Louver Slot Cooling and Full Coverage Film Cooling with Combination Coolant

Avery Stone Fairbanks

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Louver Slot Cooling and Full Coverage Film Cooling with Combination Coolant

by

Avery Stone Fairbanks

An Honors Capstone

submitted in partial fulfillment of the requirements

for the Honors Diploma

to

The Honors College

of

The University of Alabama in Huntsville

December 3, 2019

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Dec. 1, 2019

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Date

12-02-2019

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Date

12-4-19

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Honors College Dean

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Avery Fairbanks

Student Name (printed)

Student Signature

December 1, 2019

Date
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Preface

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+ Graduate Student
* Eminent Scholar in Propulsion, Professor of Mechanical and Aerospace Engineering
## Undergraduate Student
Abstract

Within the present investigation, a louver slot is employed upstream of an array full coverage film cooling holes. Cooling air is supplied using a combination arrangement, with crossflow and impingement together. The louver consists of a row of film cooling holes, contained within a specially designed device which concentrates, and directs the coolant from a slot, so that it then advects as a layer downstream along the test surface. This louver-supplied coolant is then supplemented by coolant which emerges from different rows of downstream film cooling holes. The same coolant supply passage is employed for the louver row of holes, as well as for the film cooling holes, such that different louver and film cooling mass flow rates are set by different hole diameters for the two different types of cooling holes. The results presented are different from those from past investigations, because of the use and arrangement of the louver slot, and because of the unique coolant supply configurations. The experimental results are provided for mainstream Reynolds numbers from 107000 to 114000. Full coverage blowing ratios are constant with streamwise location along the test surface, and range from 3.69 to 5.70. Corresponding louver slot blowing ratios then range from 1.72 to 2.66. Provided are adiabatic effectiveness and heat transfer coefficient values, which are measured along the mainstream side of the test plate. Both types of data show less variation with streamwise development location, relative to results obtained without a louver employed, when compared at the same approximate effective blowing ratio, mainstream Reynolds number, crossflow Reynolds number, and impingement jet Reynolds number. When compared at the same effective blowing ratio or the same impingement jet Reynolds number, spanwise-averaged heat transfer coefficients are consistently lower, especially for the downstream portions of the test plate, when the louver is utilized. With the same type of comparisons, the presence of the louver slot results in significantly higher magnitudes of spanwise-averaged adiabatic film cooling effectiveness, particularly at and near the upstream portions of the test plate. With such characteristics, dramatic increases in thermal protection are provided by the presence of the louver slot, the magnitudes of which vary with experimental condition and test surface location.
Introduction

Very little information is available regarding the use of louver slot cooling to provide thermal protection of combustor liners within gas turbine engines. Of the very limited number of past investigations which consider louver slot cooling, one of the earliest is described by Juhasz and Marek [1]. These investigators use a variety of slot arrangements within a simulated combustor segment of a gas turbine, with a rectangular cross-section. Correlation equations are provided which match experimental results, which are based upon a mixing model for local flow turbulence. According to Lefebvre [2], such slot arrangements, including annular slot configurations, are an efficient means of providing enhanced thermal protection to the inner wall of a combustor liner, provided axial injection paths are utilized. Jia et al. [3] use both experimental and numerical tools to document the performance, at different blowing ratios, of angled film cooling slots. According to these investigators, different boundary condition arrangements affect numerically obtained velocity profiles in a significant manner, but do not affect film cooling effectiveness distributions. Cooling injection angle also affects the size and development of the resulting recirculation bubbles. Ceccherini et al. [4] consider overall influences of slot, effusion, and dilution holes, using both experimental measurements and numerical predictions. The investigators indicate that cooling effectiveness magnitudes and distributions are affected in a significant manner by values of the exit velocity associated with effusion cooling. In a follow-up study with the same liner cooling configurations, Andreini et al. [5] address heat transfer coefficient behavior also using numerical prediction tools. In a later experimental investigation, Andreini et al. [6] investigate magnitudes of heat transfer coefficient, heat flux reduction, and film cooling effectiveness downstream of louver slots. Investigated are the effects of blowing ratio and velocity ratio, for experimental configuration which are associated with operating combustor components within aero engines.

Inanli et al. [7] employ six different slot configurations in combination with several different effusion cooling arrangements. Each louver device is referred to as a leap geometry, with investigation of both flat and angled leap geometries. Effusion cooling configurations utilize two different effusion hole angles. With consideration of film cooling effectiveness distributions, the straight leap geometry generally provides better performance relative to the angled arrangement, provided comparisons are made at the same blowing ratio. With the straight geometry, magnitudes of mean cooling effectiveness range from 0.60 to 0.70. Kiyici et al. [8] provide numerically predicted results for the same arrangements and experimental conditions which are employed by
Inanli et al. [7]. Considered by Kiyici et al. [8] are three different blowing ratio values, and three different slot heights. The numerical results, and the associated experimental data, show that mean effectiveness varies only slightly as either streamwise location or blowing ratio is varied. Da Silva et al. [9] describe film cooling effectiveness and local velocity variations associated with a louver combined scheme for a freestream velocity of 6 m/s and an inlet hole coolant blowing ratio of 0.87. Centerline film cooling effectiveness values range from magnitudes near 1.0, with decreasing values with streamwise development, such that values eventually approach 0.2 to 0.5, depending upon the magnitude of blowing ratio.

