An Experimental Study of Compressibility Effects on the Film Cooling Effectiveness Using PSP and PIV Techniques

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An experimental study was performed to examine effects of compressibility on film cooling effectiveness for coolant flowing from cylindrical holes in a flat plate. The incoming flow Mach number was set at 0.07, 0.30, 0.50 and 0.70. The Pressure Sensitive Paint (PSP) technique, which is based on a mass transfer analogy, was used to map the adiabatic film cooling effectiveness distribution on the surface of interest. The high-resolution PSP measurement result showed that there was marginal enhancement of film cooling performance for higher Mach number flow comparing to the $Ma=0.07$ case. A high-resolution Particle Image Velocimetry (PIV) system was also used to conduct detailed flow field measurements to uncover the underlying physics of film cooling in the high speed flow, which showed similar flow field measurement results at $M=0.4$ for both $Ma=0.07$ and 0.30 cases, but slightly different ensemble-averaged velocity distribution was found for $M=1.25$ case.

Nomenclature

$M$ = Mass flux ratio of the coolant and mainstream flow, $\rho_c V_c / \rho_o V_o$

$I$ = Momentum flux ratio of the coolant and mainstream flow, $\rho_c V_c^2 / \rho_o V_o^2$

$DR$ = Density ratio of the coolant and mainstream, $\rho_c / \rho_o$

$D$ = Diameter of hole

$\eta_{av}$ = Adiabatic film cooling effectiveness at the wall

$T_{av}$ = Adiabatic wall temperature at the wall

$T_c$ = Temperature of the coolant

$T_o$ = Temperature of the mainstream flow

$\alpha$ = Thermal diffusion coefficient

$D_s$ = Concentration diffusion coefficient of species

$Le$ = Lewis number, $\alpha / D_s$

$(C_{o, main})$ = Oxygen concentration of mainstream

$(C_{o, mix})$ = Oxygen concentration of mainstream-coolant mixture at the wall

$(C_{o, coolant})$ = Oxygen concentration of cooling flow

$(p_{o, air})$ = Partial pressure of oxygen with air as the coolant

$(p_{o, mix})$ = Partial pressure of oxygen with $N_2$ or $CO_2$ as the coolant

$(p_{o, ref})$ = Partial pressure of oxygen at reference state

$I_b$ = PSP image intensity of background noise

$I_{ref}$ = PSP image intensity of the excitation light at the reference state without flow

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I. Introduction

In search of higher thermal efficiency and power output of gas turbine engines, the turbine inlet temperature is continuously being pushed upward, causing the protection of turbine blades from extreme temperatures to be of critical importance. Without providing sufficient cooling for the turbine blades, it would be impossible for them to survive in the extremely hot conditions. Film cooling technology has developed out of the necessity to protect turbine blades from damage mechanisms such as corrosion and melting, despite the severe temperatures in which they operate. By releasing a film of coolant gas on the blade surface, the solid components can be isolated from the hot gases in the turbine stage, hence increasing their lifetime. There is an inherent desire in optimizing film cooling systems to further increase the efficiency of gas turbines, which can result in significant economic savings. A thorough understanding of the mechanism of the film behavior is essential for such optimization.

Over the past 40 years, a number of investigations have been performed by researchers to understand the fundamental principle of film cooling and therefore improve the performance of gas turbine engines. Sinha et al. (1991) performed experimental studies of the effects of varying the density ratio and blowing ratio on film cooling effectiveness over a well-insulated flat plate with 35°-inclined cylindrical holes. It was found that the effect of variable density ratio on film cooling performance cannot be scaled with mass flux ratio, velocity ratio, or momentum flux ratio. Also, decreasing the density ratio and increasing the momentum flux would lead to a significant reduction of cooling effectiveness. Goldstein et al. demonstrated and quantified a remarkable enhancement of film cooling effectiveness by using fan-shaped holes: a widened cylindrical hole with 10° diffusion angle on each side. Flow visualization and film cooling measurements both showed that the secondary coolant jet would stay close to the wall as the mass flux ratio increases, whereas the jet from the cylindrical hole would detach from the surface at higher blowing rates. Baldauf et al. utilized infrared thermography to study the influence of blowing rate, density ratio, and other factors on film cooling effectiveness over a flat surface with a row of cylindrical holes. Their results showed that the laterally-averaged film cooling effectiveness increases with increasing blowing rate (before jet lift-off), and that the coolant gas of higher momentum ratios with low density ratio will enhance the interactions between mainstream and adjacent jets. Johnson et al. studied effects of the density ratio on film cooling effectiveness using both the PSP and PIV techniques. Their experimental results indicate that coolant jets with higher density ratios tend to remain attached to the surface as the blowing ratio increases, offering higher film cooling effectiveness. Bogard & Thole reviewed other researchers’ results and listed a number of physical factors that affect the film cooling effectiveness. They found that the established empirical correlations can only provide reasonable predictions for flat surfaces with low freestream turbulence for flow from cylindrical holes, while failing to characterize effects of curvature, freestream turbulence level, length scale, and hole shape. Also, Bunker made comprehensive comparisons of film cooling performance and detailed analysis of the merits and demerits of existing shaped holes.

