Experimental convective heat transfer characterization of pulsating jet in cross flow: influence of Strouhal number excitation on film cooling effectiveness

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Abstract. In actual gas turbine system, unsteadiness of the mainstream flow influences heat transfer and surface pressure distribution on the blade. In order to simulate these conditions, an experimental film cooling study with externally imposed pulsation is performed with purpose of characterizing both effects of turbine unsteadiness on film cooling (with frequency ranges typical to actual turbine), and also to figure out the range of Strouhal number pulsation under various blowing conditions, which could possibly deliver a performance improvement in film cooling. Influence of injection flow pulsation on adiabatic effectiveness and convective heat transfer coefficient are determined from IR-thermography of the wall for distances to the hole exit between 0 and 30 D.

1. Introduction
It is known from the gas turbine cycle that higher cycle temperature produces a larger amount of work per unit mass flow and improves the (power/weight) ratio of the gas turbine. This improvement is one of the major goals of aerospace engine designers. A major problem associated in achieving this increased performance is the availability of material that can withstand such high temperature combined with material stresses (due to temperature, rotation, and aerodynamic loading). Convective film cooling is the simplest and one of the earliest techniques used. The coolant is passed through multi-pass circuit from hub to tip and ejected at the trailing edge or at the blade tip.

In actual gas turbine system unsteadiness of the mainstream flow influences heat transfer and surface pressure distribution on the blade. The unsteadiness on the turbine airfoil develops from four different sources (Ligrani et al (1996b)): potential flow interactions, shock wave passage, wake passage and random free-stream turbulence from the combustion chamber. A fundamental point of view to study such configuration is inclined jet(s) in cross flow.

Coulthard et al. (2006) experimentally investigated effects of jet pulsation (St=0.0119 to 0.1905) on a flat constant heat flux wall containing a row of five cylindrical hole inclined at 35 in streamwise direction. The pulsed case with St=0.1905 has shown an improved film cooling
effectiveness over the steady case, while in other cases of lower frequencies, the flow is subjected to significant lift-off and had small effectiveness. Direct Numerical Simulation (DNS) performed by Muldoon and Acharya (2009) has shown some improvement in wall coverage with pulsation. The authors show that to the best performance results are obtained at (M=1.5, St=0.32) compared to steady blowing.

Experimental velocity characterization have already been realized by Sultan et al. (2011) for various blowing conditions (M=0.65, 1 and 1.25) and Strouhal number pulsation (St=0, 0.2, 0.3 and 0.5). Authors showed that film concentrations and film trajectories of the injectant flow move to and from the wall with such periodic unsteadiness in the bulk flow, because of instantaneous change in both film flow rate and momentum at the cooling hole exit. Pulsation of injectant at lower Strouhal numbers (St=0.2 and 0.3, where $St = fD/U_i$) and M=1 and 1.25 is able to reorient the injectant close to the wall during the lower part of the pulsation cycle.

Convective heat transfer study has to be realized in Sultan’s experimental setup to study the influence of injected flow pulsation and M blowing ratio on both efficiency and convective heat transfer coefficient distributions.

2. Experimental setup

Film cooling experiments are performed in a subsonic, closed-loop wind tunnel equipped with a thermoregulatory system. The available thermoregulatory unit allows maintaining a desired temperature level by means of an automatic control unit containing a hot source (electric battery) and a quick response refrigeration unit. System is likely to perform more effectively with lower transition time for a temperature range of 288-298 K, even if flow velocity is considerably low.

The test section of the wind tunnel has a square cross-section of 300*300 mm² and a length of 1190 mm. The injection plate contains a row of 5 circular simple-angled holes inclined at 30 degrees with respect to the mainstream flow. In present study, only central hole was used to study effects of injectant flow pulsation (defined by $U_i$ velocity) on the ultimate wall coverage. The hole has a diameter, d, of 13 mm. The longitudinal position of holes with respect to test section inlet is hydro-dynamically fixed to a boundary layer thickness of $\delta \approx 1d$ and 0.93 d, respectively for M=0.65 and M=1. M is blowing ratio, defined by ratio between injected flow momentum and main flow momentum ($M = \rho_iU_i/\rho\infty U\infty$).