Considered within the present investigation are experimentally measured results wherein a louver slot is employed upstream of an array full coverage film cooling holes. The results presented are different from those from past investigations, because of the particular louver slot arrangement which is employed, including its location upstream of an array of full-coverage film cooling holes, and because of the unique coolant supply configurations. A combination arrangement is employed to supply the cooling air with both crossflow and an impingement jet array used together. The louver consists of a row of film cooling holes, contained within a specially designed device which concentrates, and directs the coolant from a slot, so that it then advects as a layer downstream along the test surface. This louver-supplied coolant is then supplemented by coolant which emerges from different rows of downstream film cooling holes. The same coolant supply passage is employed for the louver row of holes, as well as for the film cooling holes, such that different louver and film cooling mass flow rates and blowing ratios are set by different hole diameters for the two different types of cooling holes. Experimental results are provided for mainstream Reynolds numbers from 107000 to 114000, and full coverage blowing ratios from 3.69 to 5.70, with constant values as the flow develops in the streamwise direction along the test surface. Corresponding louver slot blowing ratios then range from 1.72 to 2.66. Along the mainstream side of the test plate, provided are measured distributions of local and spanwise-averaged adiabatic effectiveness and heat transfer coefficient values.
## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$BR$</td>
<td>effusion cooling blowing ratio</td>
</tr>
<tr>
<td>$C_d$</td>
<td>discharge coefficient</td>
</tr>
<tr>
<td>$CR$</td>
<td>main flow passage contraction ratio</td>
</tr>
<tr>
<td>$d_e$</td>
<td>film cooling hole diameter</td>
</tr>
<tr>
<td>$DH$</td>
<td>hydraulic diameter</td>
</tr>
<tr>
<td>$DR$</td>
<td>effusion cooling density ratio</td>
</tr>
<tr>
<td>$h$</td>
<td>local iso-energetic heat transfer coefficient</td>
</tr>
<tr>
<td>$\bar{h}$</td>
<td>line-averaged iso-energetic heat transfer coefficient</td>
</tr>
<tr>
<td>$I$</td>
<td>effusion cooling momentum flux ratio</td>
</tr>
<tr>
<td>$P$</td>
<td>pressure</td>
</tr>
<tr>
<td>$\Delta P$</td>
<td>differential pressure</td>
</tr>
<tr>
<td>$\dot{q}_w$</td>
<td>surface heat flux</td>
</tr>
<tr>
<td>$Re_{cf}$</td>
<td>crossflow Reynolds number, $\frac{DH_{cf}V_{cf}}{v_{cf}}$</td>
</tr>
<tr>
<td>$Re_e$</td>
<td>effusion flow Reynolds number, $\frac{V_{ef}d_e}{v_{ef}}$</td>
</tr>
<tr>
<td>$Re_i$</td>
<td>impingement flow Reynolds number, $\frac{\rho_sV_id_i}{\mu_l}$</td>
</tr>
<tr>
<td>$Re_{ms}$</td>
<td>main flow Reynolds number, $\frac{DH_{ms}V_{ms}}{v_{ms}}$</td>
</tr>
<tr>
<td>$Re_{ms,avg}$</td>
<td>main flow Reynolds number, $\frac{DH_{ms}V_{ms,avg}}{v_{ms}}$</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature</td>
</tr>
<tr>
<td>$T_{aw}$</td>
<td>local adiabatic wall temperature</td>
</tr>
<tr>
<td>$T_{surf}$</td>
<td>surface/wall temperature</td>
</tr>
</tbody>
</table>
\( T_s \) static temperature
\( T_t \) stagnation temperature
\( V \) time-averaged flow velocity
VR effusion cooling velocity ratio
x streamwise coordinate
\( X \) streamwise film hole spacing
y spanwise coordinate
\( Y \) spanwise film hole spacing

**Greek Symbols**
\( \eta \) local adiabatic film cooling effectiveness
\( \bar{\eta} \) line-averaged adiabatic film cooling effectiveness
\( \mu \) absolute viscosity
\( \rho \) static air density

**Subscripts**
Avg average value
aw adiabatic wall value
c crossflow coolant supply channel value
cf crossflow value
ef effusion jet value
i impingement value
imp impingement value
M main flow value
ms main flow value based upon inlet hydraulic diameter, and freestream flow velocity at inlet of main flow passage

ms,avg main flow value based upon inlet hydraulic diameter, and flow velocity averaged along streamwise length of main flow passage

s static value

surf surface value

t stagnation value
Experimental Apparatus and Procedures

Experimental apparatus and procedures details are provided by Rogers et al. [10] and Ritchie et al. [11]. Brief summaries of associated details are provided here.