Though numerous experimental and numerical studies have been performed to evaluate the effects of various parameters on film cooling effectiveness, most of those previous studies were conducted at relatively low Mach numbers (i.e., Ma<0.15) with the incoming airflow being incompressible. While the gas flow passing turbine blades is usually transonic, very little can be found in the literature that examines effects of compressibility of the incoming airflow on film cooling effectiveness for various coolant hole designs and arrangements. As for relatively-higher Mach number tests, Gritsch et al. measured the film cooling effectiveness over a range of external Mach numbers (0.3, 0.60 and 1.2) on three types of single holes (cylindrical hole, laterally-expanded hole, and fully-expanded hole). Their infrared measurement results show that there is little dependence of the Mach number on film cooling effectiveness for shaped holes or cylindrical holes, though for cylindrical holes there is a slight improvement at Ma=1.2. And Liess performed a similar study over a flat plate with positive pressure gradient that showed no effect of compressibility on film cooling performance. However, there are few studies on film cooling effectiveness with compressible flow over a row of cylindrical holes with zero pressure gradient, which is essential for understanding the fundamental principles of film coolant behavior for transonic flows.
In the present study, an experimental investigation was performed to examine the effects of compressibility on film cooling effectiveness of a row of five cylindrical holes over a flat plate with freestream Mach numbers of 0.07, 0.30, 0.50 and 0.70. The Pressure Sensitive Paint (PSP) technique was used to map detailed film cooling effectiveness distribution on the surface of interest based on a mass-flux analogy, in contrast to traditional temperature-based cooling effectiveness measurements. A high-resolution Particle Image Velocimetry (PIV) system was used to conduct detailed flow field measurements to quantify the dynamic mixing process between the coolant jet stream and the mainstream flows. Additionally, a comprehensive study was also conducted to examine the effects of mass flux ratio (M) and density ratio (DR) on film cooling efficiency with high speed incoming flow.

II. Experimental Setup and Test Model

A. PSP experimental rig and test model

The experimental studies were conducted in a transonic, open-circuit wind tunnel located at the Department of Aerospace Engineering of Iowa State University, shown in Figure 1(a). The tunnel has an optically-transparent test section with 63.5 mm×25.4 mm cross section and is driven by three pressurized tanks, each with about 8 m³ volume and 10 atmospheres of pressure at full capacity. A ceramic flow straightener with square 1 mm cells is installed upstream of the contraction to create a uniform low-turbulence incoming flow. In order to monitor the temperature and pressure variation during the experiment, six pressure taps and two K-type thermocouples are placed along the test rig to acquire the instantaneous temperature and pressure data inside test section and plenum. With the three tanks opened, Ma=0.70 flow can be achieved after a 20 s transient period and can be maintained stably for around 5 minutes, while each PSP experiment requires less than 3 minutes to perform. Temperature drops of less than 5°C were observed in the freestream flow. Additionally, the main flow was tripped by a 0.5 mm obstruction at the leading edge of the test section to generate a turbulent boundary layer of about 0.5 mm thickness at the leading edge of the coolant holes.

Figure 1. Schematic of the experimental rig for the present study.
Figure 1 (b) shows the experimental setup for PSP measurements. A 390 nm UV light source is projected on the PSP-coated test model. A 14-bit digital 2048×2048 pixel CCD camera fitted with a 610 nm long-pass filter is utilized to capture the emission light. The cylindrical-hole test model, shown in Figure 1 (c), is made of a hard plastic material and manufactured by a rapid prototyping 3D printer that builds the model layer by layer with a resolution of about 25 microns. A row of five coolant holes, each with a diameter $D=2$ mm, have a span-wise distance $3D$ on-center between adjacent holes. The holes have a 35° inclination angle to the breakout plane of the test plate and no additional compound angle. The test model is mounted on a plenum chamber and sealed by a thin latex runner gasket and silicon. The axial centerlines of the coolant injection holes intersect the upper surface of the test model at a distance $L=56$ mm from the leading edge of the model, which is where the boundary layer is tripped. Therefore, the boundary layer develops over a length of $28D$ before the mainstream flow encounters the coolant holes.