The pulsating injected flow ($U_i$) is heated at 310 K by a small heat-dissipating coil controlled with a temperature monitoring probe placed inside a large manifold. The schematic diagram of the injectant flow system is shown in figure 1 (a). The periodic excitation is introduced by a loudspeaker based pulsation system connected flawlessly with injection tube at identical diameter. A signal generator (ITT Instruments-GX240) is used to generate sinusoidal pulses at a required frequency, while the signal amplitude is controlled by a stereo amplifier (JB SYSTEMS-VX400). The loudspeaker used in present work is manufactured by ATOHM, and it has a diameter of 130 mm. Strouhal number, defined by ($St = f_{pulsed}d/U_i$) are equal to 0, 0.2, 0.3 and 0.5.

This experimental setup allows flow and thermal studies which can be compared to previous authors experimental and numerical setup, as shown in figure 2.

Thermography of test wall is performed by using a Cedip Jade-III infrared camera integrated with a highly functional Altair software package that contains extended utilities. The measurement setup is constructed to visualize electrical heated wall across a high transmittivity Saphir glass from a vertical distance of 1 m. It was positioned 3-4 degree off from the vertical axis to avoid reflection of the radiations from viewing window.

A full domain of wall temperature measured at a particular situation is obtained after assembling thermal images obtained at nine consecutive positions along the streamwise direction with an overlap of 5mm (0.385 d). The final temperature field consists of a region enclosed by $x/d=-4.5$ to 32.4 and $z/d=-3$ to 3.

Infrared thermography of the constant heat flux wall was performed with a CEDIP Jade
Figure 1. (a) Schematic diagram of injection flow system. (b) Phase-averaged variation of blowing ratio.

| Study | Present study | Yuen et al. (2003) | Goldstein and Eckert (1974) | Eckert (1970) | Eriksen and Goldstein (1974) |
|-------|---------------|-------------------|----------------------------|--------------|-------------------------------|
| $M = \rho U_f / \rho U_e$ | 0.65 | 1 | 1.25 | 1 | 1.33 | 0.52 | 1.04 | 0.5 | 1 | 0.5 | 0.97 | 1.46 |
| Plane z/d | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Angle of injection, θ (deg.) | 30 | 30 | 30 | 30 | 30 | 35 | 35 | 35 | 35 | 35 | 35 | 35 |
| δy/d | 0.146 | 0.146 | 0.11 | 0.150 | 0.150 | 0.150 | 0.158 | 0.124 | 0.124 | 0.171 | 0.171 | 0.171 |
| $Re_\theta = \rho U_e d / \mu_e$ | 8609 | 8609 | 6629 | 8563 | 8563 | 17700 | 18300 | 22000 | 22000 | 22000 | 22000 | 22000 |

Figure 2. Description of the fundamental test conditions of film cooling experiments referred for comparison.

MWIR camera. The experimental heat transfer model, developed by fenot et al (2005), consists of determining wall temperature distribution at different convective heat fluxes, corresponding to different electric power injected to the wall. The test wall contains a group of four streamwise running thin copper electrical circuits on the side of the wall facing the flow. Copper tracks have a thickness of 35 m, a width of 1.8mm and an inter-track space of 0.2mm. Circuits are feed by four separate power supply units of Agilent-E3634A to generate a constant wall flux with a maximum variation of 2.

Mathematically, the front convective heat flux $\phi_{conv,f}$ is deduced from known electrical flux $\phi_{elec}$, front and rear radiative heat fluxes, respectively $\phi_{rad,f}$ and $\phi_{rad,r}$, and natural convective heat flux at rear wall $\phi_{conv,r}$:

$$\phi_{conv,f} = \phi_{elec} - \phi_{conv,f} - \phi_{rad,f} - \phi_{rad,r}$$  \hspace{1cm} (1)

Adiabatic $T_{ad}$ and $h$, convective heat coefficient are deduced from linear regression:

$$T_w = \frac{1}{h} \phi_{conv,f} + T_{ad}$$  \hspace{1cm} (2)
The $T_{ad}$ is normally expressed in a non-dimensional form as shown below. This non-dimensional term is known as film cooling effectiveness $\eta$:

$$\eta = \frac{T_{ad} - T_{\infty}}{T_i - T_{\infty}}$$  \hspace{1cm} (3)

3. Experimental Results
3.1. Steady Blowing (no pulsation)
In these cases, temperature of mainstream flow was maintained at $T_{\infty} = 293.0 \pm 0.2$ K and temperature of the injecting flow $T_i$ was at $310.0 \pm 0.4$ K. The convective heat transfer coefficient of the film cooled boundary layer is non-dimensional with a convective heat transfer coefficient determined from conventional case of flow over a flat-plate ($h_0$). The term $h/h_0$ signifies relative increase in convective heat transfer coefficient due to the blowing of injectant fluid in free stream boundary layer.