**Double Wall Cooling Test Facility, Test Section, and Test Surfaces**

The present experimental facility consists of a double wall cooling test section and the equipment employed to supply properly conditioned air for the mainstream flow, the full-coverage film cooling flow, the louver slot flow, the impingement flow, and the crossflow. Within the present investigation, the film cooling air is supplied by the crossflow supply and the impingement supply. Figure 1 presents information related to the test section configuration. Included in part (a) is a three-dimensional view of the test plate, including the louver slot device and the full-coverage film cooling holes. Provided in part (b) is a side, cross-sectional view of louver slot.

![Diagram](image-url)
Figure 1: Test section configuration. (a) Three-dimensional view of test plate, including louver slot device and full-coverage film cooling holes. (b) Side, cross-sectional view of louver slot. All dimensions are given in millimeters.

Additional details regarding the test facility, test section, test surfaces, optical instrumentation arrangements, mesh heater devices, crossflow and impingement cooling arrangements, and test plate design and construction are provided by Rogers et al. [10] and Ritchie et al. [11]. According to these sources, mesh heaters are employed to generate a timewise step increase in air flow static temperature of the mainstream air, after all facility flow conditions are established. With this arrangement, transient infrared thermography is employed to measure spatially resolved distributions of surface heat transfer coefficients and adiabatic film cooling effectiveness values.

These sources [10,11] also indicate that the effusion cooling test plate contains 60 holes arranged in 6 staggered rows, with 10 holes within each row. Streamwise (X/d_e) and spanwise (Y/d_e) hole spacings are 15 and 4, respectively. Hole rows are spaced 95.25 mm apart, the holes in each row are 25.4 mm apart, such that every other row is offset by 12.7 mm. Each hole has a diameter of 6.35 mm and is inclined at an angle of 25° relative to the surface of the test plate. Effusion plate thickness is 3.0 effusion hole diameters.
The impingement plate contains holes that direct jets of air from the impingement supply plenum into a target surface, which is the cold side of the effusion cooling test plate. Each impingement jet centerline is located midway between the centerlines of effusion hole entrances. The impingement plate is made of 19 mm thick, optically transparent acrylic. As such, impingement plate thickness is 3.0 effusion hole diameters. The plate contains 60 holes arranged in 6 offset rows, with 10 holes per row. Rows are spaced 95.3 mm apart, the holes in each row are 25.4 mm apart, such that every other row is offset by 12.7 mm. Each hole has a diameter of 8.3 mm and is oriented at an angle of 90° relative to the surface of the plate. The first row of holes exits the plate 63.5 mm from the start of the plate. The crossflow passage or impingement passage height is 14 effusion hole diameters, or 14\(d_e\) [11].

**Measurement of Flow Temperatures and Pressures**

Apparatus and procedures for measurement of flow temperatures pressures, density values, mass flow rates, and velocity magnitudes are described by Rogers et al. [10] and Ritchie et al. [11].

**Impingement Flow Conditions, Crossflow Conditions, Film Cooling Flow Conditions, Louver Slot Flow Conditions, Main Flow Conditions, and Parameters Determination**

Information on determination of impingement flow conditions, crossflow conditions, film cooling flow conditions, louver slot flow conditions, main flow conditions, and determination of associated parameters, is provided by Ritchie et al. [11]. As indicated, crossflows and impingement jet arrays are used simultaneously to supply the full-coverage film cooling air, as is also described by Ritchie et al. [11]. The resulting experimental conditions associated with the present investigation, are provided within Tables 1, 2, and 3.

**Measurement of Surface Heat Transfer Coefficient and Adiabatic Wall Temperature Distributions**

Experimental apparatus and procedures for measurement of surface heat transfer coefficients and adiabatic wall temperature distributions, along the test plate surface adjacent to the main flow passage, are described by Rogers et al. [10] and Ritchie et al. [11].

To obtain the present data, the infrared radiation emitted by the film cooled interior surface of the channel is captured using a FLIR Systems Inc. ThermoVision® T650sc Infrared Camera (S/N 22700776). This camera is operated with a FLIR T197915 80 degree infrared lens. Temperatures, measured using the calibrated, copper-constantan (Type-T) thermocouples distributed along the test
surface adjacent to the flow, are used to perform the *in situ* calibrations simultaneously as the radiation contours from surface temperature variations are recorded [10,11]. When obtaining data, a sequence of digital images is captured from the infrared camera. Each digital image from the infrared camera represents an array of wall temperatures at varying $x$ and $y$ locations for a particular time.

