Heat Transfer Measurements of an Impinging Synthetic Air Jet with Constant Stroke Length

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Abstract

Synthetic jets are a relatively new technology and have shown great promise in a number of practical applications such as flow control through active boundary layer mixing and enhanced heat transfer through forced convection. It has been shown by Kercher et al. (2003) that synthetic jets can deliver similar cooling effects to conventional steady flow impinging jets without the need for an external air supply system. As highlighted in a review article by Glezer and Amitay (2002), impinging synthetic jets are therefore an extremely promising technology for applications in electronics cooling. The current research is concerned with the heat transfer distribution and flow characteristics of an impinging synthetic jet. Tests were conducted for Reynolds numbers ranging from 1100 to 4900, and nozzle to impingement surface spacings from 1 to 6 diameters. The dimensionless stroke length ($L_0/D$) was maintained constant and equal to 17 in all the reported data. The results obtained show a strong correspondence between measurements of the mean and fluctuating heat transfer distributions and local velocity and turbulence intensity measurements along the heated surface. Strong coherent vortices impact the heated surface, travel outwards while promoting mixing in the wall jet region, and finally break down into small scale turbulence.

Nomenclature

- $D$: jet diameter, m
- $\delta$: sensor thickness, m
- $f$: frequency, Hz
- $H$: height of nozzle above plate, m
- $k_\text{a}$: thermal conductivity of air, W/mK
- $k_s$: thermal conductivity of sensor barrier, W/mK
- $L_0$: jet’s stroke length, m
- $Nu$: Nusselt number
- $Nu'$: root-mean-square Nusselt number
- $\phi$: phase angle, °
- $q$: rate of heat transfer, W
- $q'$: root-mean-square heat transfer rate, W
- $\bar{q}$: heat flux, W/m$^2$
- $R$: sensor resistance, Ω
- $r$: radial distance along plate from geometric centre, m
- $Re$: Reynolds number, $Re = U_0D/\nu$
- $\rho$: density, kg/m$^3$
- $St$: Strouhal number, $fD/U$
- $T$: temperature, °C
- $\Delta T$: temperature difference, °C
- $U_0$: area averaged orifice velocity, m/s
- $U_p$: peak jet velocity, m/s
- $\bar{U}$: time and spatially-averaged jet exit velocity, m/s
- $\mu$: viscosity, kg/ms
- $V_{out}$: output voltage, V
- $\nu$: kinematic viscosity, m$^2$/s
- $\omega$: vorticity

1 Introduction

Impinging synthetic air jets can be used to transfer heat in applications ranging from the cooling of a manufacturing process to the thermal management of electronics, in particular microprocessors. This latter application is increasingly important as current trends in electronic components show a continuous increase in heat flux densities as both processor clock frequency and the number of transistors required for a given implementation grow. This increase has led to an even greater need for more efficient and higher density heat removal in closely packed systems; this need is predicted
to continue to grow by a factor of two every four years for the foreseeable future, Gunther et al. (2001). While forcing more air through the system or designing an elaborate heat sink can sometimes meet increasing thermal demands, there is not only a significant cost increase associated with more elaborate solutions but also environmental regulations can limit system (fan) noise. It is for this reason that we are looking to new technologies such as synthetic jets to reduce both the cost per watt of heat dissipated and the environmental noise produced by these systems.

Synthetic jets operate on a simple principle; a flexible membrane forms one side of a partially enclosed chamber. Opposite to the membrane is an opening, such as an orifice. A mechanical actuator, piezoelectric diaphragm or magnetic coil causes the membrane to oscillate and periodically force air into and out of the opening. This results in a non-zero mean streamwise pulsating jet formed in front of the orifice which can be directed at a heated surface to enhance cooling.

Comprehensive studies of fluid flow characteristics of both free and impinging synthetic jets have been presented by Mittal et al. (2002) and Amitay and Canelle (2005). Amitay and Canalle (2005) investigated the transitory behaviour of an isolated synthetic jet for a broad range of stroke lengths (between 5 and 50 times the slit width) and Reynolds numbers (between 85 and 408). Their study identified trends between jet formation frequency and number of cycles taken to reach a stable periodic behaviour.