The effectiveness is maximum for the case of $M = 0.65$, as shown in figure 3, which decreases gradually with increment of blowing ratio. The minimum value of effectiveness is encountered for the case of $M = 1.25$, which is roughly less than 6% in the region lying on the downstream side of the hole. At higher blowing ratios, the jet flow diffuses early in the mainstream flow due to outward orientation of jet trajectory, which reduces wall coverage. On the other hand, the heat transfer coefficient increases with increase of blowing ratio and holds maximum contour levels for $M = 1.25$. The increase in heat transfer coefficient is related to tendency of lifting-off of jet from the wall due to a higher vertical momentum, which creates a high pressure region at the leading side and a low pressure region on the trailing side of the jet. On the trailing side, the flow field establishing below the jet usually called as wake region induces turbulence due to entrainment of mainstream flow. The entrainment increases with blowing ratio, which boosts-up the mixing of interacting flow locally and thus increases convection rate. Bons et al. (1995) proposed that lift-off of coolant promotes pumping of free stream flow down to the plate surface by pair of counter rotating vortices of the film.

A comparison of baseline results obtained from infrared thermography of the test wall in steady blowing conditions is shown in the figure 4. The profiles are obtained from central plane ($z/d=0$) for a streamwise distance varying from $(x/d=0$ to 32).
The comparison of results shows that $\eta$, determined for $M=0.65$ and $\theta=30$ in the present study, corresponds well with the results of Eckert (1970), and Goldstein and Eckert (1974) given for $M=0.5$ and $\theta=35$ degree. For $\theta=35$ degree, blowing at $M = 0.5$ is referred as optimum blowing in several references, which means that the injectant flow rate and its trajectory provide an optimum coverage under these conditions (Eckert 1970). In the present case, slightly higher blowing ratio compared to reference cases seems to compensate for the loss of jet momentum in the vicinity of wall, which leads to early diffusion of the injectant flow due to the slightly lower angle of injection. For $M=1$, comparison of present measurement with Eckert (1970), and Goldstein and Eckert (1974) shows some significant difference. The difference in Reynolds numbers, which are 17000 and 22000 compared to the 8609 in the present study, and the angle of injection, which is 35 degrees compared to 30 degrees in the present study explain the difference of dynamic conditions for these cases. However, the comparison with Yuen et al. (2003a) seems to correspond well up to $x/d=13$, beyond this location some increase in adiabatic effectiveness is observed for the present results. The reason for such an increase in effectiveness is the turning of the jet fluid back towards the wall after an initial lift-off. The adiabatic effectiveness for $M=1.25$ appears fairly close to results of Yuen et al. (2003a) given for $M=1.33$.

3.2. unsteady blowing (with pulsation)

Due to lack of experimental and numerical results in literature, no comparisons with previous studies could be made. Figure 5 shows distribution of adiabatic effectiveness with respect to Strouhal numbers for a blowing ratio of $M=0.65$. The effectiveness is maximal for a case of steady blowing ($St=0$). Imposition of sinusoidal pulsation results in a gradual decline of effectiveness with the increase of Strouhal number. In the cases of pulsation, flow of injectant over the wall is no longer continuous, and it is also characterized by downstream movement of spatially accumulated mass of injectant fluid in the form of packets during each period of pulsation. This either causes an occasional detachment or leaves a weak concentration of the injectant, which flows at the wall in regular intervals. At $St=0.5$, distribution of coolant in the near wall region is significantly compromised, and film effectiveness is found to be inferior compared to the other cases.