According to Rogers et al. [10] and Ritchie et al. [11], associated measurement procedures involve the reconstruction of the heat flux from temperature traces, for each measurement location along the test surface. From these results, the heat flux is then plotted against temperature for the time period over which the heater mesh is operating. A linear relationship between the heat flux and wall temperature is expected and observed when using the linear convective heat transfer equation. From the resulting data, the slope has the magnitude of the heat transfer coefficient and adiabatic wall temperature is extrapolated for zero surface heat flux. Measured spatially resolved distributions of adiabatic surface temperature are then used to determine local values of the spatially resolved surface effectiveness, using the equation given by:

$$\eta = \frac{(T_{aw} - T_{s,M})}{(T_{t,c} - T_{s,M})}$$

(1)

The heat transfer coefficient is then defined using:

$$\dot{q}^* = h(T_{aw} - T_{surf})$$

(2)

Line-averaged adiabatic wall temperature and heat transfer coefficient values are determined by averaging the quantity of interest for a row of pixels at constant $x/d_e$ for a range of $y/d_e$. The associated $y$ range for this spanwise averaging is $8.0d_e$.

**Uncertainty Analysis Results**

Uncertainty estimates are based on 95 percent confidence levels and are determined using procedures described by Kline and McClintock [12] and by Moffat [13]. Uncertainty of thermocouple temperature readings is $\pm 0.15^\circ C$. Pressure uncertainty is $\pm 0.25$ Pa. Spatial and temperature resolutions achieved with the infrared imaging are about $0.2$ mm and $0.75^\circ C$, respectively. The experimental uncertainty of the blowing ratio is $\pm 4.0$ percent. The experimental
uncertainty of the coolant mass flow rate is also approximately ± 4.0 percent. This local coolant velocity value is a result of uncertainty in measured coolant pressure ratio (± 0.8 percent) and uncertainty in the discharge coefficient (± 3.4 percent). The uncertainty of adiabatic wall temperature is estimated to be ± 0.4°C. Main flow recovery temperature and coolant stagnation temperature uncertainty is estimated to be ± 0.25°C. Local surface effectiveness uncertainty is estimated to be ± 0.033 or about ± 8.2 percent for a nominal effectiveness value of 0.4. Experimental uncertainty magnitudes of line-averaged heat transfer coefficients are ± 8-10 percent, or approximately ± 4.5 W/m²K for a spanwise-averaged heat transfer coefficient value of 50 W/m²K.
Experimental Results

Experimental Conditions

Experimental conditions for the full-coverage film cooling and for the louver slot cooling are given in Table 1 and Table 2, respectively. Note that louver slot blowing ratios are provided for the spanwise-normal plane at the exit of the louver leap arrangement. Table 3 then provides louver slot cooling effective blowing ratios. These effective blowing ratios apply to the full-coverage holes, and represent values determined with the same overall film mass flow rates as are present in the full-coverage holes and louver slot together. Such blowing ratio effective values offer a basis of comparison of data obtained with full-coverage holes and louver slots together, with data obtained with full-coverage film cooling holes only.

**Table 1:** Full-coverage film cooling experimental conditions.

<table>
<thead>
<tr>
<th>Test</th>
<th>Blower Setting</th>
<th>V&lt;sub&gt;ms&lt;/sub&gt; [m/s]</th>
<th>Mass Flow Rate [kg/s]</th>
<th>Re&lt;sub&gt;ms&lt;/sub&gt;</th>
<th>Re&lt;sub&gt;ms, Avg&lt;/sub&gt;</th>
<th>Blower Setting</th>
<th>V&lt;sub&gt;cf&lt;/sub&gt; [m/s]</th>
<th>Mass Flow Rate [kg/s]</th>
<th>Re&lt;sub&gt;cf&lt;/sub&gt;</th>
<th>Blower Setting</th>
<th>V&lt;sub&gt;imp&lt;/sub&gt; [m/s]</th>
<th>Mass Flow Rate [kg/s]</th>
<th>Re&lt;sub&gt;imp&lt;/sub&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>33</td>
<td>5.84</td>
<td>0.718</td>
<td>112581</td>
<td>113943</td>
<td>14.0</td>
<td>0.94</td>
<td>0.042</td>
<td>9471</td>
<td>12</td>
<td>6.27</td>
<td>0.025</td>
<td>3420</td>
</tr>
<tr>
<td>2</td>
<td>33</td>
<td>5.81</td>
<td>0.714</td>
<td>111942</td>
<td>110575</td>
<td>16.9</td>
<td>0.95</td>
<td>0.043</td>
<td>9624</td>
<td>15</td>
<td>8.71</td>
<td>0.034</td>
<td>4758</td>
</tr>
<tr>
<td>3</td>
<td>33</td>
<td>5.74</td>
<td>0.706</td>
<td>110671</td>
<td>108208</td>
<td>19.9</td>
<td>0.95</td>
<td>0.043</td>
<td>9609</td>
<td>18</td>
<td>11.34</td>
<td>0.045</td>
<td>6207</td>
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<tr>
<td>4</td>
<td>33</td>
<td>5.63</td>
<td>0.692</td>
<td>108440</td>
<td>107058</td>
<td>23.2</td>
<td>0.97</td>
<td>0.044</td>
<td>9843</td>
<td>21</td>
<td>14.06</td>
<td>0.056</td>
<td>7715</td>
</tr>
</tbody>
</table>

