SURFACE HEAT TRANSFER FROM AN IMPINGING SYNTHETIC AIR JET

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ABSTRACT

The implementation of synthetic jets for use in the cooling of electronics is a relatively new technology. Steady flow impinging jets can produce relatively high heat transfer coefficients; however it has been shown by Kercher et al., (2003) that synthetic jets can deliver similar cooling effects without the need for an air supply system. Impinging synthetic jets are therefore an extremely promising alternative for applications such as cooling of electronic components, Glezer et al., (2000). A study has been undertaken of the heat transfer distribution to an impinging synthetic jet flow with two exit geometries; contoured nozzle and orifice plate. The jet is directed at a heated copper plate, which approximates a uniform wall temperature boundary condition. Nusselt number profiles generated by the synthetic jet for various driving frequencies and heights above the plate were obtained. Tests were conducted for a range of experimental parameters, which include driving frequencies from 40 Hz to 160 Hz, non-dimensional surface to jet exit spacings (H/D) from 1 to 6 and nozzle exit geometry. Heat transfer measurements were compared for both a contoured jet nozzle and an orifice nozzle. Jet exit geometry is shown to have a substantial effect on the flow characteristics of the synthetic jet; this results in significantly different heat transfer distributions.
INTRODUCTION

Impinging synthetic air jets can be used to transfer heat in diverse applications, which may vary 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 produced by microchips. 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 in the foreseeable future, Black et al. (1998).

Synthetic jets operate on a simple principle; a flexible membrane or diaphragm forms one side of a partially enclosed chamber. Opposite to the membrane is an opening, such as a jet nozzle or orifice plate. A mechanical actuator, piezoelectric diaphragm or magnetic coil causes the membrane to oscillate and periodically forces air into and out of the opening. This oscillation creates a pulsating jet that can be directed at a heated surface.

Comprehensive studies of fluid flow characteristics of both free and impinging synthetic jets have been presented by Mittal and Rampunggoon (2002), Amitay and Cannelle (2005) and Yao et al. (2000). Amitay and Cannelle (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, \( U_{ave} \), 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 (2003) 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 \( Re_f = U D / \nu \) based on the velocity scale:

\[
\bar{U} = f L_0 = \int_0^{T/2} U_0(t) dt \tag{1}
\]
where \( D \) is the orifice diameter, \( \nu \) is the kinematic viscosity, \( U_0(t) \) is the centreline velocity at the exit, \( f \) is the driving frequency, \( T = 1/f \) is the period, and \( L_o \) is the distance that a slug of fluid travels away from the orifice during the ejection phase of the cycle. With this method of calculating the velocity of the synthetic jet it is possible to acquire a Reynolds number for the jet flow:

\[
\text{Re}_{cJ} = \frac{U_0 D}{\nu}
\]  

(2)

The main variables for synthetic jet impingement heat transfer are the excitation frequency, the stroke length and the nozzle height above the impingement surface. Comprehensive reviews of heat transfer to impinging synthetic air jets have been conducted by Kercher et al. (2003) Wang et al. (2006) and Pavlova and Amitay (2006) amongst others. Campbell and Black (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 and Black (1998) conducted experiments to determine the effectiveness of synthetic air microjets in cooling packaged thermal test chips and a laptop computer processor. Kercher et al. (2003) investigated the applicability of miniaturized synthetic jet technology to the area of thermal management of microelectronic devices; they went on to characterize and optimise different driving elements, geometries, configurations, and to directly compare the cooling performance of these microjets with standard cooling fans.

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 and that in fact there exists an optimum nozzle to surface spacing for maximum desired heat transfer performance. 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 (2001) concluded that there exists a minimum dimensionless stroke length \( L_o/D \) 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 \( L_o/D \) and \( \text{Re}_{cJ} \). Smith and Swift’s (2001) findings were confirmed in a paper by Holdman and Utturkar (2003) who went on to propose a jet formation criterion. In this paper a constant is derived which is dependent upon geometric factors of the synthetic jet, factors such as the radius of curvature, orifice shape, and aspect ratio of the slot.

There have been many studies undertaken to investigate and characterise a synthetic jet flow and average heat transfer to an impinging synthetic jet, however there have been few investigations documenting the effect nozzle type has on heat transfer at relatively low H/D. It is of great importance that a better understanding is obtained of the effect different nozzle geometries have on the operation of the synthetic jet. Due to the potential of the synthetic jet to be used in confined environments such as inside computers and during manufacturing processes it is increasingly important that it is optimised for maximum heat transfer.