Due to the constantly varying nature of synthetic jet flow velocity a number of methods for calculating the characteristic flow velocity have been put forward. Smith and Glezer (2002) argued for use of an average jet velocity, $U_0$, since continuous jets with $U_{ave} = U_0$ have the same volume flux directed downstream when velocity is averaged over a cycle at the exit plane. Holdman and Utturkar (2005) stated that the governing parameters for a synthetic jet are to be based on the same “slug-velocity-profile” model, which include a dimensionless stroke length $L_0/D$ and Reynolds number $Re = UD/\nu$ based on the velocity scale:

$$\bar{U} = fL_0 = f \int_0^{T/2} U_0(t) dt$$  \hspace{1cm} (1)

where $D$ is the orifice diameter, $\nu$ is the kinematic viscosity, $U_0(t)$ is the centreline velocity at the exit (although in this paper the area averaged orifice velocity is used), $f$ is the driving frequency, $T = 1/f$ is the period, and $L_0$ is the stroke length. For this research average velocity $U_0$ and dimensionless stroke length $L_0/D$ were calculated using peak jet velocity $U_p$ measured using PIV techniques:

$$U_0 = \frac{U_p}{\pi/2}$$  \hspace{1cm} (2)  

$$L_0/D = \frac{U_0}{Df}$$  \hspace{1cm} (3)

With this method of calculating the velocity of the synthetic jet it is possible to acquire a Reynolds number for the jet flow:

$$Re = U_0D/\nu$$  \hspace{1cm} (4)

The main variables for synthetic jet impingement heat transfer are the excitation frequency, stroke length and the nozzle height above the impingement surface. Comprehensive reviews of heat transfer to impinging synthetic air jets have been conducted by Gillespie et al. (2007), Wang et al. (2006) and Pavlova and Amitay (2006) amongst others. Campbell et al. (1998) conducted a review of heat transfer to an impinging synthetic jet for a wider range of parameters that included varying the number of jets in a jet array, various levels of confinement and the addition of cross flow. Gallas et al. (2003) presented an extensive review of numerical investigations that have been conducted in the area of lumped element modelling of piezoelectric-driven synthetic jet actuators. Campbell et al. also conducted experiments to determine the effectiveness of synthetic air microjets in cooling packaged thermal test chips and a laptop computer processor while Kercher et al. (2003) investigated the applicability of miniaturized synthetic jet technology to the area of thermal management of microelectronic devices and directly compared the cooling performance of these microjets with standard cooling fans. Gillespie et al. (2007) investigated the effects of a small-scale, rectangular
Many investigations have been undertaken into the optimum operating parameters of the synthetic jet; one by Gallas et al. (2003) optimised the jet with respect to variables such as driving frequency, cavity volume, nozzle length and nozzle diameter. It has been documented by Li and Zhong (2005) and Kercher et al. (2003) that the shape of the heat transfer distribution changes significantly with nozzle to impingement surface spacing. An experimental investigation of the transitory behaviour of an isolated synthetic jet was conducted by Amitay and Canalle (2005) in which the transients were broken down into four stages and characterised. They concluded that the transients were dependent on a number of parameters including the Reynolds number, the formation frequency, and the stroke length of the synthetic jet. Smith and Swift (2003) concluded that there exists a minimum dimensionless stroke length \( \frac{L_0}{D} \) below which no synthetic jet is formed. It is also stated that the far field behaviour of synthetic jets appears to be a function of both \( \frac{L_0}{D} \) and \( \text{Re} \); these findings were confirmed in a paper by Holdman and Utturkar (2003).

Although many studies have investigated the heat transfer to an impinging synthetic jet as a function of Reynolds number, these tests have not generally been conducted while maintaining a constant stroke length. It is of great importance that a better understanding of the effect of varying Reynolds number on heat transfer independent of stroke length be obtained and due to the potential of the synthetic jet to be used in confined environments such as inside computers it is increasingly important that these parameters can be optimised for low jet to surface spacings. Furthermore, little information is available for combined flow visualisation and heat transfer profiles for small spacings and no data regarding fluctuating heat transfer profiles has been provided. The present study sets out to provide Particle Image Velocimetry (PIV) velocity and phase locked vorticity plots, detailed local heat transfer profiles, both time-averaged and fluctuating, for a synthetic jet impinging onto a heated surface at close range.

![Figure 1: Rig Design](image)

### 2 EXPERIMENTAL SETUP

The three main elements of the experimental rig are the synthetic jet, heated impingement surface and a particle image velocimetry system. Both impingement surface and synthetic jet are mounted on independent carriages that travel on orthogonal tracks; the carriage for the impingement surface is moved using a computer controlled traverse. The measurement instruments associated with the flat impingement surface are two single point heat flux sensors and two thermocouples. The rig design is presented in figure 1.

The operation of the synthetic jet relies primarily on an acoustic speaker mounted on an enclosed cavity with an orifice plate on the opposing side to provide the entrainment path for the working fluid. A driving frequency for the speaker is provided by a TTi TG315 Signal Generator, and the
signal is amplified using a Kemo® 40 Watt power amplifier. The speaker was supplied with a sinusoidal wave of specific amplitude and frequency so as to obtain the desired stroke length and Reynolds number.

