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Measured overall effusion cooling effectiveness over a wide blowing ratio range using infrared imaging

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Abstract: Experimental investigations were performed on the overall cooling effectiveness η of a flat effusion wall over a wide range of blowing ratio ($M = 0.47 \sim 5.27$). The effusion wall had a staggered multi hole pattern typical of gas turbine combustor application, with a ratio of hole pitch to row spacing $P/S = 1:2$, a porosity $PS/d^2 = 72$, and an inclination angle $\alpha = 30^\circ$. The current paper documented distribution of the overall cooling effectiveness on the wall surface, based on infrared imaging of the 2-D surface temperature field. Experimental results indicate: (1) The overall η increases along with the streamwise distance for the wide range of M due to the superposition effect of the multi row film cooling. (2) The overall η substantially benefits from the multi hole inside convective cooling. The hole convective cooling not only complements the weakest film protection at initial rows but also helps mitigate the temperature gradient. (3) The overall η increases asymptotically with increasing M , unlike adiabatic η mostly published in the past, which decreased after M reached a specific level. The current work showcased the end cooling outcome jointly driven by the filming cooling mechanism and the hole inside convective cooling mechanism.

Key words: gas turbine; effusion cooling; overall cooling effectiveness; blowing ratio; infrared imaging
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Nomenclature

d	Diameter, mm	y	Spanwise location, mm
l	Turbulence length scale, mm	η	Cooling effectiveness
L	Length, mm	λ	Thermal conductivity, W/(m • K)
$\dot{m}$	Mass flow rate, kg/s	ρ	Density of air flow, kg/m ³
M	Blowing ratio	σ	Gradient
P	Spanwise hole pitch, mm	Subscript	
Re	Reynolds number	av	Average
S	Streamwise hole spacing, mm	c	Coolant
T	Temperature, K	g	Mainstream gas
T_u	Turbulence intensity	span	Spanwise
u	Velocity, m/s	stre	Streamwise
x	Streamwise location, mm	w	Wall surface

Introduction

In future gas turbine engines the durability

of combustor liners is one of the primary concerns for the high temperature rise combustor, especially with the trend of increasing coolant

temperature and reducing coolant flow rate. Advanced liner cooling techniques must be developed for maximum utilization of the coolant cooling potential.

Effusion cooling is one of the most effective and economical cooling schemes for gas turbine combustors, as illustrated in Fig. 1. It can significantly enhance convective heat transfer (cooling) inside a large number of inclined holes before the coolant forms a cooling film on the hot-side surface, isolating the liner wall from the hot gas. Effusion cooling has been widely studied and it is in application^[1-2]. Nonetheless, fundamental research on the flow and cooling characteristics of this cooling scheme is still of great interest, to enhance understanding and design.

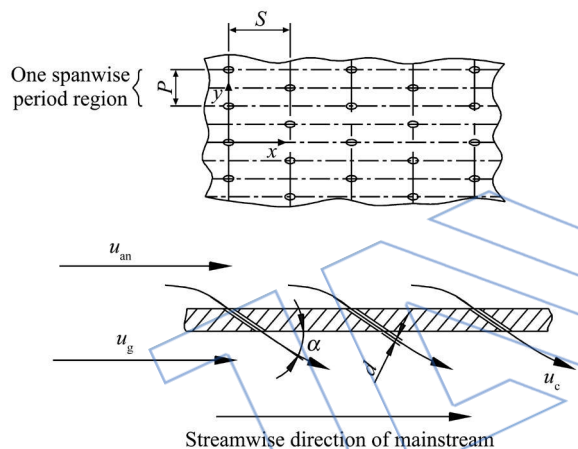


Fig. 1 Effusion cooling scheme

The overall cooling effectiveness is attributed to the adiabatic film cooling effectiveness, heat transfer on both cold-side and hot-side wall surfaces, and the convection cooling effect inside the inclined holes. Dagaut and Cathonnet et al.^[3-6] conducted a series of studies on the adiabatic cooling effectiveness and heat transfer of the effusion cooling scheme, examining the effect of multi-hole pattern, hole inclination angle, hole deflection angle and blowing ratio. They found that smaller pitch helped on spanwise cooling uniformity, the staggered multi-hole pattern had better cooling protection than the inline pattern, and blowing ratio had smaller influence on cooling effectiveness of streamwise injection than reverse injection. The cooling

and heat transfer characteristics were presented with data interpolation, instead of continuous 2-D mapping of the wall surface, as the measurement was conducted at discrete spatial points, either using a gas analyzer to measure the concentration of CO_2 (a heat-mass analogy method) or a large number of thermocouples to detect the wall surface temperature (a direct thermal method). As such, the presented cooling or heat transfer results were not so certain in the interpolated region, particularly near the holes where the highest gradient existed.