Furthermore, little information is available for the detailed Nusselt number profiles produced by synthetic jets for various frequencies and geometries at small spacings and no data regarding fluctuating Nusselt number profiles has been provided. The present study sets out to provide detailed local heat transfer profiles, both time-averaged and fluctuating, for a synthetic jet impinging onto a heated surface at close range using both a contoured nozzle and an orifice plate.
EXPERIMENTAL SETUP

The two main elements of the experimental rig are the synthetic jet and heated impingement surface. Both 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 ability of the carriage to traverse perpendicular to the synthetic jet enables the jet to be positioned at a distance relative to either sensor. The rig design and a photograph of the rig are presented in figures 1 and 2 respectively.

Figure 2: Photograph of Rig

The operation of the synthetic jet relies primarily on an acoustic speaker mounted on an enclosed cavity with either a contoured nozzle or orifice plate on the opposing side to provide the entrainment path for the working fluid. The
speaker is a Visaton® FR8 audio speaker. A signal input is required to control the driving frequency of the speaker; this 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 constant power of 2.5W for all tests and throughout the frequency range.

The cavity body was fabricated using a method of rapid prototyping called fused deposition modelling and is constructed from a type of thermoplastic. The cavity has an approximate volume of 100cm³ with nozzle diameters of 5mm. The contoured nozzle has been designed so as to maximise fluid ejection during the outward stroke of the synthetic jet cycle by implementing a chamfered transition between nozzle and cavity. The orifice plate consists of a 10mm thick plate with a 5mm diameter cylindrical bore. The synthetic jet is clamped on a carriage, which permits height adjustment from 1 to 4 diameters above the impingement surface. The jet is fixed at a normal angle of impingement.

The jet impinges onto a surface that consists of a 5mm thick flat copper plate measuring 425mm x 550mm. The surface of the copper plate is coated in a matt black paint with an emissivity of 0.96. 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 complete assembly is mounted on a traversable aluminium carriage. The underside of the plate is insulated from the surroundings through the use of 30mm thick household insulation. (90°).

The plate assembly is such that it approximates a uniform wall temperature boundary condition. The system is typically operated at a surface temperature of 60°C. The heat flux sensors are mounted flush with surface of the plate; to achieve this it was necessary to embed the sensors in the plate by machining grooves in the surface and gluing them down using an epoxy designed with high thermal conductivity.

The sensors are positioned centrally on the plate and, together with the jet and plate carriage arrangement, allow for heat transfer measurements beyond 20 diameters from the geometric centre of the jet. In the current study, testing has only been concerned with a region extending to 6 diameters from the geometric centre of the jet.

The first sensor flush mounted on the heated surface is an RdF Micro-Foil® 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. A Dantec StreamLine Constant Temperature Anemometer is used to control the temperature of the hot film. The film is maintained at a slight overheat (≈5°C) above the temperature of the copper plate. The power required to maintain the film at this temperature overheat is equal to the heat actively being dissipated from the film. The CTA essentially acts as a Wheatstone bridge where the probe, or in this case 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; the decade resistance 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 4. Due to the slight overheat of the sensor above the impingement surface, corrections for losses such as conduction to the surface must be made to acquire accurate measurements.

$$q_{\text{dissipated}} \propto \frac{V_{\text{out}}^2}{R}$$  \hspace{1cm} (4)

Laser Doppler Anemometry was employed to perform exit velocity measurements of the synthetic jet flow and hence calculate the Reynolds number of the jets at various frequencies. The LDA system used is based on a Reliant 500mW Continuous Wave laser supplied by Laser Physics. This laser system can measure two components of the velocity in orthogonal directions. Hence, the laser is split into two pairs of beams emanating from a single lens. The blue and green lasers have wavelengths of 488nm and 514.5nm respectively; this enables us to acquire time varying
coincident velocity measurements in orthogonal directions. Each of the four beams has a diameter of 1.35mm and focuses at a point 250mm from the lens. For this velocity measurement technique it is necessary to seed the fluid flow with smoke particles, which were introduced through the use of a smoke generator. It is necessary to seed an enclosed area containing the synthetic jet, lasers and particle generator; this is due to the fact that unlike continuous jets the working fluid required for the synthetic jet to work is sourced from within its operating environment and not from an external source. The LDA system employed is capable of operating in both forward-scatter and back-scatter; for the purposes of this investigation the latter was chosen. The signal is acquired and processed by a Burst Spectrum Analyser (BSA) which computed the velocity.