In the present study, the mainstream flow from the wind tunnel is provided to simulate the hot gas flow in typical turbine stage, while the coolant jet flow is supplied by a pressurized gas bottle. All experiments are conducted at room temperature (typically 293 K). Depending on the desired density ratio, various O₂-free gases (N₂, CO₂, SF₆, and others) can be utilized as coolant. For the present test, N₂, CO₂, and a mixture of SF₆ and CO₂ were chosen to investigate effects of density ratio differences on film cooling effectiveness, which is caused by temperature differences between the coolant and the hot gases in a real engine. The corresponding density ratios ($DR = \rho_1/\rho_\infty$) are 0.97, 1.53 and 2.00 for above three cases, respectively. The Mach number of the mainstream flow is held at four pre-set numbers, which are 0.07 (25 m/s), 0.30, 0.50 and 0.70; meanwhile, the mass flow rates of the coolants are adjusted to satisfy mass flux ratios ($M = \rho V_1/\rho_\infty V_\infty$) in the range of 0.40 and 1.25. All experimental cases are listed in Table 1.

Table 1. Experimental cases of present study.

<table>
<thead>
<tr>
<th>Mach number of Mainstream flow</th>
<th>$DR$</th>
<th>$M$</th>
<th>Mach number of coolant</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.07 (25m/s)</td>
<td>1.53</td>
<td>0.40, 0.60, 0.85, 1.00, 1.25</td>
<td>-</td>
</tr>
<tr>
<td>0.30</td>
<td>1.53</td>
<td>0.40, 0.85, 1.25</td>
<td>0.07–0.25</td>
</tr>
<tr>
<td>0.50</td>
<td>1.53</td>
<td>0.40, 0.85, 1.25</td>
<td>0.13–0.40</td>
</tr>
<tr>
<td>0.70</td>
<td>0.97</td>
<td>0.40, 0.85</td>
<td>0.29–0.61</td>
</tr>
<tr>
<td></td>
<td>1.53</td>
<td>0.40, 0.60, 0.85, 1.00, 1.25</td>
<td>0.18–0.57</td>
</tr>
<tr>
<td></td>
<td>2.00</td>
<td>0.40, 0.85</td>
<td>0.14–0.30</td>
</tr>
</tbody>
</table>

B. Flow field measurements utilizing the PIV technique

A high-resolution Particle Image Velocimetry (PIV) system was used to conduct detailed flow field measurements to quantify the dynamic interaction and mixing processes between the coolant and mainstream flows over the test plate. Figure 2 shows the schematic of the experimental setup for the PIV measurement. During the experiment, the mainstream airflow and the coolant jets were seeded with ~0.5 µm oil droplets generated by a Six-Jet Atomizer (TSI, Model 9306) and Atomizer Aerosol Generator (TOPAS, model ATM 210) respectively, which are suitable for high speed flow. For example, the time response of the produced seed is less than 0.60 µs for incoming flow with $Ma=0.70$ and can reach 99% of the freestream velocity after ~0.5 mm acceleration, estimated based on Chen et al.¹¹. A Nd:YAG laser (NewWave Gemini 200) was utilized to emit two pulses of 200 mJ light at 532 nm wavelength with a repetition rate of 10 Hz. Using a set of high-energy mirrors and optical lenses, the laser beam was shaped into a thin light sheet with a thickness in the measurement interest of less than 1 mm. In order to reveal the interaction between the coolant flow and the mainstream flow, the illuminating laser sheet was aligned along the mainstream flow direction, bisecting the coolant hole in the middle of the test plate. A 14-bit high-resolution digital camera (PCO2000) was used for the PIV acquisition with a field of view $7\times15\text{mm}^2$ and a magnification of 0.014 mm/pix. The camera and laser were both connected to a Digital Delay Generator (Berkeley Nucleonics, Model 565), which controlled the time interval of the lasers (3.2µs for $Ma=0.07$, 0.8 µs for $Ma=0.3$) and image acquisition process.

After PIV image acquisition, instantaneous PIV velocity vectors were obtained by using a frame-to-frame cross-correlation technique with an interrogation window size of 32×32 pixels. An effective overlap of 50% of the interrogation windows was employed in PIV image processing. After the instantaneous velocity vectors ($u_i, v_i$) are determined, the distributions of the ensemble-averaged quantities such as averaged velocity, normalized Reynolds
Shear Stress \((\tau = -u'v' / U'_\infty)\), and normalized turbulence kinetic energy \((N, TKE = \frac{1}{2}\left(v'^2 + \nu^2\right)/U'_\infty)\) were obtained from a sequence of 1000 realizations of instantaneous PIV measurements. The measurement uncertainty level for the velocity vectors is estimated to be within 2.0%, while the uncertainties for the measurements of ensemble-averaged flow quantities such as Reynolds stress and turbulent kinetic energy distributions are about 5.0%.