**Table 2:** Louver slot cooling experimental conditions.

<table>
<thead>
<tr>
<th>V&lt;sub&gt;ef&lt;/sub&gt; [m/s]</th>
<th>Mach Number</th>
<th>Re&lt;sub&gt;ef&lt;/sub&gt;</th>
<th>Discharge Coefficient</th>
<th>Density Ratio</th>
<th>Velocity Ratio</th>
<th>Momentum Flux Ratio</th>
<th>Blowing Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>20.6</td>
<td>0.06</td>
<td>8591</td>
<td>0.70</td>
<td>1.04</td>
<td>3.53</td>
<td>13.03</td>
<td>3.69</td>
</tr>
<tr>
<td>23.8</td>
<td>0.07</td>
<td>9291</td>
<td>0.71</td>
<td>1.05</td>
<td>4.10</td>
<td>17.55</td>
<td>4.28</td>
</tr>
<tr>
<td>27.0</td>
<td>0.08</td>
<td>11254</td>
<td>0.72</td>
<td>1.05</td>
<td>4.70</td>
<td>23.10</td>
<td>4.92</td>
</tr>
<tr>
<td>30.6</td>
<td>0.09</td>
<td>12787</td>
<td>0.74</td>
<td>1.05</td>
<td>5.44</td>
<td>31.01</td>
<td>5.70</td>
</tr>
</tbody>
</table>

**Table 3:** Louver slot cooling effective experimental conditions.

<table>
<thead>
<tr>
<th>Velocity Ratio</th>
<th>Momentum Flux Ratio</th>
<th>Blowing Ratio</th>
<th>Density Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>VR 1.64</td>
<td>2.82</td>
<td>1.72</td>
<td>1.04</td>
</tr>
<tr>
<td>VR 1.90</td>
<td>3.80</td>
<td>1.99</td>
<td>1.05</td>
</tr>
<tr>
<td>VR 2.18</td>
<td>5.00</td>
<td>2.29</td>
<td>1.05</td>
</tr>
<tr>
<td>VR 2.53</td>
<td>6.71</td>
<td>2.66</td>
<td>1.05</td>
</tr>
</tbody>
</table>
Table 3: Louver slot cooling effective blowing ratios.

<table>
<thead>
<tr>
<th>Effusion BR</th>
<th>Louver Slot BR</th>
<th>Effective BR Without Louver Slot</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.69</td>
<td>1.72</td>
<td>5.21</td>
</tr>
<tr>
<td>4.28</td>
<td>1.99</td>
<td>6.05</td>
</tr>
<tr>
<td>4.92</td>
<td>2.29</td>
<td>6.95</td>
</tr>
<tr>
<td>5.70</td>
<td>2.66</td>
<td>8.06</td>
</tr>
</tbody>
</table>

The louver slot supply holes, shown in Figure 1, are sized relative to the effusion hole diameter to give dominate louver slot cooling, relative to the moderate thermal protection provided by the array of full-coverage film cooling holes. This is accomplished as the louver slot produces a thick, uniform layer of cooling air along the test surface downstream. The resulting thermal protection is accomplished as this cooling air acts as a thermal insulator and as a heat sink.

The experimental conditions illustrated by the data in Table 1 are obtained as the impingement jet Reynolds number is varied, with the crossflow Reynolds number approximately constant. The range of experimental conditions associated with these tests is relatively small. This is a result of the use of crossflow and an impingement jet array together to supply the coolant. The result is a range of experimental conditions, such as the ones which are given in Table 1. Outside of these conditions, one or the other coolant supply arrangements may act in a non-optimal manner. Such an occurrence is caused by coolant flow from one source which may overwhelm and reverse the coolant flow from the other source. For example, if impingement jet magnitudes are excessive relative to the crossflow, flow from impingement jets may move in the nominal upstream direction causing the crossflow to reverse. Alternatively, if crossflow magnitudes are excessive relative to the impingement jets, crossflow may enter impingement jet hole exits, resulting in reversal of the nominal impingement flow direction.

Test Section Velocity, Pressure, Blowing Ratio, and Discharge Coefficient Variations

Additional understanding of the impingement jet and crossflow coolant supply arrangement is provided by the data in Figure 2. Here, the normalized pressure drop is presented as it varies with blowing ratio for the impingement and crossflow passages, the crossflow and main flow passages, and the impingement and the crossflow passages. These data are provided for a main flow
Reynolds number $Re_{ms,avg}$ of 107000 to 114000. Note that the sum of the first two of these pressure drops is equal to the third pressure drop at each blowing ratio considered. Note that all three types of pressure drop increase with blowing ratio.