Additional full-field velocity measurements have been performed 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 light sheet is aligned with the synthetic jet axis. 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 32×32 pixels at 50% overlap. The laser is fired using a trigger signal, with a preset time delay δt between trigger detection and laser firing. The trigger is generated by the signal generator driving the loudspeaker. Depending on the value of δt, the PIV system measures an instantaneous velocity field at a certain phase. The time delay δt 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.

The 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. It is clear from the results presented that the uncertainty in Nu’V is, in general, less than 30%. A complete calibration and uncertainty analysis for this experimental set-up is presented by O’Donovan (2005).

RESULTS AND DISCUSSION

As a selected result of the PIV measurements, Fig. 3 shows the phase-locked dimensionless vorticity field ωD/U0 for the synthetic jet with the orifice. The phase corresponds to maximum ejection. The vorticity corresponds to ω = (1/2)(∂V/∂x − ∂U/∂y). In Fig. 3, positive values (blue) indicate clockwise rotation. The locations of peak vorticity correspond to vortex rings, propagating down towards the heated plate. Figure 3 shows that only two or three successive vortices can be discerned, before the vortices break up and the jet structure becomes increasingly turbulent. For a smaller ratio of height to stroke length H/L0, the vortices do impinge onto the surface and stretch outwards.
At $H/D = 1$ the mean heat transfer distribution has a local minimum at the stagnation point for both geometries; this can be seen in figures 4 and 6, and also to a lesser extent for $H/D = 2$ in figures 8 and 10. 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 at low $H/D$ the heat transfer increases to a maximum at approximately $r/D = 0.5$, which can be seen for the full range of driving frequencies in figures 8 and 10. This is due to the acceleration of the flow in the wall jet as it escapes the nozzle lip. With increasing radial distance the heat transfer then reduces before rising to a secondary peak (at $r/D$ of between 1 and 2), this secondary peak can clearly be seen in figure 4 for $H/D = 1$. It can be seen that the prominence of this second peak increases with increasing driving frequency. It is thought that these secondary peaks are a result of transition to turbulence of the wall jet boundary layer that develops from the stagnation point.

However although the secondary peak is present for both the contoured nozzle and the orifice plate at $H/D = 1$ it is noted that as the height above the plate is increased to $H/D = 2$ the peak is almost non-existent for the orifice, whereas a notable swelling can still be seen between $r/D = 1$ and 2 for the contoured nozzle. This can be seen in figure 8 and 10. Beyond this secondary peak the heat transfer distribution decays once more with increasing radial distance.

It is noted that as the height above the impingement surface is increased, the maximum fluctuation now occurs at the centre of the jet and decreases radially; the reason for this is due to vortex interaction. It can be seen in figure 3 that as the jet approaches the plate from a greater height there is increased interaction between vortices before impact, this interaction causes breakdown of the coherent vortical structures to occur resulting in increased mixing. It is this mixing which produces the high turbulence intensities observed at greater $H/D$ for both geometries. The result of this interaction is that the jet stream is fully turbulent prior to impacting upon the plate. As such, when $H/D$ is increased, the local minimum in the heat transfer distribution at the stagnation point gradually changes to an overall maximum and the stagnation point is now a time-averaged stagnation point where there exists a high degree of turbulence; the influence of which can be seen in figures 4 and 6.

Distributions of the fluctuating heat transfer are presented in figures 5, 7, 9 and 11 for the same range of driving frequencies and nozzle to surface spacings. The fluctuating Nusselt number is an indication of the instability in the
flow along the impingement surface. These distributions are similar in shape to the mean heat transfer distributions for both nozzle geometries, which indicates that the magnitude of the mean heat transfer is highly dependent on the turbulence in the wall jet. The most notable aspect of these distributions at low height above the plate are the relatively low fluctuations at $r/D = 0$ followed by a substantial increase in magnitude as $r/D$ approaches 1. The initial depression can be explained by considering the free jet; as it exerts pressure on the wall jet within the stagnation zone the fluctuations are maintained low at the stagnation point for relatively low values of $H/D$, this results in the low fluctuating heat transfer observed in all distributions. The observed large peaks at $H/D$ of 1 are a result of vortex rings generated by the synthetic jet, which can be seen in figure 3, impacting upon the impingement surface. The impacts of these coherent vortices result in the observed large heat transfer fluctuations in both figure 5 and 7. As the radial distance increases further the wall jet escapes the effects of the free jet and undergoes transition to a more turbulent flow, it is this increase in turbulence intensity that leads to the observed secondary peaks at $r/D = 1 - 2$ in the time averaged Nusselt number distributions in figures 4 and 6, and the slight elevations in the fluctuating Nusselt number distributions at $r/D = 2$, which are clearly seen at all frequencies in figure 11.