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\[ \eta = \frac{T_{aw} - T_{aw0}}{T_{aw} - T_c}, \]  

where \(T\) is the temperature of the main stream, \(T_{aw}\) is the adiabatic wall temperature of the surface under inspection, and \(T_c\) is the initial temperature of the coolant flow.

In general, experimental measurements of film cooling effectiveness can be classified into two categories: nonisothermal methods and the isothermal methods. Nonisothermal methods are based on conventional thermal measurement techniques, such as thermocouples\(^\text{12}\), liquid crystal thermometry \(^\text{13}\), infrared thermography \(^\text{14}\), and Temperature Sensitive Paint (TSP)\(^\text{15}\), to measure the surface temperatures and then compute the adiabatic film cooling effectiveness directly. Though it is a straightforward way to reveal useful information of film cooling effectiveness, there always exist concerns regarding the adiabatic wall temperature measurement due to effects of heat conduction within the test plate. However, isothermal methods—such as the Pressure Sensitive Paint (PSP) technique\(^\text{15}\)—can eliminate errors due to conduction successfully since they are based on a mass transfer analog to the temperature ratio in equation (1), freeing the measurement of any questions about heat conduction. The Lewis number \((L_e = \frac{\alpha}{D_s})\) is on the order of 1 at atmospheric pressure, which means the thermal boundary layer and concentration boundary layer are of the same order, then the differential equations involving heat and mass transfer can be approximated as analogous. For high Mach number flow, the mass transfer analogy is valid because the effect of pressure \((P)\) on the thermal diffusion coefficient \(\alpha = \frac{\lambda}{\rho c_p} = \left(\frac{\lambda}{c_p}\right)\frac{1}{P}\) and concentration coefficient \(D_s = D_{s,\text{an}} \left(\frac{T}{T_{\infty}}\right) \frac{P_{\text{an}}}{P}\) is cancelled in the Lewis number.

For PSP measurements, the surface of the test model is coated with an oxygen-sensitive paint layer (ISSI UniFIB). The paint consists of light-sensitive molecules called luminophores mixed into a gas-permeable polymeric binder. When the luminophores are exposed to UV light, they are excited and emit light at 650 nm wavelength. However, in the presence of diatomic oxygen gas the excited electrons return to their ground state via a radiationless
transition. This process is called oxygen quenching\textsuperscript{15}. Consequently, the intensity of the photoluminescence is inversely proportional to the concentration of oxygen, which means a reduced concentration of oxygen would result in higher intensity of emission light.

During the isothermal film cooling measurements, some coolant gas is chosen that contains no oxygen, whereas the mainstream of the wind tunnel consists of atmospheric air, which is approximately 21% oxygen. When the coolant gas is in contact with the PSP, the partial pressure of atmospheric oxygen is reduced, resulting in a subsequent reduction of oxygen quenching such that the photoluminescence is caused to increase. The mass transfer analogy states that the film cooling effectiveness can be expressed according to the distribution of oxygen concentration along the surface of interest. Replacing the temperature in equation (1) with the concentrations of oxygen, the film cooling effectiveness can be written as follows:

\[
\eta = \frac{(C_{\text{o}})_{\text{main}} - (C_{\text{o}})_{\text{mix}}}{(C_{\text{o}})_{\text{main}} - (C_{\text{o}})_{\text{coolant}}} = \frac{(C_{\text{o}})_{\text{main}} - (C_{\text{o}})_{\text{mix}}}{(C_{\text{o}})_{\text{main}} - (C_{\text{o}})_{\text{coolant}}},
\]

(2)

Because the PSP method provides a means of measuring the partial pressure of the oxygen rather than its concentration, Charbonnier et al (2009)\textsuperscript{16} derived an expression (3) that accounts for the effect of the difference in molecular mass between the coolant gas and the main stream flow. By relating the concentration of oxygen to the partial pressure of oxygen ratio, they found the following expression for the film cooling effectiveness:

\[
\eta = 1 - \frac{1}{\left[ \left( \frac{p_{\text{o}_2}}{p_{\text{o}_2}} \right)_{\text{wall}} - 1 \right] \frac{M_c}{M_{\text{air}}} + 1},
\]

(3)

where \( M_c \) and \( M_{\text{air}} \) are the molecular masses of the coolant gas and mainstream air, respectively.