Brankovic et al.^[7] obtained the film cooling effectiveness and heat transfer coefficient for a full coverage film cooling wall with an inclination angle of 20° and hole spacing of $16d$ to $10d$, using the transient liquid crystal technique. Expensive liquid crystal paint was applied on the measured surface and complicated liquid crystal color analysis was conducted after the experiment. They found that the plate with a smaller hole spacing offered higher cooling effectiveness given the same blowing ratio. Increasing blowing ratio led to lower cooling effect due to film liftoff from the surface. The experimental results indicated that cooling film coverage was fully developed at about $80d$ from the first row. Probably due to high expense and inconvenience of the experiment, Ling's research only dealt with the first several rows and farther downstream rows, with a few rows between skipped. A steady-state wide band liquid crystal technique was also used to study film cooling effectiveness and heat transfer by Martinez Botas^[8]. This study showed that film cooling effectiveness increased as the film remained attached to the wall surface, with the optimum blowing ratio close to 0.5.

The infrared (IR) imaging technique has been widely used for non-intrusive surface temperature measurements in film cooling studies^[9-12]. Compared to the liquid crystal technique, advantage of IR technique was evident, as a wider temperature range could be dealt with and more detailed 2-D temperature field information

tion could be obtained with higher accuracy.

Gustafsson et al.^[9] investigated the temperature distribution on effusion-cooled plates with different temperature ratios and velocity ratios between the hot gas and coolant, different hole spacing, inclination angle, and thermal conductivity of the test plates. For a given amount of cooling air, the wall temperature decreased with reducing the injection angle due to more attached jets. Higher thermal conductivity of the plates led to decreased surface temperature in front of the first row of holes. Almost linear relationship was found between the temperature ratio and the surface temperature at a large velocity ratio. Harrington et al.^[10] performed an experimental and computational investigation on the adiabatic effectiveness of large-scaled full coverage film cooling plates with normal holes, considering the effect of the mainstream turbulence. The separation of the coolant jet from the surface was found by both measurement and prediction at blowing ratios higher than 0.65. Averaged cooling effectiveness decreased when the mainstream turbulence was elevated. With 7.14*d* hole spacing, cooling coverage was fully developed at the streamwise location of 30*d* for the blowing ratio of 0.25 and at 50*d* for the blowing ratio of 0.65. Both film cooling effectiveness and heat transfer coefficient downstream of a single hole were obtained from a test by means of the transient IR thermography technique^[11].

Coulthard et al.^[12] studied the effect of jet pulsing on the cooling effectiveness, flow temperature and heat transfer of the film cooling with a row of 35° inclined holes. In the steady blowing case, when the blowing ratio was lower than 0.5 the film cooling jet was attached to the wall surface, resulting in high cooling effectiveness. Increasing blowing ratio reduced the cooling effect due to the film liftoff. Low convective heat transfer coefficient of the film corresponded to high cooling effectiveness, and the overall heat flux through the wall was dependent more on film cooling effectiveness than convective heat transfer. Two kinds of vortices, horseshoe vor-

tices wrapping around the film cooling jets and kidney vortices associated with the jets, had a large influence on the cooling effectiveness and heat transfer downstream of the cooling holes. Similar findings are seen from computational fluid dynamics (CFD) studies of film jets^[13-14].

The influence of the blowing ratio on the film cooling was also studied through the velocity profile measurements for a full coverage effusion cooling wall^[15]. In this 4.9*d* hole spacing full coverage effusion cooling scheme, the velocity profile was fully developed at the streamwise location of 70*d*, while the fully developed thermal state did not occur at the same location. It was found that the streamwise velocity and turbulence profiles were on scale with the blowing ratio for all locations and blowing ratios. The jet separation downstream of the holes was shown in the velocity profile, with counter-rotating vortices entraining hot mainstream flow toward the wall surface, leading to low cooling effectiveness. The cooling jet penetration into the hot mainstream was also studied by another CFD effort^[16]. Using a tracer gas in the coolant, this study nicely displayed the film jet liftoff at high blowing ratios, which corresponded to low adiabatic cooling effectiveness.

Among the influencing factors on the adiabatic cooling effectiveness for effusion walls, the blowing ratio, defined as $M = \rho_{uc} u_c / \rho_g u_g$, was found to have the greatest effect. While the adiabatic cooling effectiveness increases with increasing blowing ratio at low blowing ratios, the trend turns to the opposite after blowing ratio is elevated to some level, due to the coolant lift off from the wall surface. Such effects of blowing ratio on cooling effectiveness have been widely seen in previous research^[7-8, 10, 12, 15-16].

It is noted that prior research dealt with various ranges of blowing ratio with different multi-hole arrangements and geometries. It is hard to quantify the effect of blowing ratio though the qualitative trend is evident. To complement the lack of research on this topic, the current work extended measurements to a wide

range of blowing ratio ($M = 0.47 \sim 5.27$) on the same multi-hole effusion configuration such as a consistent and complete documentation on the effect of blowing ratio was made available. Other than adiabatic film cooling effectiveness, which was mostly published in the past, the current paper presents the overall cooling effectiveness of the effusion wall, providing an additional perspective to inspect the effusion cooling characteristics. Considering its merit as reviewed above, IR imaging was adopted to measure the wall surface temperature, allowing for a complete mapping of the 2-D temperature field and hence the overall cooling effectiveness distribution.