**Figure 2:** Test section passages pressure drop variations with initial blowing ratio for the louver and full-coverage film cooling configuration and $Re_{ms,avg}=107000$ to 114000.

The resulting variations of local main flow freestream velocity and spatially averaged effusion flow velocity with blowing ratio are then shown in Figure 3. Because the present test section inlet and outlet areas are the same, no significant pressure gradient is present in the main flow passage, and the freestream velocity is constant with streamwise position. Figure 3 shows that this freestream velocity is also invariant with blowing ratio. The associated main flow static pressure is also generally invariant with $x/de$ streamwise location and blowing ratio, as shown in Figure 4. Because the blowing ratio is determined using the spatially averaged effusion flow velocity, Figure 3 also shows that these quantities are linearly dependent and proportional to each other.
Figure 3: Local main flow freestream velocity and spatially-averaged effusion flow velocity variations with initial blowing ratio for the louver and full-coverage film cooling configuration and $Re_{ms,avg}=107000$ to 114000.

Figure 4: Local main flow static pressure variations with normalized streamwise location for the louver and full-coverage film cooling configuration and $Re_{ms,avg}=107000$ to 114000.

Figure 5 then presents local blowing ratio variations with normalized streamwise location for different initial blowing ratio values. For each of these initial values, local BR magnitudes are invariant with streamwise development as $x/de$ increases. This is, of course, partially a consequence of the zero streamwise pressure gradient which is present within the main flow passage. The discharge coefficient data, shown in Figure 6, are also provided for different impingement jet Reynolds numbers, as the crossflow Reynolds number is approximately constant.
Note that these data are provided for a wider range of experimental conditions than are tabulated within Table 1. Values increase from 0.70 to 0.77 as impingement jet Reynolds number increases from 3420 to approximately 17000.

**Figure 5**: Local blowing ratio variations with normalized streamwise location for the louver and full-coverage film cooling configuration and $Re_{ms,avg}=107000$ to 114000.

**Figure 6**: Discharge coefficient variations with impingement jet Reynolds number for the louver and full-coverage film cooling configuration and $Re_{ms,avg}=107000$ to 114000.

**Local Surface Adiabatic Film Cooling Effectiveness and Local Surface Heat Transfer Coefficient Variations**

Figure 7 shows local surface adiabatic film cooling effectiveness variations. Figure 8 then shows local surface heat transfer coefficient variations. Both types of data are provided for the
louver and full-coverage film cooling configuration, with an initial blowing ratio BR of 3.7 and a main flow Reynolds number $Re_{ms,avg}$ of 107000 to 114000. Both sets of data evidence influences of the horseshoe vortex which forms near the test surface around each coolant concentration as it exits a film cooling hole. The signature of such a three-dimensional vortex is locally augmented values of both effectiveness and heat transfer coefficient. Figures 7 and 8 show that the influences of the coolant jets are often present along the test surface in the vicinity of each film hole exit location, and generally persist for some distance downstream of each hole exit. Also, present downstream of many film cooling holes are locally increased magnitudes of both effectiveness and heat transfer coefficient, which are shaped in a v-shaped pattern directed in the downstream direction of individual hole exits.

**Figure 7:** Local surface adiabatic film cooling effectiveness variations for the louver and full-coverage film cooling configuration for a blowing ratio BR of 3.7 and $Re_{ms,avg} = 107000$ to 114000.

**Figure 8:** Local surface heat transfer coefficient variations for the louver and full-coverage film cooling configuration for a blowing ratio BR of 3.7 and $Re_{ms,avg} = 107000$ to 114000.
Spanwise-Averaged Surface Heat Transfer Coefficient Variations

Figures 9 to 13 show line-averaged heat transfer coefficient variations with streamwise development for different initial blowing ratios for \( \text{Re}_{\text{ms,avg}} = 107000 \) to 114000. The first of these figures provides data for the louver and full-coverage film cooling configuration. The remaining figures provide comparisons of results from the effusion cooling only configuration, and from the louver and full-coverage film cooling configuration.

Within Figure 9, magnitudes of line-averaged heat transfer coefficient are relatively low, considering the magnitudes of mainstream Reynolds number, crossflow Reynolds number, and impingement jet Reynolds number which are employed [11]. When considered at a particular \( x/de \) location, values are lowest when \( \text{BR} = 3.7 \), and then are approximately invariant with blowing ratio for \( \text{BR} \) values from 4.3 to 5.7. Note that coefficients show only small variations with streamwise development for each blowing ratio \( \text{BR} \) value. Such variations are associated with the relatively thick and relatively uniform layer of cooling air along the test surface which is generated by the louver slot, The thermal protection provided by such a layer is tied to its actions as a heat sink and as a thermal insulator. Resulting heat transfer coefficient trends in Figure 9 evidence reduced magnitudes of local turbulent thermal transport, which occurs, in part, because of reduced local advection speeds within the substantial wake which forms downstream of the louver leap configuration.