The pressures listed in equation (3) can be calculated based on the normalized intensity of the emission light, which is directly related to the partial pressure of oxygen. As for the relationship between the normalized intensity ratio and normalized pressure ratio, it can be determined by calibrating the PSP. Through the calibration curve, the following relations between the normalized intensity ratios and partial pressure ratios can be found:

\[
\frac{I_{\text{ref}} - I_b}{I_{\text{air}} - I_b} = f \left( \frac{p_{\text{o}_2}}{p_{\text{o}_2}} \right)_{\text{ref}},
\]

(4)

\[
\frac{I_{\text{ref}} - I_b}{I_{\text{mix}} - I_b} = f \left( \frac{p_{\text{o}_2}}{p_{\text{o}_2}} \right)_{\text{ref}},
\]

(5)

where \( I_{\text{ref}} \) is the reference intensity, recorded at atmosphere pressure with flow off, \( I_{\text{air}} \) is the emission intensity with the wind tunnel on and pure air injected as the coolant, \( I_{\text{mix}} \) is the intensity with the flow on and oxygen-free coolant gas flow on. Finally, \( I_b \) is the background image, taken with the flow and illumination off to isolate effects of dark current and ambient illumination. Detailed information of how to conduct a PSP film cooling effectiveness experiment can be found in Johnson & Hu\textsuperscript{5} and Yang & Hu\textsuperscript{17}.

Figure 4 depicts the schematic of the experimental setup used in the present study to determine the PSP calibration curve. A test plate painted with PSP paint was mounted in an enclosed test cell. The paint chosen for the present study is ISSI Uni-FIB due to its low temperature sensitivity (~0.5%/C). A constant temperature thermal bath system was used to control the temperature of the copper plate mounted inside the test cell. During the calibration experiment, a vacuum pump was employed to depressurize the test cell and the pressure within the test cell was measured with a digital sensor array (DSA 3217 Module, Scanivalve Corp). A reference pressure of atmosphere pressure was used in the present study, which corresponds in the wind tunnel experiments to a condition of \( \eta=0 \). Since it is feasible that some effectiveness experiments might indicate a possible maximum of \( \eta=1.0 \) locally, corresponding to a true zero absolute \( \text{O}_2 \) pressure, it was deemed necessary to calibrate for this condition rather than rely upon calibration curve extrapolation. For this data point the cell was flushed with pure \( \text{CO}_2 \) until the PSP emission reached a steady value. Figure 5 shows the normalized PSP calibration curves at various temperature levels (14°C, 20°C, 26°C). It is interesting to note that, though the absolute value of the emission intensity of the
excited PSP molecules are temperature-dependent, the PSP calibration curves obtained at different temperatures collapse into a single curve when appropriately normalized. It is worth noting that all calibration points for each temperature condition were normalized by reference conditions of the same temperature, e.g. measurements made at 14°C were also normalized by a reference image taken at 14°C, and likewise for the other temperatures.

For the PSP image processing, in order to reduce the effects of camera noise on the measurements results, spatial averaging was performed in the study on square interrogation windows of 9×9 pixels with 50% overlap to ensure complete sampling of the measurement data. The acquired PSP images typically have a magnification of 0.055mm/pix, which results in a spatial resolution of 0.22 mm for the PSP measurement results. The measurement uncertainty for the centerline film cooling effectiveness given in the present study was estimated to be on the order of 3% for η = 0.5, 10% for η = 0.2, which is predicted based on Taylor Series Method 18–20. As for lateral averaged effectiveness, it is about 12% for η = 0.2.

Figure 4. Experimental setup for PSP calibration

Figure 5. The calibration curve for PSP measurements

III. Result and Discussion

In the present study, the characteristics of the boundary layer profile of the mainstream flow without coolant injection over the test plate were measured by the high resolution PIV system. Figure 6 shows the boundary layer velocity profiles taken from the time-averaged flow field distribution ahead of coolant hole. From the measurement results it can be seen that the boundary layer thickness (δ99≈1.2D) of 25 m/s flow is slightly larger than that of the Ma=0.30 case (δ99≈1.0D). The displacement thickness of the two cases are 0.13D and 0.11D, respectively.

Figure 6. The boundary layer profile of incoming flow ahead of injection hole at Ma=0.07 and 0.30
Though a number of experimental studies have been conducted in recent years using the PSP technique with mass transfer analogy to achieve quantitative film cooling effectiveness measurements, the PSP technique is still a fairly new technique in comparison to conventional temperature-based measurement methods. It is necessary to validate the reliability of the PSP measurements as an effective experimental tool for turbine blade film cooling studies. Therefore, a comparison was conducted in the present study to provide a quantitative comparison of film cooling effectiveness measurement results by using the PSP technique with mass transfer analogy against those derived directly from temperature-based measurements\cite{1,21,22} for the same film cooling design under the same or comparable test conditions, which is shown in Figure 7. Focusing on the centerline effectiveness [Figure 7 (a)], the measured results in the present study match previous temperature-based measurement results reasonably well from $x/D=2$ to $x/D=30$. As for the laterally-averaged effectiveness, though there is a slight difference in the range of $2<x/D<7$, the general trend is consist with published film cooling effectiveness measurement results. The discrepancy could be caused by differences in the experimental conditions of the present study with the thermal-based film cooling studies and the different spatial resolutions along the spanwise direction, especially since thermocouples provide a limited number of discrete data points.