1 Experimental methods and conditions

1.1 Parameters and equations

The overall cooling effectiveness is defined as a normalized temperature ratio, involving temperature of the hot mainstream, coolant air, and wall surface, as shown in Eq. 1^[11,13]. It addresses the total cooling effect that the coolant provides to keep the wall surface from heating up by the hot mainstream. Three temperatures needed in Eq. 1 can be directly measured by the heat transfer experiment.

$$\eta = \frac{T_g - T_w}{T_g - T_c} \quad (1)$$

Both maximum temperature and temperature gradient of the liner wall affect the durability of a gas turbine combustor. Therefore, not only the value of cooling effectiveness but also its gradient of a cooling scheme is of great interest. The streamwise gradient and spanwise gradient of cooling effectiveness are defined as Eq. 2 and Eq. 3 respectively.

$$\sigma_{\text{stre}}(\eta) = \frac{\Delta\eta}{\Delta(x/d)} \quad (2)$$

$$\sigma_{\text{span}}(\eta) = \frac{\Delta\eta}{\Delta(y/d)} \quad (3)$$

To view the overall cooling variation trend in the streamwise direction, the spanwise aver-

aged cooling effectiveness at each streamwise location (x/d) is calculated as Eq. 4:

$$\eta_{\text{span av}} = \sum_{j=1}^{N_{\text{span}}} \eta_j / N_{\text{span}} \quad (4)$$

Where N_{span} is the number of data points in the spanwise direction.

By further averaging in the streamwise direction, Eq. 5 calculates the total averaged cooling effectiveness of the effusion cooling wall.

$$\eta_{\text{av}} = \sum_{i=1}^{N_{\text{stre}}} \eta_{\text{span av}, i} / N_{\text{stre}} \quad (5)$$

1.2 Experimental facility and techniques

In the present experiments, temperature of the hot mainstream and coolant air was measured using thermocouple probes. Temperature distribution on the wall surface was measured with an IR imaging camera in steady state and calibrated by reference thermocouples embedded in the test effusion plate. The measured surface was painted with a high emissivity matt black paint, with thickness less than $50\mu\text{m}$ to ensure uniform heat emission. All of the thermocouples used were of type K.

Fig. 2 shows a schematic of the experimental setup. The size of the test section was $220\text{ mm} \times 80\text{ mm} \times 100\text{ mm}$ (length $\times$ width $\times$ height), and the opposite side to the test surface had a $200\text{ mm} \times 30\text{ mm}$ (length $\times$ width) rectangular window opening for the infrared imaging measurement. As no glass was used, it was entirely transparent to infrared radiation. The air flow leakage through the window opening was checked by silk threads and found to be very weak compared to the mainstream. CFD simulation was also conducted to inspect the effect of

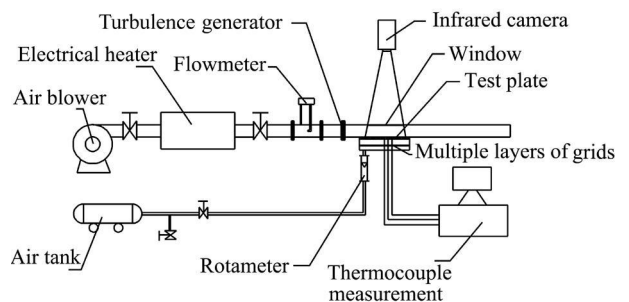


Fig. 2 Schematic of the experimental setup

the leakage on the mainstream. It was found that less than 5% of the total flow fled through the window opening and the mainstream flow field was hardly affected by the leakage flow.

The mainstream air was supplied by an air blower at atmospheric pressure, and heated to 373 K by an electrical heater, with Reynolds number being 57 000. In order to simulate actual combustor turbulence conditions^[10], a grid turbulence generator was placed 240 mm upstream of the test plate, which led to the mainstream turbulence intensity $T_u = 14.7\%$ and length scale $l = 9\text{ mm}$ ^[17]. An orifice flow meter was used to measure the mainstream flow rate.

Coolant air was provided by a high pressure air tank, led into a chamber assembled on the test plate, and was kept at atmospheric temperature. To eliminate the impingement effect of the coolant air at the chamber intake, multiple layers of grids were placed in the chamber. A thermocouple probe was placed at the entrance of the chamber to measure the coolant temperature. The coolant flow rate was measured with a rotameter.

A ThermoCAM P60 infrared camera was used to measure the effusion wall surface temperature, with the thermal sensitivity of $0.08\text{ }^\circ\text{C}$ at $30\text{ }^\circ\text{C}$, accuracy of $\pm 2\text{ }^\circ\text{C}$, $7.5\text{ }\mu\text{m}$ to $13\text{ }\mu\text{m}$ spectral range and 320×240 pixels image output. Given the 1.3 mrad spatial resolution of the infrared camera and the measurement distance, one pixel in the infrared image corresponded to $0.57\text{ mm}\times 0.57\text{ mm}$ approximately.

1.3 Effusion cooling test plate

On the test plate, multi-holes with a 30° inclination angle were arranged in a staggered pattern ($P/d = 6$ and $S/d = 12$). The length to diameter ratio L/d of the holes was 7.5. The plate was made of Bakelite composite material with low heat conductivity ($\lambda = 0.453\text{ W}/(\text{m}\cdot\text{K})$). Low heat conduction could help the cooling effectiveness contours to show detailed cooling characteristics of the full coverage film cooling better^[2,9]. The plate size was $220\text{ mm}\times 80\text{ mm}$,

with a thickness of 6 mm.