It is significant that although fluctuating heat transfer magnitudes for both orifice and contoured nozzle are quite similar, it is observed that as the height above the plate is increased from a low $H/D$, the high fluctuations generated due to the impact of coherent vortices reduce and vortex mixing causes turbulence to dominate. This is especially true for the orifice. Thus, it can be seen in figure 4 and 6 that the peaks produced at low $H/D$ for the orifice are not only far less pronounced than those produced by the contoured nozzle, but they also degrade to turbulence much sooner with increasing distance from nozzle exit. It is thought that this is due to the more uniform flow produced by the contoured nozzle. This more uniform flow enables the contoured nozzle produce vortices with significantly higher momentum than those formed by the orifice; it is this increase in momentum that maintains vortex coherency to a greater $H/D$.

Comparing figures 5 and 7 for low $H/D$ it can be seen that the more uniform flow produced by the contoured nozzle also results in an increased pressure exerted on the wall jet within the stagnation zone. This higher pressure further lowers fluctuations at the stagnation point significantly below that of the orifice. It is also evident that the more developed flow produced by the contoured nozzle prolongs the existence of the secondary peak in the heat transfer distribution to a greater height above the plate than that produced by the orifice, as seen in figures 5 and 7. This is because the more uniform flow is maintained to a greater $H/D$, and as a result the wall jet only escapes the effects of the free jet at a greater $H/D$ and must still undergo transitions to a more turbulent flow below this. For the orifice above $H/D = 1$, much of the flow has transition to turbulence and no secondary peak is evident, this is confirmed by the fluctuating heat transfer profile shown in figure 5.
Figure 4: Orifice Plate $\textit{Nu}$ Distributions; $f = 80\text{Hz}$

Figure 5: Orifice $\textit{Nu}'$ Distributions; $f = 80\text{Hz}$

Figure 6: Contoured Nozzle $\textit{Nu}$ Distributions; $f = 80\text{Hz}$

Figure 7: Nozzle $\textit{Nu}'$ Distributions; $f = 80\text{Hz}$
Figure 8: Orifice $Nu$ Distributions; $H/d = 2$

Figure 9: Orifice $Nu'$ Distributions; $H/d = 2$

Figure 10: Contoured Nozzle $Nu$ Distributions; $H/d = 2$

Figure 11: Contoured Nozzle $Nu'$ Distributions; $H/d = 2$
CONCLUSION

Heat transfer profiles and fluctuating heat transfer distributions have been presented for an axially symmetric synthetic air jet impinging normal to a heated surface for both a contoured nozzle and orifice plate. This investigation has shown that for identical operating parameters, overall heat transfer achieved by an orifice is less than that achieved by a nozzle. It is also shown that for low nozzle to impingement surface spacings, secondary peaks similar to those reported in the literature for impinging continuous jet flows occur in the mean heat transfer distributions for both geometries. These secondary peaks are more prominent for the contoured nozzle than for the orifice plate, due to the more uniform flow produced by the contoured nozzle. It can be seen in the heat transfer and fluctuating heat transfer distributions that it is this more uniform flow produced by the contoured nozzle that delays the onset of flow deterioration to turbulence.

It has also been proposed that the use of the contoured nozzle aids with the formation of vortices with higher momentum and greater coherency, which has been confirmed by the fluctuating heat transfer distributions presented in this paper. It is for this reason that the secondary peaks present in the mean heat transfer distributions are present to a greater height above the surface for the contoured nozzle compared to that of the orifice plate. These secondary peaks have been attributed to an abrupt increase in turbulence in the wall jet boundary layer. The current investigation presents data in support of this assertion and reports on the influence of nozzle geometry on the overall heat flux profile. Fluctuating heat transfer distributions presented indicated that the magnitude of the mean heat transfer is not only highly dependent on the turbulence in the wall jet but also on the type of flow produced by the nozzle geometry. Future work in this area will include the analysis of the time varying heat flux signal to provide a greater understanding of the effect nozzle geometry has on the local heat transfer. Further analysis using particle image velocimetry may give also a greater insight into the exact flow patterns generated by each geometry.

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