The jet impinges onto a surface that consists of a 5mm thick flat copper plate measuring 425mm x 550mm. To the underside of the plate a silicon rubber heater mat is glued with a thin layer of adhesive. The mat is approximately 1.1mm thick. The underside of the plate is insulated from the surroundings. The plate assembly is such that it approximates a uniform wall temperature boundary condition. The system is typically operated at a surface temperature of 40°C.

The first sensor is an RdF Micro-Foil® heat flux sensor. This sensor measures the temperature differential across a known thermal barrier using a differential thermopile. We can calculate the heat flux through the sensor by using the following equation:

$$\bar{q} = k_s \frac{\Delta T}{\delta}$$  \hspace{1cm} (5)

where $\Delta T$ is the temperature difference across the thickness ($\delta$) of the barrier and $k_s$ is the thermal conductivity of the barrier (kapton). A single pole T-type thermocouple is also embedded in this sensor to measure the local temperature.

A Senflex® Hot Film Sensor operates in conjunction with a Constant Temperature Anemometer (CTA) to measure the fluctuating heat flux to the impinging jet. This sensor consists of a nickel sensor element that has been electron beam deposited onto a 0.051mm thick Upilex S polyimide film. The hot film element has a physical area of approximately 0.1mm x 1.44mm and is less than 0.2µm thick. The sensor has a typical cold resistance of between 6 and 8 Ohms. Two copper strips are also deposited on the film; these provide terminals for connection to the CTA. These strips have a resistance of approximately 0.002 Ω/mm. The hot film is maintained at a slight overheat (≈5°C) above that of the copper plate using a Dantec StreamLine CTA. The power required to maintain the film at this overheat is equal to the heat actively being dissipated from the film. The CTA essentially acts as a Wheatstone bridge where the hot film acts as one resistor in the bridge. The resistance of the film varies with temperature and therefore, this film temperature can be controlled by varying a decade resistance which forms one arm of the Wheatstone bridge. The square of the voltage required to maintain the film at a constant temperature is proportional to the heat transferred to the air as described in equation 6.

$$q_{\text{dissipated}} \propto \frac{V^2_{\text{out}}}{R}$$  \hspace{1cm} (6)

The heat transfer rig used in this paper is similar to that used by O’Donovan (2005). The mean and fluctuating Nusselt numbers have calculated uncertainties of 5.7% and 30.0% respectively. These uncertainties are based on a worst case scenario where the uncertainty is a percentage of the smallest measurements. A complete calibration and uncertainty analysis for this experimental set-up is presented by O’Donovan (2005).

Flow visualisation and velocity measurements have been performed on a separate rig using particle image velocimetry (PIV). The PIV system comprises of a New Wave Nd:YAG twin cavity laser and a LaVision FlowMaster 3S CCD-camera (1280x1024 pixels, 12 bit) with 28 mm lens. A glycol-water aerosol is used for seeding, with particle diameters between 0.2 and 0.3 µm. Customised optics are used to generate a 0.3 mm thick light sheet. The CCD-camera is mounted perpendicular to the light sheet. The pulse separation time of the laser is determined such that the maximum particle image displacement does not exceed ¼ of the interrogation window, Raffel et al. (1998). The velocity fields have been processed with LaVision’s DaVis 6 software, using multi-pass cross-correlation with an interrogation window size decreasing from 64×64 to 16×16 pixels at 50% overlap.
The laser is fired using a trigger signal, with a preset time delay $\delta t$ between trigger detection and laser firing. The trigger is generated by the signal generator driving the loudspeaker. The time delay $\delta t$ is varied stepwise to obtain phase-locked measurements from 0 to 360 degrees. In each phase, about 100 velocity fields are acquired and averaged, yielding the phase-resolved velocity field.

Figure 2. Streamline plot of the time-averaged velocity at $Re = 3700$ and $Lo/D = 17$ for (a) $H/D=1$ (b) $H/D=2$ (c) $H/D=4$

Figure 3. Phase Locked Vorticity Plot for $H/D = 1$ and $Re = 3700$ and $Lo/D = 17$ for (a) $\varphi=120$ (b) $\varphi=180$ (c) $\varphi=240$ (d) $\varphi=300$ (e) $\varphi=0$ (f) $\varphi=60$

Figure 4. Phase Locked Vorticity Plot for $H/D = 2$ and $Re = 3700$ and $Lo/D = 17$ for (a) $\varphi=120$ (b) $\varphi=180$ (c) $\varphi=240$ (d) $\varphi=300$ (e) $\varphi=0$ (f) $\varphi=60$

Figure 5. Phase Locked Vorticity Plot for $H/D = 4$ and $Re = 3700$, $Lo/D = 17$ and $\varphi=120$
3 Results and Discussion