1.4 Experimental conditions

The experimental conditions are shown in Table 1. The mainstream to coolant temperature ratio (T_g/T_c) was kept at $1.33\pm 2.7\%$. It did not represent the real temperature ratio in gas turbine combustors where T_g/T_c is usually up to 2.5. The experimental results by Gustafsson et al.^[9] suggested that T_w/T_c increased linearly with T_g/T_c , so the relationship between T_w/T_c and T_g/T_c could be represented by Eq. 6, with η being a constant to maintain the linear relationship of the two ratios. Eq. 6 can be rewritten as Eq. 1, which means by looking at Eq. 1, the cooling effectiveness (η) is independent of the hot mainstream temperature within a certain range ($T_g/T_c < 1.9$ ^[9]). Or at least one can conclude from findings of the Reference[9] that is hardly responsive to T_g/T_c .

$$\frac{T_w}{T_c} = (1 - \eta)\left(\frac{T_g}{T_c} - 1\right) + 1 \tag{6}$$

Table 1 Experimental conditions

Parameters	Values
Atmosphere pressure/ Pa	101 945
Atmosphere temperature/ K	289. 65
Mainstream velocity u_g / (m/ s)	21
Mainstream Reynolds number Re_g	80000
Mainstream temperature T_g / K	379. 15~ 385. 15
Coolant temperature T_c / K	281. 15~ 288. 75
Temperature ratio T_g/T_c	1. 33 $\pm$ 0.036
Effusion jet Reynolds number Re_c	830~ 9400
Blowing ratio M	0. 47~ 5. 27

1.5 Experimental data uncertainty

The infrared images were adjusted according to the experimental parameters, including wall surface emissivity, atmospheric temperature and humidity, and distance between the camera and wall surface, in the software named ThermoCAM researcher provided by FLIR (forward looking infrared) systems. Then the surface temperature data in the infrared images was re-

duced to the local and averaged overall cooling effectiveness. In order to obtain accurate surface temperature, the emissivity of the test wall surface was calibrated by reference temperature readings from three thermocouples embedded in a flat plate calibration specimen without cooling.

Fig. 3 shows the comparison between the infrared temperature measurement (with the emissivity set to be an optimum value of 0.86) and the thermocouple readings. 91% out of all data points fall in the variation band of $\pm 5\text{K}$. So the emissivity of the measured surface in the experiments was set to 0.86. Based on the experimental conditions and the accuracy of the instruments, the experimental data of the overall cooling effectiveness was analyzed to have an uncertainty of $\pm 11\%$ in the present research using the standard error propagation analysis.

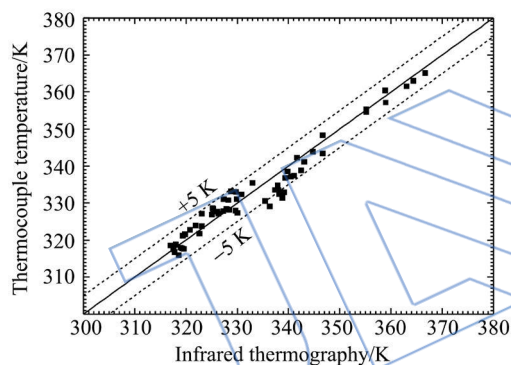


Fig. 3 Comparison of infrared thermography to thermocouple temperature

2 Experimental results and discussion

There were eight columns of holes (making a spanwise coverage of $7P$) on the test plate. The measurement area was chosen to cover the middle region between the third and forth column.

2.1 Overall cooling effectiveness distribution

The measured overall cooling effectiveness (η) contours at different blowing ratios are shown in Fig. 4. The beginning of the streamwise location ($x/d = 0$) is set at the exit center of the first row of holes. It should be pointed out that the η distribution on the entire measured

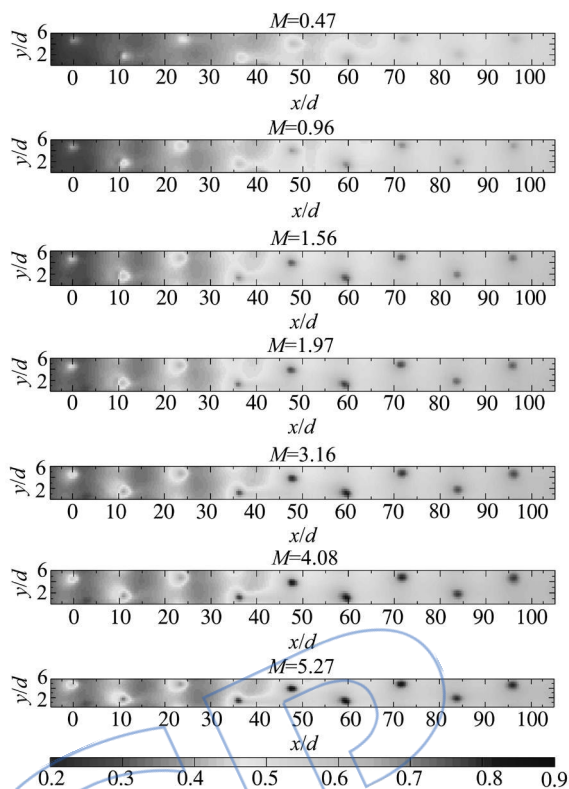


Fig. 4 Contours of overall cooling effectiveness (η)