![Figure 9](image_url)

**Figure 9:** Line-averaged heat transfer coefficient variations with streamwise development, provided at different blowing ratios, for the louver and full-coverage film cooling configuration, \( \text{Re}_{\text{ms,avg}} = 107000 \) to 114000.
Comparisons of line-averaged heat transfer coefficients are provided in Figs. 10 and 11 for the effusion cooling only configuration, and for the louver and full-coverage film cooling configuration. These data are also provided for a main flow Reynolds number $Re_{ms,avg}$ of 107000 to 114000. The full-coverage film cooling data in these figures are provided for initial blowing ratio BR values of 5.3 and 7.3, respectively. The corresponding louver slot cooling effective blowing ratios (from Table 3) are 5.2 and 7.0, respectively. These effective values result for the combined louver and full-coverage film cooling configuration with effusion and louver slot blowing ratios of 3.7 and 1.7 for the Figure 10 results, and 4.9 and 2.3 for the Figure 11 results. With this comparison, the spanwise-averaged heat transfer coefficients show less variation with streamwise development location, relative to results obtained without a louver employed. In addition, spanwise-averaged heat transfer coefficients are consistently lower, especially for the downstream portions of the test plate, when the louver is utilized. Partially responsible is the blockage provided by the louver leap geometry, which gives a strong wake flow and relatively low velocity distributions just above the test surface. Also evident from the results given in Figs. 10 and 11 are smaller signatures in coefficient variations from the presence of individual rows of film cooling holes. The local coefficient gradients, which are a signature of a row of holes are pronounced for the full-coverage film data in these figures, with variations evident at $x/de$ in the vicinity of 15, 30, 45, 60, and 75. In contrast, variations at the locations with combined louver and full-coverage film cooling are much less evident.

**Figure 10:** Comparisons of line-averaged heat transfer coefficient values with streamwise development (at the same approximate effective blowing ratio) for the effusion cooling only configuration.
configuration, and for the louver and full-coverage film cooling configuration, \( \text{Re}_{ms, \text{avg}} = 107000 \) to 114000.

**Figure 11:** Comparisons of line-averaged heat transfer coefficient values with streamwise development (at the same approximate effective blowing ratio) for the effusion cooling only configuration, and for the louver and full-coverage film cooling configuration, \( \text{Re}_{ms, \text{avg}} = 107000 \) to 114000.

Similar quantitative and qualitative conclusions are reached in regard to the spanwise-averaged heat transfer coefficient data which are given in Figs. 12 and 13. Within the first of these figures, data are compared for impingement jet Reynolds numbers of 3420 and 3506. Within the second of these figures, data are compared for impingement jet Reynolds numbers of 7715 and 7418.

Results such as the ones presented in Figs. 10 and 11, as well as in Figs. 7 and 8, indicate that flow and surface thermal characteristics are very sensitive to the placement, location, and alignment of the louver leap device. Of particular importance are the extent and symmetry between the edges of the device and the side walls of the test section. Small geometric variations have been found to affect measured data in a significant manner, depending upon the experimental conditions which are considered.
Figure 12: Comparisons of line-averaged heat transfer coefficient values with streamwise development (at the same impingement jet Reynolds number) for the effusion cooling only configuration, and for the louver and full-coverage film cooling configuration, $Re_{ms,avg}=107000$ to 114000.

Figure 13: Comparisons of line-averaged heat transfer coefficient values with streamwise development (at the same impingement jet Reynolds number) for the effusion cooling only configuration, and for the louver and full-coverage film cooling configuration, $Re_{ms,avg}=107000$ to 114000.
Spanwise-Averaged Surface Adiabatic Film Cooling Effectiveness Variations

Figures 14 to 18 show line-averaged adiabatic film cooling effectiveness variations with streamwise development for different initial blowing ratios for $\text{Re}_{\text{ms,avg}}=107000$ to 114000. The first of these figures provides data for the louver and full-coverage film cooling configuration. The remaining figures provide comparisons of results from the effusion cooling only configuration, and from the louver and full-coverage film cooling configuration.

Figure 14 indicates that magnitudes of line-averaged adiabatic film cooling effectiveness vary by relatively small amounts, at a particular x/de streamwise location, as the initial blowing ratio is changed. When considered at a particular x/de location, values are then lowest when BR=3.7 and when BR=5.7. These effectiveness magnitudes vary only by small amounts with streamwise development for all BR values considered.

![Figure 14](image)

**Figure 14:** Line-averaged adiabatic film cooling effectiveness variations with streamwise development, provided at different blowing ratios, for the louver and full-coverage film cooling configuration, $\text{Re}_{\text{ms,avg}}=107000$ to 114000.