![Figure 7](image_url)

**(a) Centerline effectiveness of cylindrical hole  (b) Lateral averaged effectiveness of cylindrical hole**

*Figure 7. Comparison of measured film cooling effectiveness of present study with previous studies at $M=0.60$ and $DR=1.53$*

Figure 8 shows a comparison of the film cooling effectiveness between the $Ma=0.07$ and $Ma=0.70$ cases at various mass flux ratios. On the whole, the spatial distribution [Fig. 8 (a) & (b)] of the low speed and high speed experiments are similar. The footprint length along the streamwise direction in these contours first increases with increasing $M$ then begins to decrease with increasing $M$. When $M$ is smaller than 0.60, the coolant jet streams are to remain attached in the near field, resulting in a comparably large region with uniform and high film cooling effectiveness. For mass flux ratio $M$ higher than 0.60, the film cooling contours become narrower and shorter, which indicates that the high momentum coolant stream partially separates from the plate.

More quantitative information of film cooling performance can be acquired through detailed comparisons of the centerline and laterally-averaged film cooling effectiveness between low speed and high speed tests shown in Figs. 8(c)–(f). It is worth mentioning that the lateral averaged effectiveness is performed over a spanwise domain of $-4.5<z/D<4.5$, or three full periods of the hole spacing pattern. At mass flux ratios of $M=0.40$ and 0.60, the centerline and lateral averaged film cooling effectiveness [Fig. 8(c) & (e)] of high speed cases almost collapse with the measurement results of low speed cases from $x/D=7$ to 20, except marginal enhancement of film cooling performance in the region near the coolant hole. This phenomenon is also observed in the spatial distribution of film cooling contours [Fig. 8(a) & (b)] that the shape of higher efficiency region (red and white color) for $Ma=0.70$ mainstream flow are much sharper and linear than that of the 25 m/s case, which are possibly caused by compressibility effects of incoming flow. It is well-known that gas density is highly dependent upon Mach number, and that flow at a higher Mach number results in a decreasing density. For a relatively-low mass flux ratio ($M<1.25$) in the high speed test, the Mach number of injection is relatively small compared to the mainstream flow (refer to Table 1), which leads to a slight increase of the density ratio ($DR$). Meanwhile, the momentum ratio ($I = \rho \frac{V^2}{\rho V^2}$) is closely related to $DR$, and can be rewritten as function of the mass flux ratio and density ratio ($I = M^2 / DR$). For a certain gas with constant $M$, the momentum ratio of streams will decrease as the increasing of
density ratio. Therefore, based on the above analysis, it is obvious that the coolant stream will remain closer to the surface due to the enhancement of the density ratio caused by compressibility effects. By increasing the mass flux ratio to 0.85, 1.00 and 1.25, the film cooling performance of high Mach number tests show marginally higher centerline and lateral effectiveness against corresponding low Mach number experiments. The spatial distributions for the $Ma=0.70$ cases show slightly wider coolant coverage than corresponding $Ma=0.07$ flow cases, which indicates that there is strong suction between adjacent coolant holes for the high Mach number cases.

![Spatial distribution of film cooling effectiveness at $Ma=0.07$](image1)

![Spatial distribution of film cooling effectiveness at $Ma=0.70$](image2)

![Centerline effectiveness at $M=0.40, 0.60$](image3)

![Centerline effectiveness at $M=0.85, 1.00, 1.25$](image4)

![Lateral averaged effectiveness at $M=0.40, 0.60$](image5)

![Lateral averaged effectiveness at $M=0.85, 1.00, 1.25$](image6)

Figure 8. Comparison of film cooling effectiveness between mainstream in $Ma=0.07$ and $Ma=0.70$ tests at various mass flux ratios.
Figure 9 shows measured film cooling effectiveness at two fixed blowing ratios in $Ma=0.70$, which can be applied to reveal the effects of the density ratio on high speed film cooling performance of the coolant jet streams more clearly and quantitatively. The three density ratios ($DR=0.97$, 1.53, 2.00) are achieved by using $N_2$, $CO_2$, and a mixture of $SF_6$ and $CO_2$, respectively, as the coolant gases. It can be seen that there is little difference in film cooling performance for coolant streams with relatively low blowing ratios (like $M=0.40$), indicating that the coolant flow is able to remain well-attached on the plate surface. However, the film cooling effectiveness was found to become dependent upon the density ratio when the mass flux ratio exceeds 0.85, for which it is observed that the higher density ratio leads to higher film cooling effectiveness. This effect is because the momentum flux ratio of light coolant gas is higher than that of dense gas at same mass flux ratio ($I = M^2 / DR$), which results in earlier detaching from the test plate.