rectangular plane is shown, including the wall surface (the real object in question) and the elliptical hole exit area. The η value shown in the hole exit area is based on the hole inside surface temperature captured by the IR camera. As the hole inside surface temperature is higher than the coolant temperature, the corresponding η value, shown as green or blue “cold spots” in the contours, is smaller than unity. Some previous research only published information of the wall surface, leaving the hole exit area as black out without showing any data. The current paper chose to include the hole exit area for a complete presentation of the entire 2-D temperature field “seen” by the IR camera. The information of the hole exit area can be used as reference when inspecting the subjected wall surface.

Looking at all seven η contours lined up in Fig. 4, one can find some general trends. Along the streamwise distance (x/d), η gradually increased. This is due to the well known superposition effect of the multi-row film cooling, with upstream coolant contributing to the downstream area. Increasing blowing ratio (M) from

0.47 to 5.27, η simply increases. After M reached the medium level, the increase rate of η with further increasing M becomes mitigated. In the local wall surface area surrounding the hole exit, η has a circumferentially symmetrical distribution, unlike the adiabatic cooling effectiveness, whose distribution is usually streamwise stretched as it is purely driven by the flow mechanism of the coolant jet and its interaction with the main flow. In addition to the flow mechanism, the current overall cooling effectiveness distribution is also driven by the “cold source” from the holes, another mechanism composed of heat convection (cooling effect) inside the holes and heat conduction inside the wall (cooling spreading). Previous research found that the heat convection within the effusion holes increased with the coolant jet Reynolds number^[4], equivalent to the increase of M in the current case. From that standpoint, increasing M is always beneficial to the cooling outcome, and it can compensate the possible drop of adiabatic film cooling effectiveness at high blowing ratios.

Based on data shown in Fig. 4, the streamwise and spanwise gradient of η , σ_{stre} and σ_{span} , were derived and the contours are shown in Fig. 5 and Fig. 6 respectively for all the blowing ratios. From these contours, it can be seen that the gradient of overall cooling effectiveness (or can be taken as the wall temperature gradient) is repeatedly distributed from one row to another on the entire rectangular measured wall surface, with the middle rows (row 3~7) have a little higher gradient. In a row unit of area, the gradient is rather evenly distributed for most of the wall surface except for the near-hole area, where the gradient is the highest. This result indicates that for the combustor liner protection, the near-hole area would be the weakest spot from the temperature gradient standpoint though the temperature there could not be the highest. This could be primarily due to the “cold source” of inside-hole convective cooling.

In order to better view the η variation at different locations and with different blowing ratio

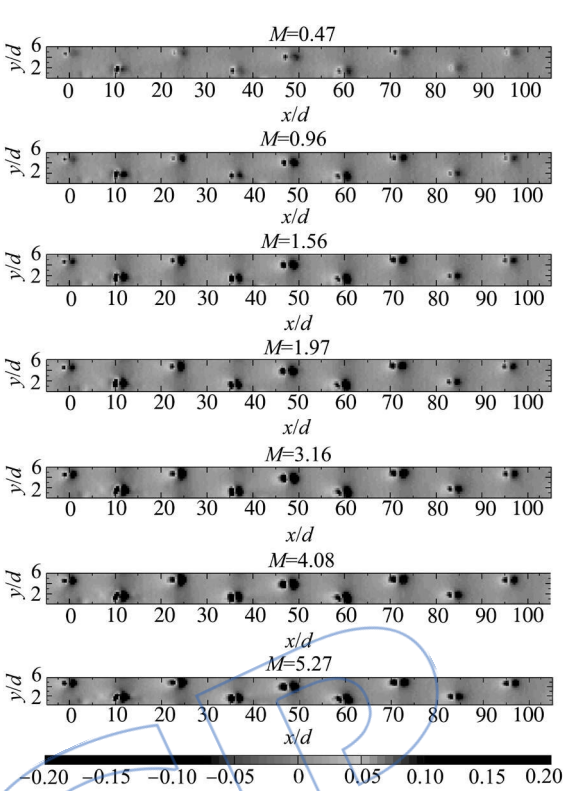


Fig. 5 Streamwise gradient of overall cooling effectiveness (σ_{stre})