Figures 15 and 16 show comparisons of line-averaged adiabatic film cooling effectiveness for the effusion cooling only configuration, and for the louver and full-coverage film cooling configuration. These data are also provided for full-coverage film cooling initial blowing ratio BR values of 5.3 and 7.3, respectively. The corresponding louver slot cooling effective blowing ratios
(from Table 3) are again 5.2 and 7.0, respectively. Figures 17 and 18 show similar data comparisons based upon similar values of impingement jet Reynolds number. In all cases, the adiabatic film cooling effectiveness data for the effusion cooling only configuration show much larger variations with streamwise development as x/de increase, as well as much larger local variations near rows of film cooling holes at x/de of 15, 30, 45, 60, and 75. In particular, local maximum effectiveness values are evident at these locations. In contrast, significantly smaller line-averaged effectiveness changes with streamwise development, as well as near all near row hole locations, are present for the combined film cooling louver slot configuration. This is because the coolant from this slot is so plentiful that it is believed to surround and encapsulate volumes around the coolant trajectories which emerge from the full-coverage film cooling holes. The result is a relatively thick and relatively uniform layer of cooling air which is provided by the louver slot along the test surface downstream. Because of the nature of this layer, and the distributions of coolant which comprise this layer, contained within are significant heat sink and insulating characteristics.

**Figure 15**: Comparisons of line-averaged adiabatic film cooling effectiveness values with streamwise development (at the same approximate effective blowing ratio) for the effusion cooling only configuration, and for the louver and full-coverage film cooling configuration, $R_{ms,avg} = 107000$ to 114000.
Figure 16: Comparisons of line-averaged adiabatic film cooling effectiveness values with streamwise development (at the same approximate effective blowing ratio) for the effusion cooling only configuration, and for the louver and full-coverage film cooling configuration, $Re_{ms,\text{avg}}=107000$ to 114000.

Figure 17: Comparisons of line-averaged adiabatic film cooling effectiveness values with streamwise development (at the same impingement jet Reynolds number) for the effusion cooling only configuration, and for the louver and full-coverage film cooling configuration, $Re_{ms,\text{avg}}=107000$ to 114000.
Figure 18: Comparisons of line-averaged adiabatic film cooling effectiveness values with streamwise development (at the same impingement jet Reynolds number) for the effusion cooling only configuration, and for the louver and full-coverage film cooling configuration, $Re_{ms,avg}=107000$ to 114000.

The qualitative trends of the present line-averaged adiabatic film cooling data are consistent with louver slot results which are presented by Inanli et al. [7] and Kiyici et al. [8], even though the designs and lay-out of the slots and film cooling holes are different. Like the present data, results from these sources also show very little variation with streamwise development location, and with changing blowing ratio as streamwise location is maintained constant.
Summary and Conclusions

Considered within the present investigation are experimentally measured results wherein a louver slot is employed upstream of an array full coverage film cooling holes. The results presented are different from those from past investigations, because of the particular louver slot arrangement which is employed, including its location upstream of an array of full-coverage film cooling holes, and because of the unique coolant supply configurations. A combination arrangement is employed to supply the cooling air with both crossflow and an impingement jet array used together, such that the crossflow Reynolds number approximately constant, as the impingement jet Reynolds number is varied. As a result, the range of experimental conditions associated with these tests is relatively small because, outside of the experimental conditions which are employed, one or the other coolant supply arrangements may act to overwhelm and reverse the coolant flow from the other source.

The louver consists of a row of film cooling holes, contained within a specially designed device which concentrates, and directs the coolant from a slot, so that it then advects as a layer downstream along the test surface. This louver-supplied coolant is then supplemented by coolant which emerges from different rows of downstream film cooling holes. The same coolant supply passage is employed for the louver row of holes, as well as for the film cooling holes, such that different louver and film cooling mass flow rates and blowing ratios are set by different hole diameters for the two different types of cooling holes. Experimental results are provided for mainstream Reynolds numbers from 107000 to 114000, and full coverage blowing ratios from 3.69 to 5.70, with constant values as the flow develops in the streamwise direction along the test surface. Corresponding louver slot blowing ratios then range from 1.72 to 2.66. Along the mainstream side of the test plate, provided are measured distributions of local and spanwise-averaged adiabatic effectiveness and heat transfer coefficient values.

Both types of data show less variation with streamwise development location, relative to results obtained without a louver employed, when compared at the same approximate effective blowing ratio, mainstream Reynolds number, crossflow Reynolds number, and impingement jet Reynolds number. When compared at the same effective blowing ratio or the same impingement jet Reynolds number, spanwise-averaged heat transfer coefficients are consistently lower, especially for the downstream portions of the test plate, when the louver is utilized. With the same type of comparisons, the presence of the louver slot results in significantly higher magnitudes of spanwise-
averaged adiabatic film cooling effectiveness, particularly at and near the upstream portions of the test plate. With such characteristics, dramatic increases in thermal protection are provided by the presence of the louver slot, depending upon experimental condition and test surface location.
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References