![Figure 9](image)

(a) Centerline effectiveness ($M=0.40, 0.85$)  (b) Lateral averaged effectiveness ($M=0.40, 0.85$)

**Figure 9. Effects of density ratio on film cooling effectiveness at Ma=0.70**

In order to explore the effects of compressibility of high speed flow on film cooling effectiveness, a series of film cooling measurements were conducted under various incoming flow conditions in a range of Mach numbers, results shown below. Figure 10 shows the film cooling effectiveness as a function of Mach number for various mass flux ratios ($M=0.40$, 0.85, 1.25). By carefully inspecting the film cooling performance of all mass flux ratio cases, it can be seen that the centerline film cooling efficiencies of $Ma=0.30$, 0.50 and 0.70 cases in the near-hole region ($x/D<3$) are all modestly-higher or equal to that of the $Ma=0.07$ (25 m/s) cases at corresponding mass flux ratio, which can be explained by the analysis mentioned above (related to Figure 8). For relatively-low mass flux ratio at $M=0.40$, it is clear that the mainstream Mach number has a limited effect on the film cooling effectiveness for cylindrical holes, which demonstrates very consistent film cooling performance for all Mach number cases. As the mass flux ratio is increased to 0.85, similar phenomenon can be observed from the film cooling measurements results, except the laterally-averaged film cooling effectiveness, which is slightly enhanced. By inspecting the corresponding spatial distribution [Fig.10 (d)], the coolant coverage indeed appears slightly wider in the lateral direction than that of the reference $Ma=0.07$ case, so do the $Ma=1.25$ cases. This phenomenon is caused by a suction force between the coolant holes for high Mach number cases. Since the distance between the parallel coolant streams is only 6 mm ($3D$), the higher injection velocity of coolants results in stronger attraction force between adjacent streams. As for the injection velocity, it is determined by the mainstream Mach number ($Ma_e$) and mass flux ratio ($M$), and can be written as $Mach_{coolant} = \frac{M}{DR} Mach_{e}$, which indicates that higher incoming flow velocity and $M$ lead to higher injection velocity. Based on the above discussion, the laterally-averaged film cooling effectiveness of $M=0.85$ and 1.25 are enhanced as the incoming flow Mach number increases due to the generated suction force between coolant streams.

Though we know the film cooling performance of high Mach number flow is slightly better than that of low speed in high mass flux ratio range, the underlying flow physics pertinent to film cooling needs to be addressed. In order to gain more insight into the dynamic mixing process between the coolant jet stream and mainstream flows, a high resolution PIV system was used to conduct detailed flow field measurements over the flat plates near the
coolant jet region. Due to the limited capability of the oil atomizer used in the present study, sufficient seed density for PIV could only be produced for $Ma=0.30$ in the required field of view ($7 \times 15$ mm$^2$). Therefore, the highest Mach number in the present PIV study was set to 0.30. Figure 11 shows the comparison of PIV measurement results between $Ma=0.07$ and $Ma=0.30$ cases at $M=0.40$ and 1.25. The two instantaneous velocity field results, shown in Figs. 11(a) & 11(b), demonstrate intensive gas mixing between the coolant jet and mainstream flow at $M=1.25$. Concentrating on $M=0.40$, the ensemble-averaged velocity of both $Ma=0.07$ and 0.30 were found to be able to stay attached on the surface of the model, forming a film over the test plate in the region downstream of coolant hole, which is consistent with the PSP measurements described previously. However, the phenomenon completely changed for $M=1.25$ in that the ensemble-averaged velocity of coolant jets from both cases were found to lift off from the test plate and penetrate into the mainstream, resulting in poor cooling effectiveness. Though the outline of time-averaged velocity distribution of both tests are similar, there are some small differences between them. It is obvious that there is a low speed area (blue color) in Fig. 11(e) for $Ma=0.07$ case which indicates that the coolant stream totally separates from the surface at around $x/D=1.5$, while no such area is found for $Ma=0.3$ case. Additionally, the predicted thickness of coolant jet based on streamlines distribution in $x$-$z$ plane for $Ma=0.07$ ($\approx 0.4D$) is smaller than that of $Ma=0.3$ ($\approx 0.6D$), manifesting that the coolant gas of $Ma=0.3$ case remains slightly closer to the test plate comparing to low speed case which can be utilized to explain why the centerline film cooling effectiveness of higher Mach number cases marginal higher than $Ma=0.07$ case.