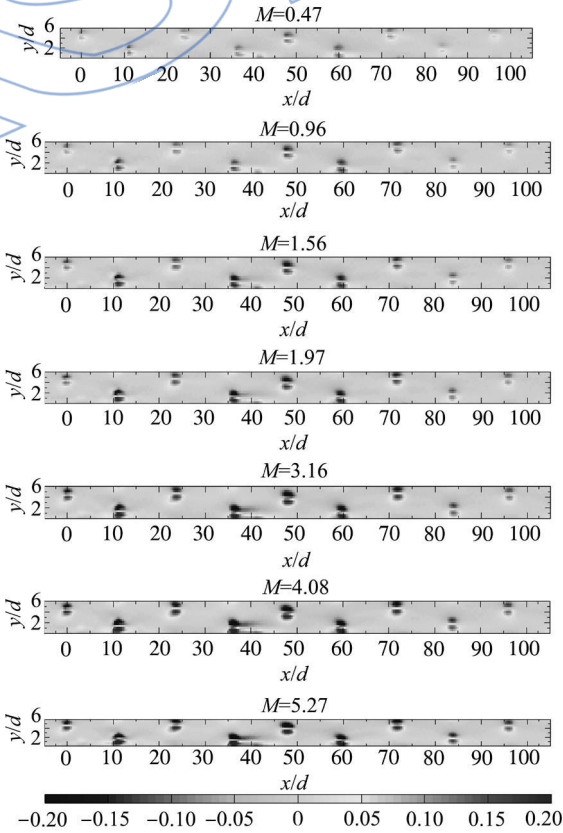


Fig. 6 Spanwise gradient of overall cooling effectiveness (σ_{span})

os, data of Fig. 4 along three streamwise lines was pulled out and plotted in Fig. 7, at three spanwise locations: $y/d=0$, 1.5 and 3.

It is seen again the general trend of η gradual increase along the streamwise direction, the first cooling mechanism of the effusion cooling, the superposition effect of the multi-row film cooling.

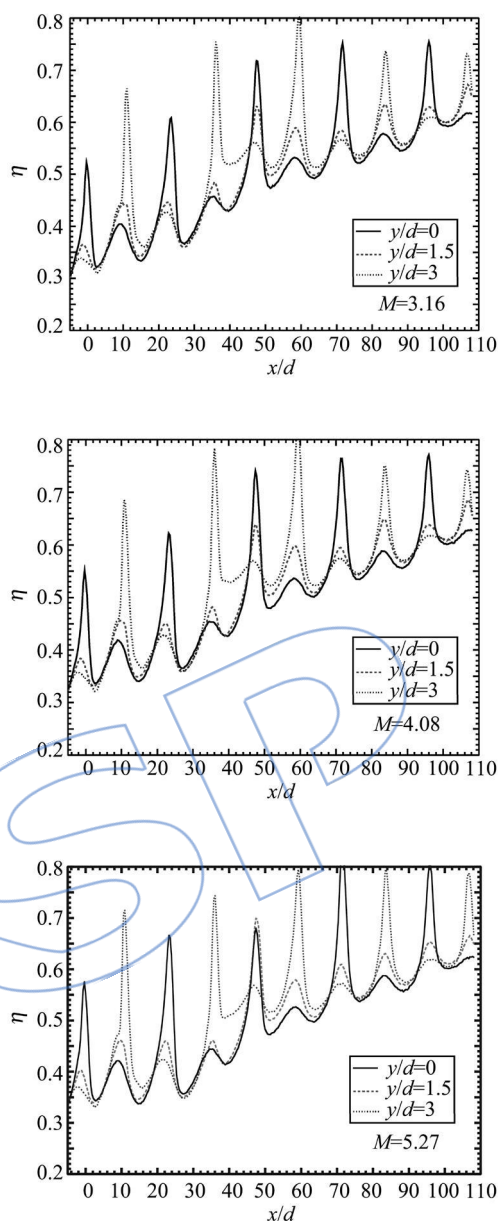
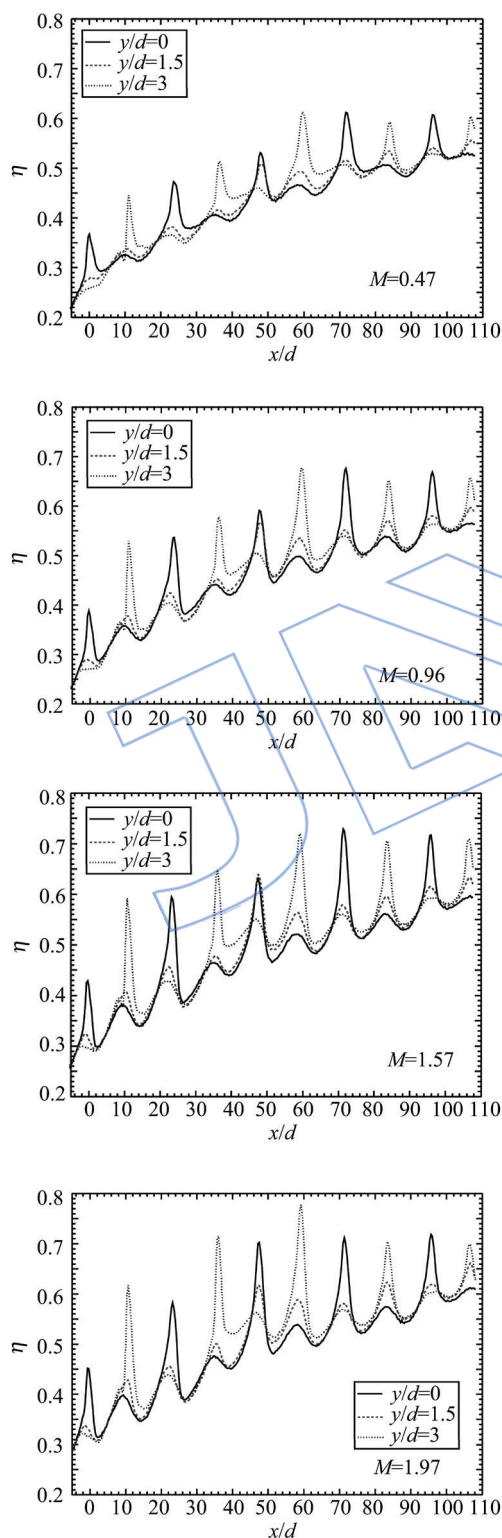