Figures 3 to 5 show phase-locked dimensionless vorticity fields $\omega D/U_0$ for the synthetic jet. The starting phase has been chosen to correspond to maximum ejection. The vorticity corresponds to $\omega = \frac{1}{2} (\partial V/\partial x - \partial U/\partial y)$ where positive values (blue) indicate clockwise rotation. The locations of peak vorticity correspond to vortex rings, propagating down towards the heated plate. Figure 3 (a) shows a vortex ring forming at $r/D = 1$ with maximum flow velocity occurring in the centre of the jet; the ejection phase continues to travel radially until $\phi = 240^\circ$ where it can be seen to reach a maximum radial distance of about 2.2 diameters before subsequently moving back towards the centre. The change in vortex direction corresponds to the change in jet flow direction. It is evident from figure 3 that at this low height above the plate the vortex fails to escape the confinement of the jet and so becomes recirculated into the working fluid. This finding is confirmed in figure 2 (a) where the recirculation zones can clearly be seen to extend up to $r/D = 2.2$. The effect this recirculation has on heat transfer is significant and can be seen in figures 6 and 7 for $H/D = 1$, where both heat transfer and fluctuating heat transfer drop significantly at the radial distance corresponding to where the vortex becomes re-entrained.

Figure 4 (a) once again shows the formation of a vortex ring at a radial distance of approximately 1, with high flow velocity occurring at the plate surface; as the phase increases the vortex travels outward radially. At this height above the surface the plate lies in the vortex formation region; this results in a high velocity flow occurring between the vortex and the plate at a radial distance of $r/D = 0.7$. This flow travels radially outwards parallel to the plate surface. Heat transfer and fluctuating heat transfer profiles seen in figures 8 and 9 show how this high velocity parallel to the surface affects the heat transfer. It can be seen that the mean heat transfer distribution has a local minimum at the stagnation point for Reynolds numbers of 2300 and above. This local minimum can occur also for steady impinging jets as has been shown by Hoogendoorn (1977) and Lytle and Webb (1994). This local minimum is due to low turbulence at the stagnation point, where the flow approximates true stagnation. With increasing radial distance from the stagnation point the heat transfer increases to a maximum at $r/D = 0.7$, which can be seen for the range of Reynolds numbers of 2300, 3700 and 4900. This peak corresponds with the high velocity which was seen to occur in figure 4 (a) at the same radial distance. These high flow velocities result in the large peaks observed in the fluctuating heat transfer distribution for the same height, as seen in figure 9; it can also be seen in $y??$ for $H/D = 1$. With increasing distance from the centre both heat transfer and fluctuating heat transfer continue to reduce. It is also apparent from Figures 8 and 9 that with increasing Reynolds number the magnitudes of both heat transfer and fluctuating heat transfer increase almost linearly due to increasing fluid temperature.

It is noted that as the height above the impingement surface is increased, the maximum heat transfer fluctuation now occurs at the centre of the jet and decreases radially. Figure 5 shows that for nozzle to surface spacings above $H/D = 2$ the plate no longer lies within the vortex formation region and so upon impact the peak surface velocity now occurs at the centre of the jet. At this height the net fluid momentum impacts at $r/D = 0$ and spreads radially, resulting in a large peak in heat transfer and fluctuating heat transfer occurring at $r/d = 0$; this is evident in figures 10 and 11.

Figure 6 shows that as the spacing is increased from $H/D$ of 4 to H/D of 6 there is a decrease in the peak heat transfer. This is due to the interaction and subsequent breakdown of the coherent vortical structures as they approach the plate, resulting in increased mixing and hence the “spreading” of the heat transfer curve; it also results in the higher turbulence intensities with each height increase, as observed in figure 7.

It is worth noting that as the height above the plate is increased the recirculation effect is reduced. It can be seen in figure 2 that at $H/D = 1$ there exist strong recirculation zones on either side of the jet. Although these recirculation zones still exist at $H/D = 2$ it is evident that there is far more entrainment, increasing again at $H/D = 4$. There is evidence to suggest that this trend increases up to $H/D = 10$, as reported by Gillespie et al. (2007).

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4 Conclusion

Heat transfer profiles, fluctuating heat transfer distributions and phase locked vorticity plots have been presented for an axially symmetric synthetic air jet of various Reynolds number with constant dimensionless stroke length of 17 impinging normal to a heated surface. It was found that higher Reynolds numbers produced greater heat transfer. The observed trends in the heat transfer profiles
for different jet to plate spacings have been adequately explained by comparison to phase-locked velocity measurements. Three different regimes can be discerned for low, moderate and large values of the jet to plate spacing. Based on the phase-locked flow fields, the transition between moderate and large spacing seems to be related to the synthetic jet stroke length. Future work in this area will include the analysis of varying stroke length with constant Reynolds number.

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