The distribution of convective heat transfer coefficient with respect to Strouhal number for a blowing ratio of $M=0.65$ is shown in figure 6. The implementation of jet pulsation does not seem to produce any significant difference in convective heat transfer coefficient. Fields of convective heat transfer also feature the presence of a pair of symmetric tines about the central
Figure 5. Adiabatic effectiveness estimated for $M=0.65$, (a) $St=0$, (b) $St=0.2$, (c) $St=0.3$ and (d) $St=0.5$

Figure 6. Convective heat transfer coefficient distribution estimated for $M=0.65$, (a) $St=0$, (b) $St=0.2$, (c) $St=0.3$ and (d) $St=0.5$

plane ($z/d=0$) in some cases, which are believed to be caused by kidney vortices associated with film cooling jets. Coulthard et al (2007) called them as fork-tines in the earlier studies. They use a row of five streamwise injection holes in film cooling with jet pulsation and showed their existence in various cases of unpulsed and pulsed jets film cooling. Generally, the lateral spreading of these tines increases with the increase of blowing ratio.

Figure 7 shows the variation in injectant flow distribution for $M=1.25$. The case of steady blowing shows that effectiveness of injectant distribution on the downstream side of the hole has lowest values due to ascending of the injectant fluid away from the wall. For present blowing ratio, some signs of an upstream horseshoe vortex wrapping around the jet flow are also observed. This vortex system evolves due to intense blockage offered by the upstream side of the jet on the interaction between the two flows. Low frequency pulsations cause an increase in effectiveness, as can be observed for the cases of $St=0.2$ and $0.3$ compared to steady blowing. The time-span associated to the lower part of the pulse cycle is expected to improve wall coverage with reduction of blowing ratio. Also, the higher momentum jet fluid pulsating at $St=0.3$ is able to improve lateral dispersion of the injectant fluid. The distribution of adiabatic effectiveness on the near downstream region of the hole indicates superior lateral spreading of injectant flow for $St=0.3$ compared to the pulsation at $St=0.2$.

Distributions of heat transfer coefficient for the range of Strouhal number pulsation in a flow configuration of $M=1.25$ are shown in figure 8. Under steady blowing, the fields of heat transfer coefficient possesses higher magnitudes compared to $M=0.65$, since development of wider wake region permits greater entrainment of mainstream flow. For the pulsation cases of $St=0.2$ and
0.3, an increase in convective coefficient has been observed. The passing of turbulent wake structures ejected from the near downstream region of the hole mainly contributes to increment in heat transfer coefficient.

3.3. Net Heat Flux Reduction

The variation of spatially-averaged NHFR (Net Heat Flux Reduction) as a function of Strouhal number is shown in figure 9. NHFR is defined by:

\[
NHFR = 1 - \frac{h}{h_0} (1 - \eta \theta) \quad \text{with} \quad \eta = \frac{T_{ad} - T_\infty}{T_i - T_\infty} \quad \text{and} \quad \theta = \frac{T_i - T_\infty}{T_w - T_\infty}
\]  

At M=0.65, film cooling performance is reduced by 19 %, 29 % and 43 % due to pulsing at St=0.2, 0.3 and 0.5 respectively. At M=1, film cooling performance is reduced by 66 % and 26 % for St=0.2 and 0.5, while performance is increased by 36 % for St=0.3. At M=1.25, performance is increased by 116 %, 117 % and 53 % for pulsing at St=0.2, 0.3 and 0.5 respectively. At M=1, pulsation at St=0.2 increases both adiabatic effectiveness and convective heat transfer coefficients. Therefore, there is no net-gain in film cooling performance. It can be observed that the film cooling case with M=0.65 delivers higher performance in both steady blowing and pulsating cases compared to other blowing ratios. For St=0.2, film cooling performance is nearly similar for both M=1 and M=1.25.
4. Conclusions
The effect of injectant flow pulsation on film cooling performance was studied by using technique of infrared thermography. The results for adiabatic effectiveness, convective heat transfer coefficient and Net Heat Flux Reduction (NHFR) are presented. In steady blowing, film cooling effectiveness was superior for the blowing ratio of M=0.65, while convective heat transfer was superior for the blowing ratio of M=1.25.

For M=0.65, effectiveness has reduced consistently in an orderly manner with application of injectant flow pulsation, whereas convective heat transfer coefficient has not shown significant variations with pulsation. The overall performance given by NHFR has also shown a consistent decline with pulsation at different Strouhal numbers for M=0.65.

The case of M=1 have shown a noticeable increase in film cooling effectiveness as well as in heat transfer coefficient for St=0.2 compared to the other cases.

For M=1.25, an increase in film cooling effectiveness and convective heat transfer coefficient was observed for St=0.2 and 0.3 compared to St=0 and 0.5. The NHFR, for M=1.25, have shown an increase in all cases compared to steady blowing.

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