Fig. 7 Overall cooling effectiveness along three streamwise lines $y/d=0$, 1.5 and 3

The periodic peaks for $y/d=0$ and 3 correspond to the “cold spots” of the holes in Fig. 4. After cutting the peak with a $2d$ -long segment (the hole exit elliptical diameter), the truncated plateau of η value clearly indicates the high cooling effectiveness around the hole edge area, a strong inside-hole convective cooling effect spreading by the wall heat conduction. And due to this, on the line of $y/d=1.5$ small η peaks are evident, on the lines of $y/d=0$ and 3 smaller η peaks follow the big peaks before reaching the next row of big peaks. The results showcased

the second cooling mechanism of the effusion cooling, the hole inside convective cooling.

For reference interest, the current overall η results are compared to adiabatic η , as shown in Fig. 8. It should be pointed out that the referred adiabatic $\eta^{[6]}$ of was based on a closer arranged hole pattern thus its higher η values would not be a surprise. The intent of Fig. 8 is to make a qualitative comparison and it is fulfilled in that: (1) After the first row of holes, overall η is over 0.3 due to the hole inside convective cooling effect, while adiabatic η is much lower as cooling film is not superimposed there. The non-zero cooling effectiveness upstream of the first row of holes is due to the heat conduction in the plate, and was reported in previous studies on non-adiabatic test plates^[9, 15]. (2) For adiabatic η , there was neither “small peak” at $y/d=0.25P/d$, nor “smaller peaks” between two big peaks at $y/d=0$ and $0.5P/d$ as seen for overall η , as there was no hole inside cooling source in this case. (3) Overall η is more evenly distributed than adiabatic η in the 2-D wall surface, either streamwise or spanwise, due to the contribution of the hole inside convective cooling, a favored characteristic for the end application.

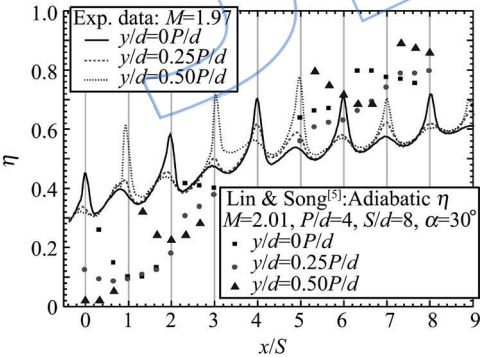


Fig. 8 Comparison of overall cooling effectiveness and adiabatic cooling effectiveness

2. 2 Averaged overall cooling effectiveness

The above discussion focused on the local η and it reflected the effect of blowing ratio M . To better view the effect of M , spanwise averaged $\eta_{span-av}$ at different M are plotted together, as shown in Fig. 9. It can be seen that $\eta_{span-av}$ increas-

es with M more significantly at low M than at high M . For M exceeding 1.57, the increase of $\eta_{span-av}$ is minor. This is also clearly shown with the total averaged η_{av} in Fig. 10. When M increases from 0.47 to 1.97 η_{av} increased from 0.442 to 0.506, improving by 14.5%. But η_{av} only varies within 1.5% after M is higher than 2.

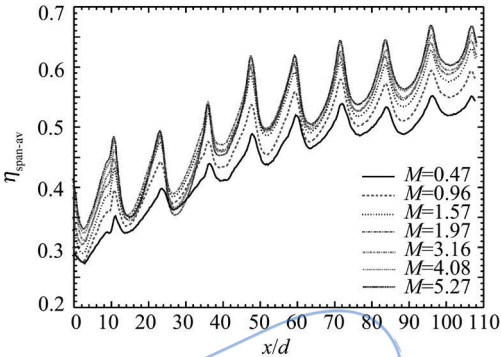


Fig. 9 Spanwise averaged overall cooling effectiveness

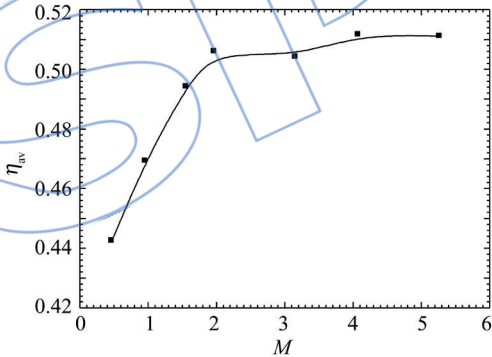


Fig. 10 Total averaged cooling effectiveness

Previous research^[7-8, 12] indicated when M was up to about 1, the coolant jets mostly penetrated into the mainstream and the film lifted off from the wall surface, and the adiabatic cooling effectiveness decreased with increasing M after M exceeded 0.5 due to the film jets liftoff. As described in the Reference[12], higher film convective heat transfer coefficient corresponded to lower adiabatic cooling effectiveness. All these two factors would cause the cooling effectiveness to go lower when M increases after some value like 0.5~1, differing from the trend found by the current study and the Reference[9]. The explanation comes from the cooling contribution of the convective cooling within multi-holes and heat conduction in the plate. Obviously, er

