured on the side wall. For the same reason given for the no-showerhead case, the inflow from one end of the supply channel produced higher heat transfer coefficients than when flow entered from both ends of the supply channel. It is, furthermore, noticed that the heat transfer coefficient levels are in general higher than those for the no showerhead cases. This behavior was continuously observed in our previous studies [30,31] as well. The mere presence of showerhead holes draws the impinging jets more effectively towards the leading-edge walls thus producing higher heat transfer coefficients. Data reduction was also performed usi
ng the logarithmic meat temperature difference for the calculation of the impingement heat transfer coefficients and the film temperature for the evaluation of air properties. The LMTD-based results for the entire 112 tests showed a reduction of less than 1% in the Nusselt numbers for the no-showerhead flow cases and an increase of less than 1% for the showerhead flow cases. Figure 10 shows the results of the two methods for the showerhead flow cases. The LMTD results for the no showerhead cases are not shown since the data points were practically on top of each other.

Figure 11 shows the heat transfer results of a combined showerhead and gill hole flows (symmetric flow). Percentages of outflow from the showerhead and gill holes were 73.5% and 26.5% respectively. Compared with the 100% showerhead case, a slight decrease and a slight increase in heat transfer coefficients on the leading-edge and side walls were measured, respectively. These changes go along with the changes of mass flow rates through the showerhead and gill holes. Again, for the same reason given for the no-showerhead case, the inflow from one end of the supply channel produced higher heat transfer coefficients than when flow entered from both ends of the supply channel.

Figure 12 shows the heat transfer results of a combined showerhead and gill hole flows. In this outflow arrangement, the gill holes were open only on one side. Percentages of outflow from the showerhead and gill holes were 85% and 15% respectively. Compared with the case of gill holes on both sides, a maximum increase of 6.5% in heat transfer coefficient on the side with open gill holes and a decrease of 6% on the side with plugged gill holes were measured. A slight increase of about 4% in heat transfer coefficient was measured on the leading-edge wall. These changes go along with the changes of mass flow rates through the showerhead and gill holes. Again, for the same reason given for the no-showerhead case, the inflow from one end of the supply channel produced higher heat transfer coefficients than when flow entered from both ends of the supply channel.

Figure 13 shows the heat transfer results of a combination of showerhead and cross or circular flows.
Representative numerical heat transfer coefficient variation on the side wall.

Geometry and Outflow
- No Shower Nominal $P_{\text{jet}}/P_{\text{amb}}$
- No Shower One-Sided $P_{\text{jet}}/P_{\text{amb}}$
- No Shower Crossflow $P_{\text{jet}}/P_{\text{amb}}$
- No Shower Crossflow $P_{\text{LE}}/P_{\text{amb}}$
- 100% Showerhead $P_{\text{jet}}/P_{\text{amb}}$
- 73.5% Showerhead + 26.5% Gill $P_{\text{jet}}/P_{\text{amb}}$
- 85% Showerhead + 15% Gill $P_{\text{jet}}/P_{\text{amb}}$
- 85% Showerhead + 15% Gill $P_{\text{LE}}/P_{\text{amb}}$

Figure 17  Representative numerical heat transfer coefficient variation on the side wall.

Static pressures in the feed and leading-edge channels, normalized with the lab pressure, for all outflow arrangements are shown in Figure 16. Those cases with no showerhead holes show lower pressure ratios across the crossover and gill holes and different outflow arrangements show very close pressure ratios. The showerhead cases, however, show much higher pressure ratios due to lower showerhead holes exit area compared to those of the gill holes. For this reason the case of 100% showerhead outflow represents the maximum pressure ratio while the no showerhead case with the crossflow arrangement showed the lowest pressure ratio since the flow exited from the open end of the leading-edge channel.

Numerical Results

Representative CFD results are compared with the experimental data in Figures 14 and 18. CFD models with constant heat flux boundary conditions identical to the tested geometry for each case were run on PC Pentium4, 1.6 GHz machines with 512 MB memory. A typical case took about 1000 iterations and about four to five hours to converge. Good agreement between the measured and numerically calculated impingement heat transfer coefficients are observed for the no-showehead case. The CFD results for the leading-edge heat transfer coefficients are underpredicted.
with a maximum difference of about 16% for smallest Reynolds number and a minimum of about 1.3% in the middle range of the Reynolds number. The CFD results for the side wall heat transfer coefficients, however, are generally overpredicted with an average difference of about 10%. Typical heat transfer coefficient variations on the leading-edge and side walls for the no-showerhead case are shown in Figures 15 and 17. Since, taking advantage of symmetry, only half of the domain was modeled, the high heat transfer coefficient regions in these figures correspond to the impingement location in Figure 15 and gill hole location in Figure 17. An area-weighted Nusselt number is reported in these two figures. Finally, a comparison is made between the test and CFD results for the 100% showerhead case in Figure 18. A remarkable underestimation is observed by the CFD both on the leading-edge and side walls. It is speculated that the mixing effects of jet after impingements and their interaction with the leading-edge and side walls were not captured by the CFD. A more dense mesh, combined with a different turbulence model, including a two-layer, could improve the CFD results. That effort is underway and results will be reported later.

CONCLUSIONS

Leading-edge impingement heat transfer coefficients with and without the presence of showerhead holes were measured for a variety of outflow arrangements. Heat transfer results on the leading-edge nose and on the side walls were measured and reported separately. Representative CFD results were also presented. Major conclusions of this study were:

a) for the same coolant flow, the heat transfer coefficients were higher for those cases with the showerhead flow,
b) The no-showerhead cases produced its highest heat transfer coefficient levels when coolant was ejected from the gill holes on one side of the channel,
c) the agreement between the numerical and experimental results for the no-showerhead flow case was reasonable with an average difference of about 10%. However, the CFD results were underpredicted up to 30% when impingement with showerhead flow was modeled.

REFERENCES