Figure 10. Film cooling effectiveness as function of Mach number for $M=0.40$, 0.85, 1.25
Figure 12 shows the detailed quantitative augmentation of film cooling efficiency as a function of Mach number, where \( Ma=0.07 \) is utilized as the reference, and the centerline and lateral averaged effectiveness are defined as 
\[
\eta_{CL} - \frac{\eta_{CL}(Ma=0.07)}{\eta_{CL}(Ma=0.07)} \quad \text{and} \quad \eta_{LA} - \frac{\eta_{LA}(Ma=0.07)}{\eta_{LA}(Ma=0.07)}
\]
respectively. When \( M=0.40 \) and 0.85, the film cooling effectiveness difference of centerline and lateral are within 10% and 15%, respectively, compared to the reference value for the \( Ma=0.30, 0.50, \) and 0.70 cases. For larger mass flux ratios, such as \( M=1.25 \), the effectiveness difference are marginally higher due to the suction force generated between coolant streams. This force cannot only increase the lateral coolant coverage, but also slightly pull down the coolant jets to plate surface, resulting in a modest increase in the centerline effectiveness. Additionally, the augmentation of the film cooling is positive related to the Mach number of mainstream flow, in that higher-speed incoming flow leads to improved film cooling performance.

Figure 11. Comparison of PIV measurement results between \( Ma=0.07 \) and \( Ma=0.30 \) cases at \( M=0.40 \) and 1.25
Figure 13 shows the comparison of area-averaged film cooling effectiveness (in range of x/D=0 to 20; z/D=-1.5D to 1.5D) as a function of Mach number at M=0.40, 0.85, 1.25, which can be applied to describe the cooling performance from a different perspective. The area-averaged film cooling effectiveness was computed by equation \[ \bar{\eta} = \frac{\int \eta \, dA}{A}, \] where A is the total interest area and \( \eta \) is the corresponding effectiveness at each location. Based on Fig. 13, it is obvious that the overall averaged film cooling efficiency is independent of the Mach number for M=0.40 while marginal enhancement is observed for M=0.85 and 1.25 as the Mach number increases. Therefore, when the mass flux ratio is not particular high, it is acceptable to predict the film cooling performance of high speed flow by conducting experiments in a relatively low-speed wind tunnel.

![Figure 13. The area-averaged film cooling effectiveness as a function of Mach number at M=0.40, 0.85, 1.25](image)

**IV. Conclusion**

In the present study, a series of experimental investigations were performed to measure the film cooling effectiveness of a row of five cylindrical holes over a flat plate with mainstream Mach number at 0.07, 0.30, 0.50 and 0.70. The effects of incoming flow compressibility, mass flux ratio, and density ratio on film cooling effectiveness under high speed flow were examined in great detailed based on the quantitatively measurements of PIV and PSP. The PSP experiment results reveal that when mass flux ratio is lower than 0.6, the film cooling
efficiency of $Ma=0.7$ cases increase as the $M$ increase, while it reverses dramatically as $M$ further increased. And there is little difference in film cooling performance for coolant streams of various density ratios with relatively low blowing ratios (like $M=0.40$), indicating that the coolant stream is able to remain well-attached on the plate. However, the film cooling effectiveness was found to become dependent upon the density ratio when the mass flux ratio exceeds 0.85, for which the higher density ratio leads to higher film cooling effectiveness. As for the incoming flow compressibility effect, the PSP measurement results show that the centerline film cooling effectiveness of $Ma=0.30$, 0.50 and 0.70 cases in the near hole region $(c/D<3)$ are modestly-higher or equal to that of the $Ma=0.07$ (25 m/s) case at corresponding mass flux ratio, which is caused by effects of compressibility of the incoming flow. But, the film cooling effectiveness remains approximately constant at relative low $M$ (such as $M=0.40$) beyond the near-hole region, which can be verified by the similar ensemble-averaged flow characteristics of PIV experiment for both the $Ma=0.07$ and $Ma=0.30$ cases. Because of the suction force between the coolant holes at $M=1.25$, there is marginal increasing in both centerline and laterally-averaged film cooling effectiveness as the mainstream Mach number increases. The detailed PIV measurement results of $M=1.25$ cases demonstrate that the coolant gas of $Ma=0.3$ case stays slightly closer to the test surface comparing to that of $Ma=0.07$ case, which is consisted with above PSP measurement results. All in all, it is reasonable to predict the film cooling performance, when mass flux ratio is not particular high, of high speed flow by conducting relatively low-speed wind tunnel testing.

References


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