hanced convective cooling within multi holes can counteract or even overrule the degraded filming cooling effect when blowing ratio is increased.

3 Conclusions

The overall cooling effectiveness η of a full coverage effusion cooling plate was studied by measuring the surface temperature using infrared imaging over a wide range of blowing ratios M . The following conclusions are obtained:

The overall η increases along with the streamwise distance for the wide range of M tested, due to the superposition effect of the multirow film cooling.

The overall η substantially benefits from the multi-hole inside convective cooling. In addition to the film cooling contribution (adiabatic η), the hole inside convective cooling not only complements the weakest film protection at initial rows but also helps mitigate the temperature gradient.

The overall η increases asymptotically with increasing M , with the critical M to be about 2, after which the increase becomes minor. The critical M for the adiabatic η is usually less than 1, beyond which the adiabatic η goes down with increasing M . This is due to the hole inside convective cooling, which simply increases with M , compensating the film cooling degrade.

Future experiments to dissect the cooling contribution from the film cooling and hole inside convective cooling, and examine the effect of the wall heat conductivity would offer more insight to the subject.

Acknowledgments

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References:

[1] Andrews G E, Mkpadi M C. Full coverage discrete hole wall cooling: discharge coefficients [R]. ASME 83-GT-79, 1983.

[2] Kasagi N, Hirata M, Kumada M. Studies of full coverage film cooling: Part 1—Cooling effectiveness of thermally conductive wall [R]. ASME 84-GT-37, 1981.

[3] Dagaut P, Cathonnet M. Oxidation of neopentane in a jet stirred reactor from 1 to 10 atm: an experimental and detailed kinetic modeling study [J]. Combustion and Flame, 1999, 118(1-2): 191-203.

[4] Senecal P K, Schmidt D P, Nouar I, et al. Modeling high speed viscous liquid sheet atomization [J]. International Journal of Multiphase Flow, 1999, 25: 1073-1097.

[5] Lin Y, Song B, Li B, et al. Investigation of adiabatic wall film effectiveness of inclined multihole wall film cooling with different angles to wall surface [J]. Chinese Journal of Aeronautics, 1999, 20(3): 202-204.

[6] Bank R V D, Savary N, Linne M, et al. Integrated lean low-emission combustor design methodology [R]. EU Project AST 3-CT-2003-502961, 2007.

[7] Brankovic A, Ryder R C, Jr., Hendricks R C, et al. Measurement and computation of supersonic flow in a lobed diffuser-mixer for trapped vortex combustors [R]. NASA/TM-2001-211127, 2002.

[8] Martinez Botas R F, Yuen C H N. Measurement of local heat transfer coefficient and film cooling effectiveness through discrete holes [R]. ASME 2000-GT-243, 2000.

[9] Gustafsson K M B, Johansson T G. An experimental study of surface temperature distribution on effusion-cooled plates [J]. Journal of Engineering for Gas Turbines and Power, 2001, 123(2): 308-316.

[10] Harrington M K, McWaters M A, Bogard D G, et al. Full coverage film cooling with short normal injection holes [R]. ASME 2001-GT-0130, 2001.

[11] Ekkad S V, Ou S, Rivir R B. A transient infrared thermography method for simultaneous film cooling effectiveness and heat transfer coefficient measurements from a single test [J]. Journal of Turbomachinery, 2004, 126: 597-603.

[12] Coulthard S M, Volino R J, Flack K A. Effect of jet pulsing on film cooling: Part I—Effectiveness and flow-field temperature results [J]. Journal of Turbomachinery, 129: 232-246.

[13] Burdet A, Abhari R S, Rose M G. Modeling of film cooling: Part II—Model for use in three dimensional computational fluid dynamics [J]. Journal of Turbomachinery, 2007, 129: 221-231.

[14] Leylek J H, Zerkle R D. Discrete jet film cooling: a comparison of computational results with experiments [J]. Journal of Turbomachinery, 1994, 116: 358-368.

[15] Scritture J J, Thole K A, Burd S W. Investigation of velocity profiles for effusion cooling of a combustor liner [J]. Journal of Turbomachinery, 2007, 129: 518-526.

[16] Yusop N M, Andrews G E, Ingham D B, et al. predictions of adiabatic film cooling effectiveness for effusion film cooling [R]. ASME GT2007-27467, 2007.

[17] Lefebvre A H. Weak extinction limits of turbulence heterogeneous fuel/air mixtures [J]. Journal of Engineering for Power, 1980, 102: 416-421.